Paper No. 26

VIBRATION TREATMENT OF BO 105 ROTOR

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Summary

The BO 105 helicopter with a four-bladed hingeless rotor has relatively good vibration characteristics for cruise speed and high speed as well. Transition vibration levels are somewhat higher. Modifications of the rotor have been evaluated in a program for possible vibration reduction at the source. Flight test measurements with the basic aircraft are used to determine the origin of the higher harmonic vibrations and to define possible modifications of the rotor. Analyzing the vibratory loads at the rotor hub gives detailed understanding of the working of usable vibration treatment means. Both improved mass distribution of the rotor blades and blade mounted pendulum absorbers have been selected to reduce the higher harmonic vibratory loads at the hub and in the fuselage as well. The effectiveness of the modifications has been proved by flights and by the analyzed measured data.

1. Introduction

The BO 105 is a small helicopter with a gross weight of 2300 kp. The aircraft has a four-bladed hingeless rotor with fiberglass rotorblades. The level of fuselage vibrations at forward speeds has always been relatively good because of the low "g"-level and a comparatively high exciting frequency of 28 Hz. The relatively high frequency is less annoying as a principal forcing frequency than the forcing frequencies of two- or three-bladed rotors or the usually lower forcing frequencies of larger helicopters. The transition vibrations of the BO 105, as with many helicopters, are larger than those in cruise. Analytical studies and experimental programs were conducted to improve the transition vibration levels by reducing the vibratory loads at the source. At first there were cures which can be used together with the production blades, such as tuning weights and pendulum absorbers. The application of such modifications are limited by allowable centrifugal forces of the rotor hub. Therefore a new concept for rotor treatment based on a modification of the production blade was started.

2. Source of Vibration

Usually for a four-bladed rotor the principal forcing frequency is 4/rev. Figure 1 shows the representative vibration characteristic of BO 105. It can be seen that only the 4/rev vibrations are significant. The 8/rev vibrations are of secondary importance. Therefore all vibrations discussed in the this paper are related to 4/rev vibrations.

For a rigid rotor normally all 3/rev, 4/rev and 5/rev components of shear forces and blade moments can induce 4/rev fuselage vibrations. For finding the main source of 4/rev vibration the correlation between vibratory hub loads and vibrations are determined. The cross plots of 3/rev and 5/rev flap bending moments and the 4/rev vibrations of the fuselage - see figure 2 - show the result of this investigation. Obviously there exists a fairly direct relationship between the hub loads and the vibrations. This fact confirmes that the 3/rev and 5/rev hub moments, caused by the flap bending motion are the primary source of the high transition peak of the measured vibrations. Before starting a study how we can influence the vibrations we have measured the spanwise distribution of vibratory bending loads of the main rotor blades at different flight conditions. In figure 3 the 3/rev flap bending moments are listed for some flight conditions which are significant for the transition peak of vibration. The shapes of these forced moment lines over span have nodes very near to the theoretical node of the mode shape due to the 2^{nd} flap bending natural frequency. Figure 4 points out the analogous correlation between the 5/rev forced flap bending motion and the 3^{rd} flap bending natural frequency. These results are the background for the analytical studies.

3. Analytical Studies

Based on the above evaluation, it was concluded that it was necessary to reduce the level of 3/rev and 5/rev flapwise blade bending motion. A possible cure for the production blades consists of using blade mounted pendulum absorbers and blade tuning weights. The application of tuning weights seems to be suitable with resonance problems. In this case they can be very effective. In figure 5 there are plotted the flap bending moment at center of rotation for the production blade and for the same blade with a 5/rev tuning weight and a 3/rev pendulum absorber against the forcing frequency of an exciting force at the blade tip. The plot demonstrates generally the necessity of a 5/rev tuning weight to diminish the dynamic response of the blade at 5/rev. The effect of the position of a tuning weight on the important three harmonic parts (3/rev, 4/rev and 5/rev) are shown in figure 6. The 3/rev and the 5/rev moments are responsible for the moment excitation and the 4/rev moment is responsible for the 4/rev vertical shear forces. It can be seen that there is only one position for influencing the 5/rev moment in a positive way without changeing the 3/rev and 4/rev for the worse. But at this position - ≈2.25 m - the effect of the tuning weight is not optimal. There is the possibility of a better solution by using more than one tuning weight. In our case there was the indication for two weights, the first at a radius of about 2.0 metres and a second at a position of about 2.6 metres.

Both, the 5/rev and the 3/rev tuning weights, could not be realized for a production blade because of the high additional centrifugal force to the production rotor blade. Therefore it was decided not to take a 3/rev tuning weight which would cause a too great centrifugal force but a 3/rev pendulum absorber at a radial position of about 0.8 metres. The influence of both, the 3/rev pendulum absorber and the 5/rev tuning weight is plotted in figure 5. The mode of action of the pendulum absorber follows from figure 7 and figure 8 in more detail. At the position of the pendulum absorber an additional shear force is introduced, see figure 7, which causes a step in the radial shear force distribution and a break in the moment line. The tuning frequency of the absorber is to be selected in such a way that the moment at the center of rotation is equal to zero. In figure 8 it is shown how to determine this optimal tuning frequency for the 3/rev pendulum absorber. The 3/rev pendulum absorber is working optimal if the tuning frequency is somewhat lower than 3 and the phase angle between the amplitude of the absorber and its driving mounting point has a value of 90°. It is important to know that at this optimal frequency the absorber is not working in resonance. On the other hand the optimal tuning frequency of a pendulum absorber depends on the mass of the pendulum and the radial position of its mounting point. The active mass itself is determined on the condition that a known excitation at the mounting point would not produce an amplitude greater than 15° - 20°, which is a reasonable amplitude that avoids excessive detuning due to non-linearity. Figure 9 shows the measured amplitudes of two different 3/rev pendulum absorbers. The heavier one has amplitudes of maximal 17°.

4. Pendulum and Weight Sizing

During a long-term program 3/rev and 5/rev pendulum absorbers as well as tuning weights were sized and built. But at last a 3/rev pendulum absorber in combination with a 5/rev tuning weight had been preferred. Blade flap bending moment at the blade root had been measured for flight conditions due to the transition peak. In a first analysis the equivalent blade tip excitation force that would produce the transition peak was determined. In [1] the dimensioning of the 3/rev pendulum is described in detail. The active weight of the pendulum absorber is about 2 kp and the radial position of the mounting point at 13% rotor radius. For the 5/rev tuning weight 2.7 kp were used. A comparison of the standard blade and the same blade with vibration kit - 3/rev pendulum absorber and 5/rev tuning weight - is given in figure 5. The results show the theoretical improvement for an exciting blade tip force of 100 kp at all driving frequencies.

5. Test Program

The above mentioned vibration kit is shown in figure 10. The evaluation of such a vibration kit is not only measured by the vibration reduction but by the reduction of the vibratory loads of the rotor hub. Figure 11 and figure 12 point out the improvement of the 3/rev and 5/rev hub bending load. The effect of the vibration kit is obvious. The plots include vibratory loads for level flight and 1.5 g turns at 60 up to 100 kts. Similar results are given for the vibrations at the pilot's seat in figure 13 to figure 15. The vibration improvement is higher for flight conditions with higher vibration level.

Every absorber has the disadvantage of moving parts and in connection with it the problem of high costs. Therefore an other blade configuration was proposed which based on a modification of the standard blade weight distribution. A saved weight of about 1.5 kp at the blade tip should enable a new blade device with the same centrifugal force at the blade root.

Figure 16 shows the weight distribution of a vibration treated rotor blade. It is worth noticing that this blade can be produced without changeing the production tools. The weight saved at the blade tip gives the possibility of adding two tuning weights, the first inbord of the blade (1 kp) and the second outbord (2 kp). The analytical study of this vibration treated rotor blade makes one expect a similar vibration reduction as for the standard blade with vibration kit. The mass tuned blades were tested on a preproduction helicopter.

For the investigation of the higher harmonic vibration behaviour in the transition range the test program included level flight and rotor torque (thrust) at 30 kts airspeed. In figure 17 and figure 18 the effect of mass tuning on the vibratory hub moments is presented. The attainable reduction reaches up to more than 50%. As mentioned above blade modifications could introduce higher 4/rev shear forces. Figure 19 demonstrates that there is no effect of mass tuning on the 4/rev flap bending moments. That means that the 4/rev shear forces are unchanged by the vibration treated blades. Figure 20 and figure 21 prove the vibration reduction of the fuselage. The transition vibrations are improved by 50%. The mass tuned blades will be flight-tested on a representative production helicopter in the near future.

6. Conclusion

A vibration improvement kit and a vibration treated main rotor blade have been developed with the goal of reducing the vibratory loads at the source. The effectiveness of both have been demonstrated. The vibration reduction mainly in the transition is obvious. Similar results have been obtained for a standard rotor blade with added pendulum absorber and tuning weight and for a mass tuned rotor blade.

7. Reference

1. R. Gabel, G. Reichert, Pendulum absorbers reduce transition vibration. 31 th Annual National Forum of the American Helicopter Society, Washington, D.C., May 1975

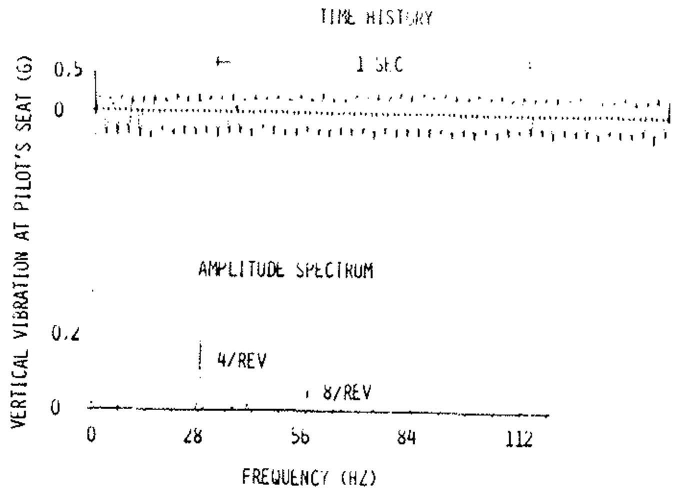


Figure 1: Representative vibration characteristic of BO 105

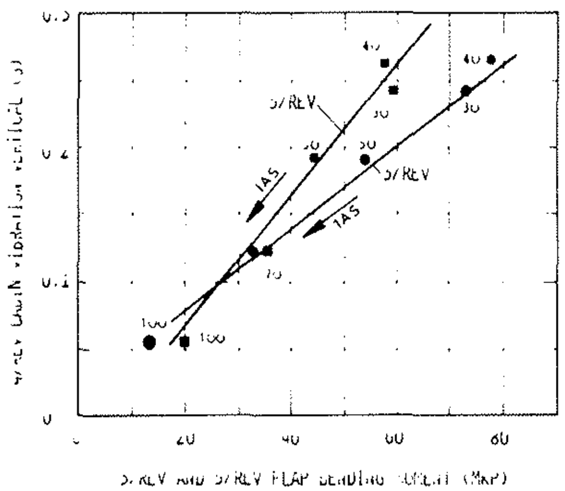


Figure 2: Cross plot of flap bending moment and vibration

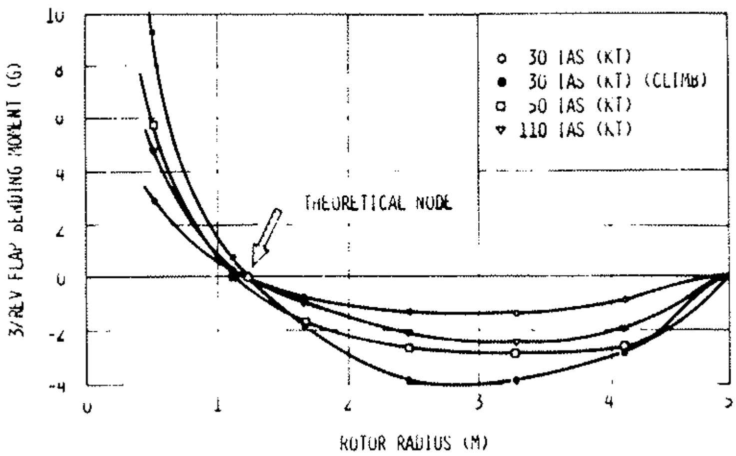


Figure 3: Correlation between 3/rev forced vibration and mode shape due to 2nd flap bending natural frequency

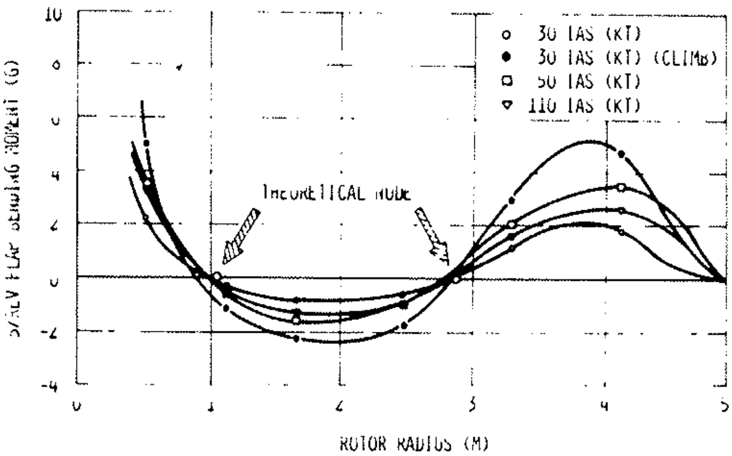


Figure 4: Correlation between 5/rev forced vibration and mode shape due to 3rd flap bending natural frequency

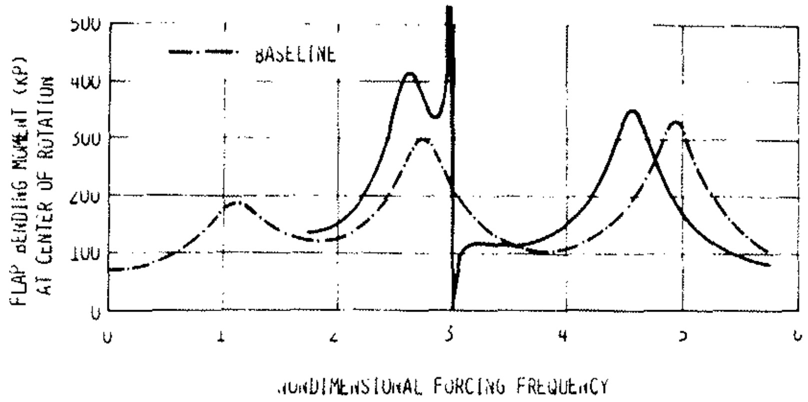


Figure 5: Influence of a 3/rev pendulum absorber and a 5/rev tuning weight on the flap bending moment at center of rotation

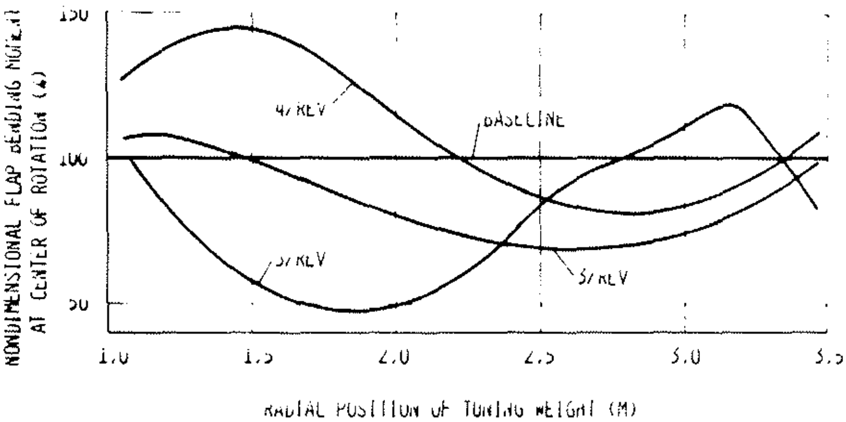


Figure 6: Influence of the position of a 3 kp tuning weight on flap bending moment

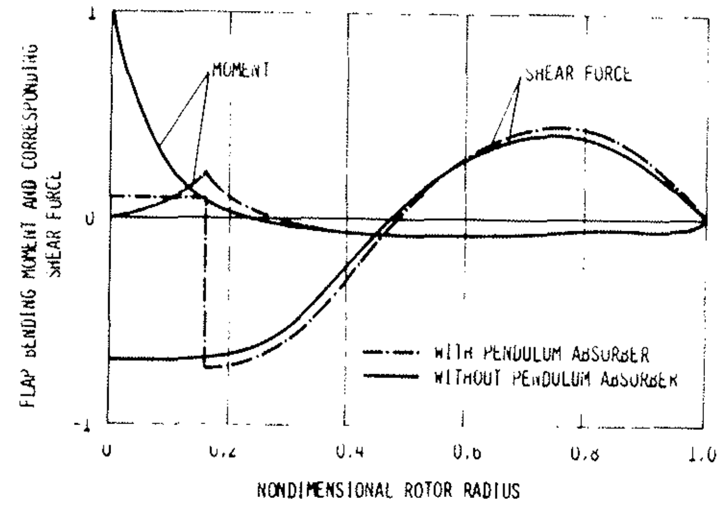


Figure 7: Influence of a 3/rev pendulum absorber on the moment- and shear force distribution of a rotor blade

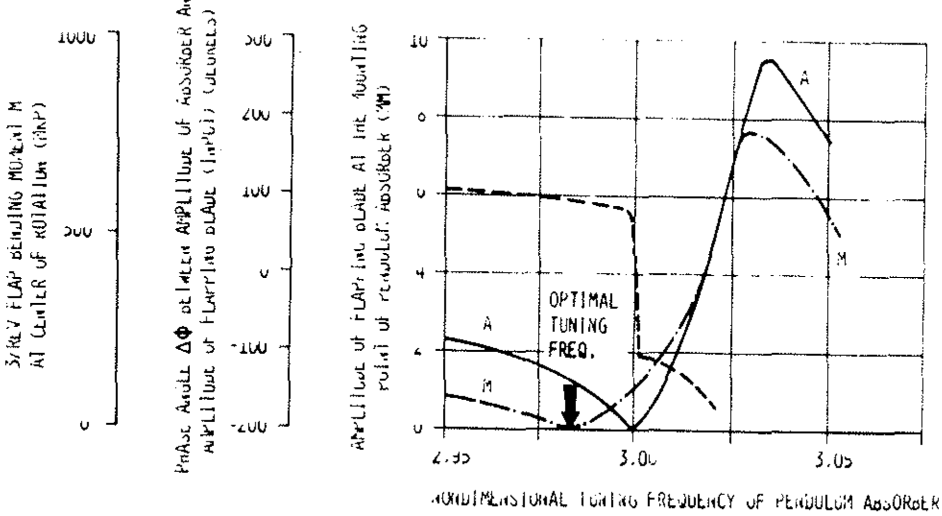
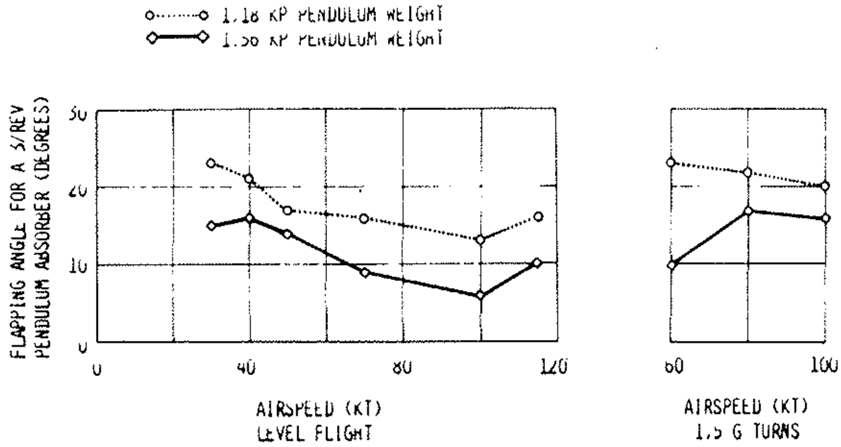
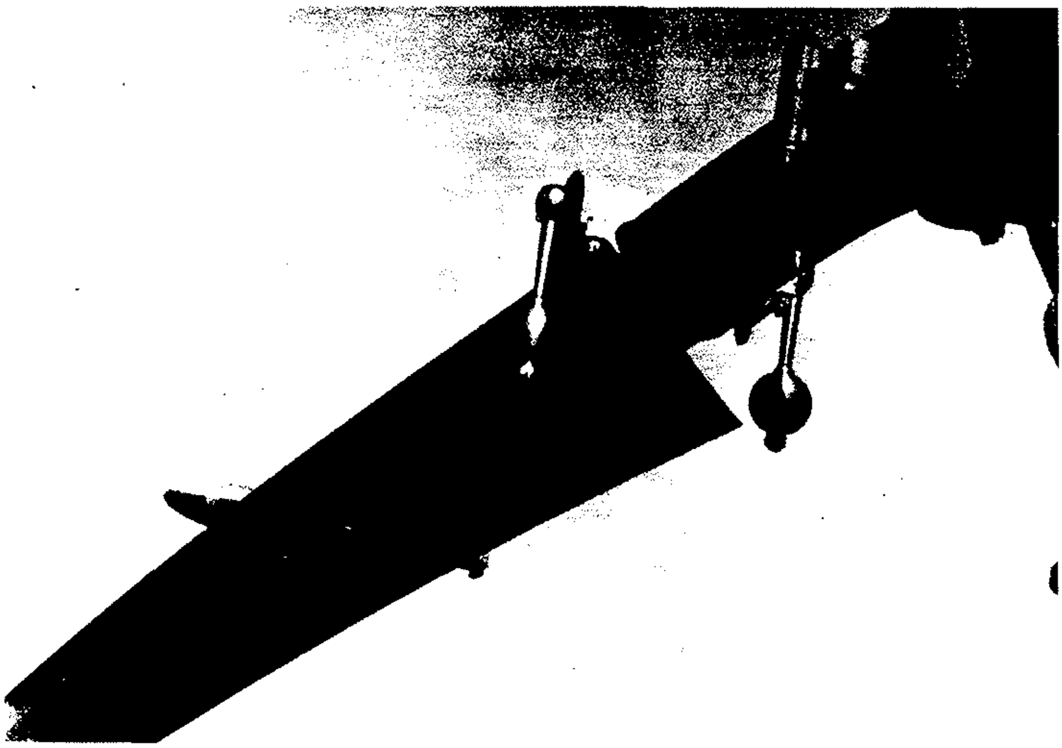


Figure 8: Influence of tuning frequency of a blade mounted 3/rev pendulum absorber on 3/rev blade characteristics due to a 3/rev exciting force at blade tip



<u>Figure 9</u>: Influence of pendulum weight of a 3/rev blade mounted pendulum absorber on the flapping angle of the absorber



<u>Figure 10</u>: BO 105 main rotor blade with 3/rev pendulum absorber and 5/rev tuning weight

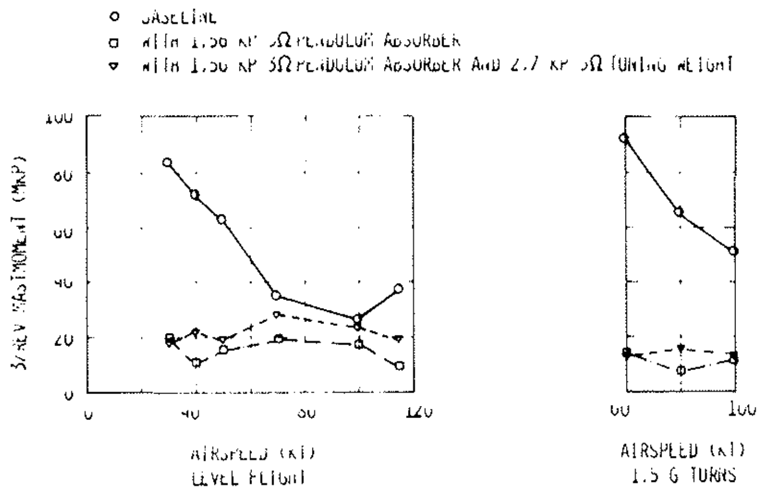


Figure 11: Effect of 3/rev pendulum absorber and 5/rev tuning weight on 3/rev vibratory loads (mastmoment)

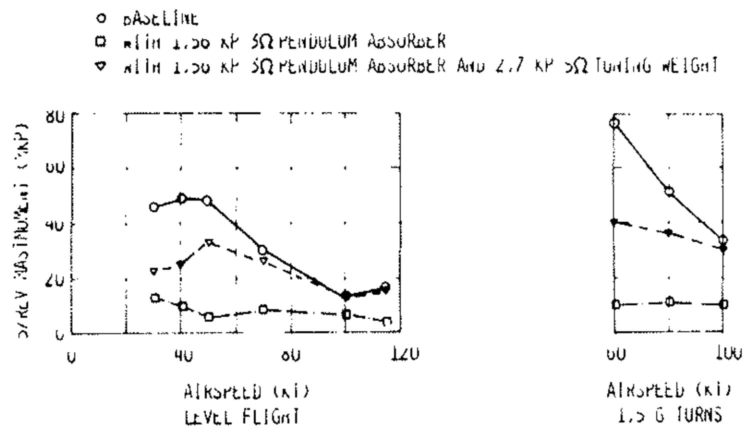


Figure 12: Effect of 3/rev pendulum absorber and 5/rev tuning weight on 5/rev vibratory loads (mastmoment)

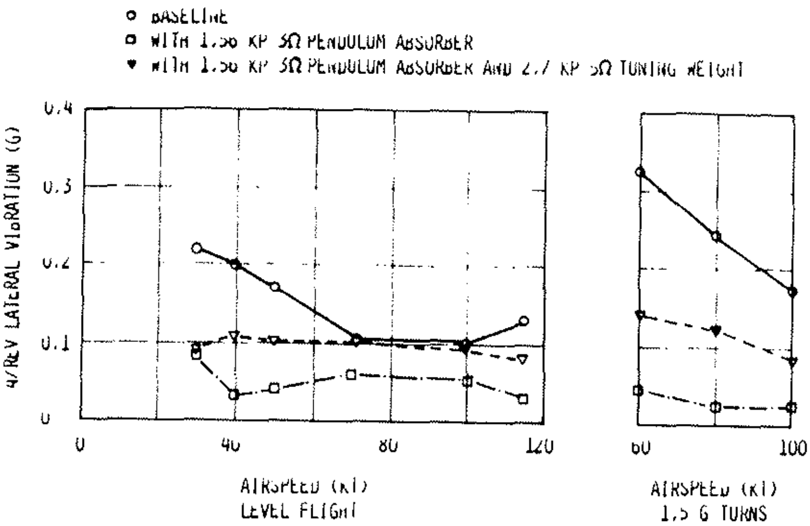


Figure 13: **Effect of 3/rev pendulum absorber and 5/rev tuning weight on 4/rev longitudinal vibration**

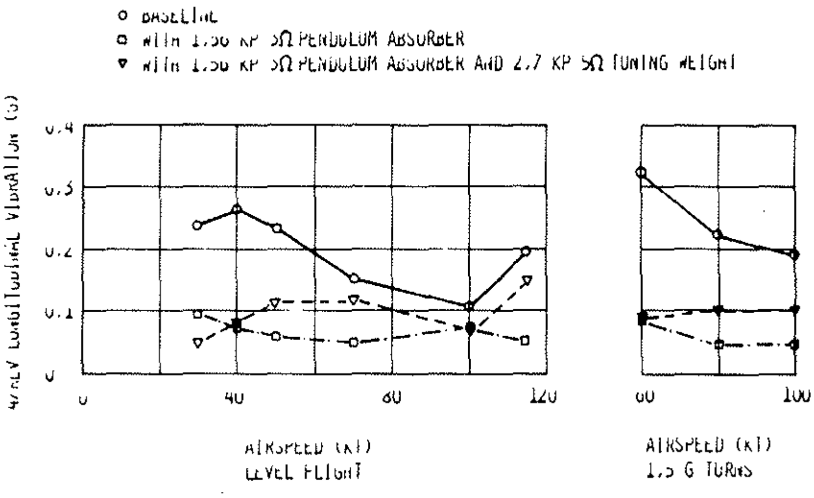


Figure 14: **Effect of 3/rev pendulum absorber and 5/rev tuning weight on 4/rev lateral vibration**

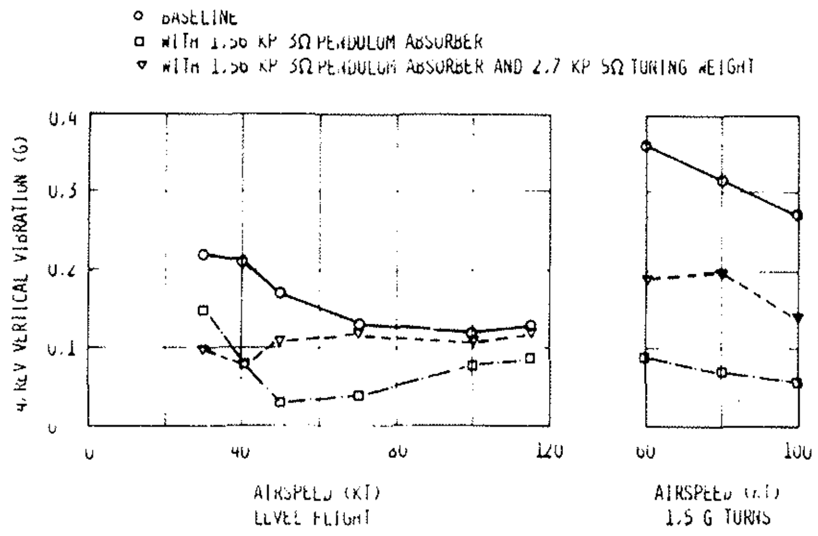


Figure 15: Effect of 3/rev pendulum absorber and 5/rev tuning weight on 4/rev vertical vibration

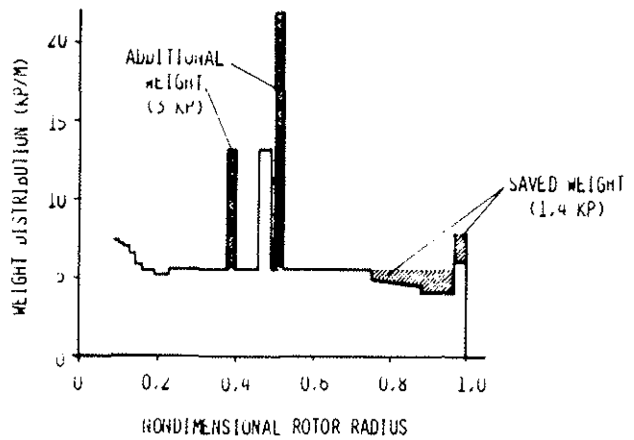


Figure 16: Weight distribution of a vibration treated rotor blade

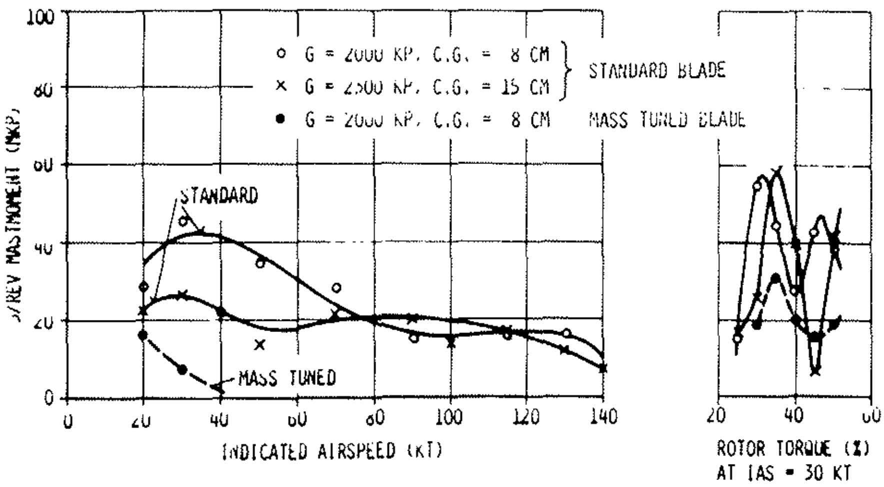


Figure 17: **Effect of mass tuning on 3/rev vibratory loads (mastmoment)**

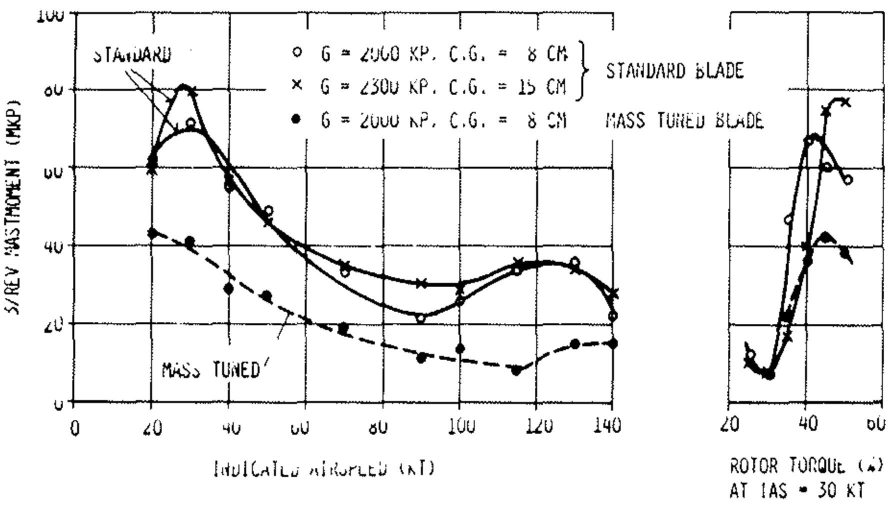
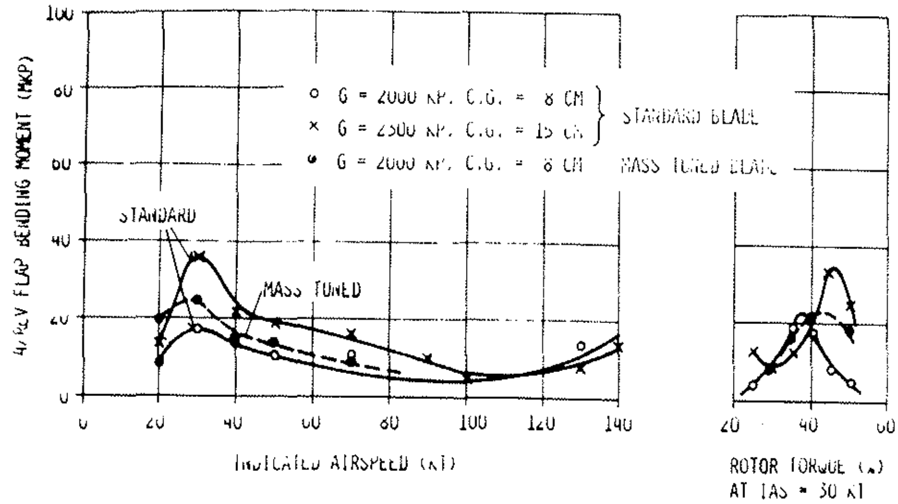
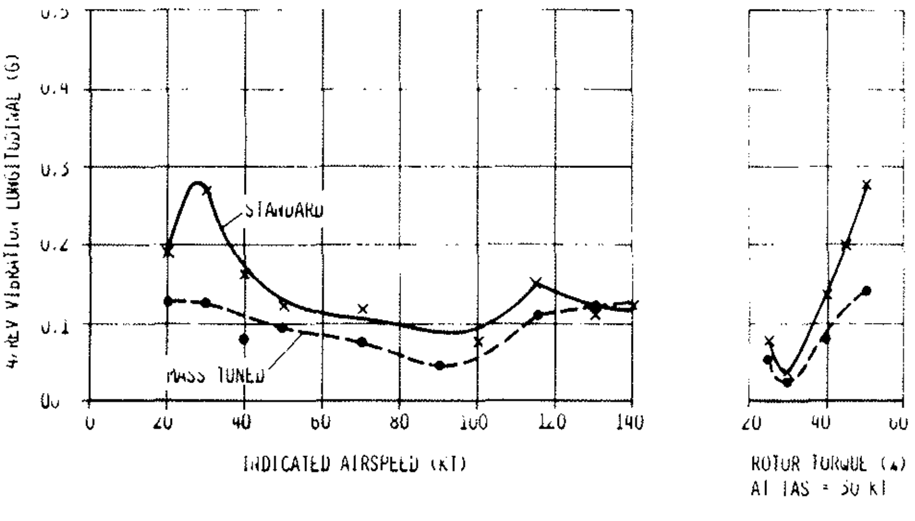


Figure 18: **Effect of mass tuning on 5/rev vibratory loads (mastmoment)**



<u>Figure 19</u>: Effect of mass tuning on 4/rev vibratory (flap bending moment)



<u>Figure 20</u>: Effect of mass tuning on 4/rev longitudinal cabin vibration

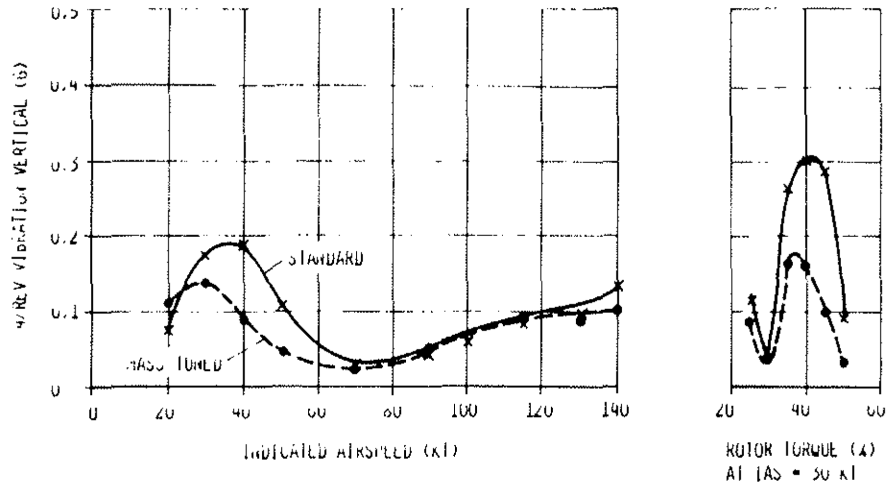


Figure 21: **Effect of mass tuning on 4/rev vertical cabin vibration**