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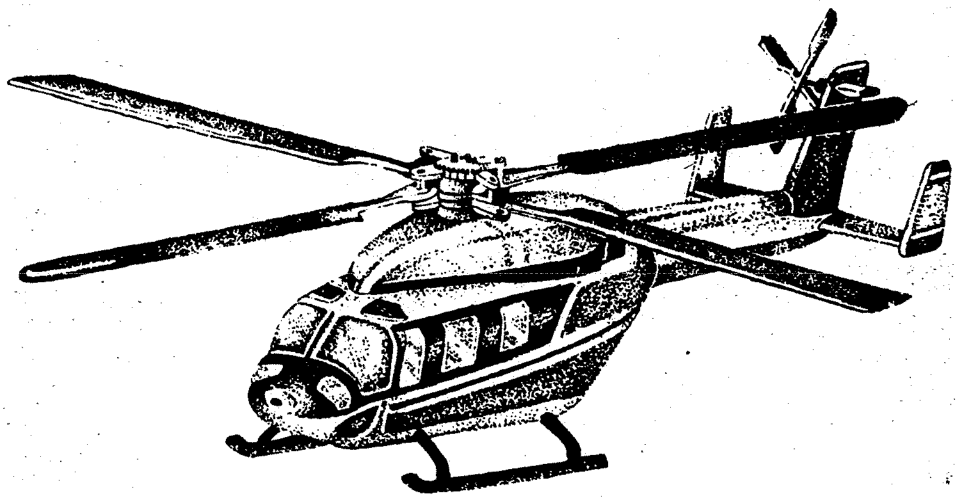

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Organised by

DYNAMICALLY TUNED BLADE PITCH LINKS FOR HUB LOADS REDUCTION

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ABSTRACT

This paper presents an analytical study of a new passive vibration device for helicopters, the dynamically tuned main rotor blade pitch link, which may replace the current pitch link with an axial spring-damper combination. The starting point for this study is an earlier analytical study which had investigated the reduction of vibratory hub loads using the rotorcraft analysis CAMRAD/JA. In this earlier work, hub loads reduction was studied by modification of the blade torsional response via the introduction of large amounts of blade root torsional damping. In the present study a comprehensive rotor analysis code (UMARC) was modified to analyze the loads generated by a main rotor blade with tuned pitch links instead of the conventional pitch links. Including pitch link damping (and stiffness) instead of the modal torsional damping (as done in the earlier work) results in a more practical evaluation since pitch links (with no damping) are an integral part of current production rotorcraft. An analytical study was conducted using the four-bladed S-76 articulated rotor blade. For an advance ratio of 0.38 and a C_T/σ of 0.080, pushrod damping in combination with reduced pushrod stiffness resulted in significant reductions (20 to 25%) in 4/rev fixed system hub loads (longitudinal inplane shear, rolling and pitching moments), with no change in the lateral inplane and vertical shears. At the same time, the 1/rev pushrod loads increased by approximately 50%. The design of a dynamically tuned blade pitch link may involve the redesign of a production pushrod in order to accommodate the required stiffness/damping; and any additional fatigue considerations arising from the increased 1/rev loads can be included in this redesign phase. The present reductions in hub loads signify that the dynamically tuned blade pitch link is a promising concept.

INTRODUCTION

Since the early days of rotorcraft development, vibration reduction has been a central topic of research. This will continue to be the case until vibration levels comparable to those in fixed wing aircraft can consistently be achieved without an excessive weight penalty. Research in rotorcraft vibration reduction techniques is motivated primarily by the need to maximize structural component life, reduce crew fatigue and provide a more comfortable environment for passengers. Research in vibration reduction has focused on two main approaches:

1. Structural optimization, i.e. structural design or modification to achieve favorable fuselage and blade dynamic characteristics.
2. Active vibration control (higher harmonic and/or individual blade control using hydraulic actuators/actively controlled fixed system actuators).

Vibration prediction is a complex multidisciplinary problem and existing vibration prediction analyses are not reliable enough to guarantee success of structural optimization approach. Also, the approach is limited to the design phase of a new aircraft. In the case of existing aircraft, structural optimization is unattractive since complete redesign of major structural components is typically involved. Finally, the aircraft is subjected to a wide variety of loadings and operating conditions and it is difficult to arrive at an optimum design which satisfies all conditions. Active control is also promising but has drawbacks in the form of weight penalty and additional power requirements. Also, the maintainability and reliability aspects of active vibration control systems may make them less attractive in many applications.

A third category of vibration reduction technology is comprised of discrete passive devices, such as pendular absorbers in the rotating system, antiresonance isolators for gearbox isolation, spring-mass absorbers in the fuselage, etc. Although these devices also involve a weight penalty, and in some cases increased maintenance requirements, they are in general simple, relatively inexpensive, and can applied on an as-needed basis in combination with other vibration reduction measures. Thus, regardless of technology advances in structural design methodology or active control, it seems advantageous to have a selection of effective passive devices available. This paper examines the vibration reduction potential of one such passive device, the dynamically tuned spring-damper pushrod.

An early study by Miller and Ellis (Ref. 1) examined the effects of torsional frequency on blade root shears. A simple rigid blade model was used and the torsional frequency was varied by adjusting a root torsional spring. The model thus applies to the case of a variable stiffness pushrod. The study showed that reductions in root torsional spring stiffness could lead to substantial reductions in blade vibratory shears. Subsequent investigators (for example, Refs. 2 and 3)examined the influence of blade torsional frequency cnblade response. However, these studies were directed more towards reduction of the control system vibratorv loads associated with stall flutter than to reduction of hub loads and fuselage vibration.

A spring-damper pushrod to modify blade torsional dynamics was first investigated in the early 1970's at Sikorsky Aircraft (Ref. 4), again with the goal of reducing vibratory loads in the control system arising from stall flutter. The spring and damping values were selected based on an analytic investigation of a single flight condition known to produce high stall-induced vibratory loads. The investigation culminated in a flight test of a set of spring-damper pitch links on a CH-54B helicopter. The devices were quite effective; at high speed, vibratory control loads in the rotating system were reported reduced by nearly 50%. The cockpit vibration levels were unchanged, but it is not clear whether this was based on pilot comments or on actual vibration measurements.

Recently, Kottapalli (Ref. 5) has suggested that the introduction of large values of torsional damping at a discrete location near the blade root could reduce blade elastic motions and vibratory hub loads. The study was conducted using a full elastic blade analysis (CAMRAD/JA, Ref. 14). The torsional damper was represented by applying an equivalent damping to the first elastic torsion mode. The effects of applying the damping at a discrete location were not investigated. No specific damping device was discussed, but it is clear that tuned spring-damper pushrod such as that tested in Ref. 4 could be adapted to the purpose. This is an attractive possiblity since the tuned spring-damper pushrod replaces the conventional pushrod and therefore can be installed in both new and existing rotorcraft.

Reference 13 further examines the possibilities of vibration reduction via pushrod tuning. Unlike the Ref. 4 study, the analysis focuses on the influence of tuning on vibratory hub shears rather than stall-induced control loads. The pushrod is represented as a discrete element and the effects of pushrod damping on the blade root boundary condition are realistically represented. The investigation is in several parts. First, the rigid blade pitch-flap analysis of Miller and Ellis (Ref. 1) is repeated using a trimmed forward flight model. Then, a more extensive investigation with a comprehensive rotor analysis is conducted. Measured and predicted pushrod load data are compared to validate the analytic model. The effects of pushrod stiffness and damping on fixed and rotating system hub loads are examined.

The present study, using the same analytical results as of Ref. 13, provides additional interpretations of the dynamically tuned blade pitch link concept.

RIGID BLADE PITCH-FLAP ANALYSIS

Prior to the actual investigation, the rigid blade pitch-flap analysis of Miller and Ellis (Ref. 1) was repeated keeping in mind the present context. The goal was to gain insight into the pushrod tuning problem using a simple model, eliminating complicating factors such as blade elastic motions, pushrod kinematics and blade twist. Unlike Ref. 1, the present analysis directly includes the effects of forward flight. The physical model (Fig. 1) is a rigid blade, free to flap and pitch about centrally located coincident hinges. The blade is restrained at the root through a torsion spring. Collective and cyclic pitch inputs are applied to the blade through the torsion spring to trim the rotor to a prescribed value of thrust and zero first harmonic flapping. The rotor shaft angle is adjusted to achieve propulsive force based on an assumed fuselage equivalent flat plate area. The flapping frequency may be adjusted with an assumed flapping hinge spring. The model assumes a flap pitch hinge sequence and zero pitch-flap (δ_3) coupling. The equations of motion are solved with the finite element in time method, and blade loads are computed via a force summation scheme.

The study was conducted assuming a blade with characteristics in Table 1. This is essentially the blade examined in Ref. 1. The first flapping frequency was set at 1.05/rev, typical of an articulated blade.

Figure 2 shows blade root loads and fixed system hub loads as a function of torsional frequency for a coupled trim condition at μ = 0.4. Note in this study only pushrod stiffness variations (i.e. variations in torsional frequency) are investigated. The torsional damping is set zero. In Figure 2(a), the 3/rev load shows a considerable reduction over the baseline value (ν_θ = 5) as the rotational torsional frequency approaches 3/rev. A sudden fourfold increase in the 3/rev blade vertical shear is encountered as the torsional frequency is further lowered to just under 3/rev. There is a very slight reduction in the 2/rev load local to ν_θ = 3. The 4/rev vertical load is coupled to 3/rev blade pitching motion due to large l/rev variation in dynamic pressure and hence also shows a large peak at this torsional frequency. There is also a small reduction in 3/rev shear when the torsional frequency nears 4/rev. These basic features observed in the 3/rev shear are also present in the 4/rev vertical shear near ν_θ = 4 and in the blade root moment results in Fig. 2(b). The peaks in amplitude, however, are of lesser magnitude.

Strictly speaking, since this is a centrally hinged blade, the 3/rev vertical shears would be of interest from a vibration point of view only with three blades. However, with hinge offset, these vertical shears produce 3/rev rotating system hub moments contributing to fixed system moments with four blades.

Figure 2(c) shows fixed system 4/rev hub moments, assuming a four bladed rotor. Note the values of c/R and CT chosen for the single blade yield CT = .0064 and (CT/σ = 0.l0 for a four bladed rotor). The slight reduction in fixed system moments near $\nu\theta$ = 3 (compared with the baseline, ν_θ= 5) is due to the corresponding reduction in 3/rev rotating system loads (figure 2(b)).

Identifying a single combination of pitch, flap, and airloads harmonics which are responsible for the reduction in vibratory load near ν_θ= 3.05 may not be possible. This is a two degree of freedom system; the flapping and torsion degrees of freedom are coupled through mass, damping, and stiffness terms and both contribute to the vibratory load through inertia terms. Therefore one should not expect a simple, single degree of freedom resonance phenomenon resulting in a sharp peak in response or a 90' phase shift. Indeed, the local minimum in vibration and associated side peak is reminiscent of the characteristic of a spring-mass vibration absorber (e.g., Ref. 6).

Nevertheless, these results tend to confirm the conclusion of Ref. 1, that is, for this rigid blade model, vibration reduction is indeed possible by proper selection of the blade torsional frequency. The optimum frequency lies just above 3/rev. The underlying effect is not a simple resonance phenomenon; the relative phasing of the various harmonics of the loads may play an important role. A vibration penalty may result from operating slightly off the design point.

ELASTIC BLADE ANALYSIS

In Ref. 5 the effects of the torsional damping were represented by an equivalent modal torsional damping of the first elastic torsional mode of the rotor blade. This was done using CAMRAD/JA (Ref. 14) in which there is no provision to directly model blade root torsional damping. This approach in essence assumes that the damping is distributed uniformly over the length of the blade. Although this simplifying assumption is appropriate for a preliminary study such as Ref. 5, in the present analysis it was considered essential to model the pitch link as a discrete dynamic element applied at its proper location. The spring/damper pitch link changes the blade root boundary conditions and affects the blade root torsional dynamics in a manner that may not be properly captured if uniform distributed damping is assumed.

The present investigation was conducted using UMARC, a comprehensive rotor aeroelastic analysis based on finite elements in time and space (Ref. 7). UMARC was modified in Ref. 13 in the following manner. The spring-damper pushrod (Fig. 3) was modeled by modifying the analysis to release the blade root boundary constraint. Appropriate energy terms were added to the stiffness and damping matrices. These include diagonal terms on the root pitch degree of freedom due to the pushrod stiffness and damping, and both diagonal and coupling terms on the blade flap degree of freedom arising from the δ_3 coupling. The pushrod motions were calculated from the control inputs and the elastic blade root pitch and flap deflections. The pushrod loads were obtained from the pushrod motions together with the known pushrod stiffness and damping values.

<u>Validity of modal approximation with damped boundary condition</u>

The analysis is configured to apply a modal approximation to the equations of motion. The basis of the modal reduction was the set of normal modes obtained with only the stiffness and mass terms in the equations of motion. This leads to another modeling issue, namely, how well can these undamped normal modes represent the blade dynamics in the presence of a damped boundary condition? Physically, one would expect that given enough damping, the blade root motion would be restricted to the point where a cantilever boundary condition would apply, with a corresponding increase in torsional frequency. This is the "bridging" phenomenon mentioned in Ref. 4. However, the modes obtained without the damper include deflections at the blade root and do not satisfy the cantilevered boundary condition.

In Figure 4, a uniform elastic rod is supported at one end with a spring and damper arranged in parallel. Undamped normal modes for this system are calculated from the closed form solution for a spring supported torsion rod. A set of the first N of these modes are then used to synthesize the system including the damper. The resulting modal damping matrix is fully populated. The calculated frequency for the first damped torsion mode is shown in Figure 4 as a function of the root damping coefficient. In the figure the damping coefficient has been normalized by $\sqrt{GJ\mu}$ and the frequencies by $(\pi / 2)\sqrt{GJ / \mu\ell^2}$, the exact solution for the cantilever case. The figure shows that above a certain value of the root damping coefficient, the predicted frequency is above that for the cantilever case. As expected, this overprediction becomes less severe as the number of modes is increased. However, for all the cases shown, as long as the calculated torsional frequency remains below $\omega_d / \omega_{r\ell}$ =1, there is relatively little sensitivity to the number of normal modes used. In the analysis to follow, the blade is represented by the first seven modes. Although this includes only one mode which can be categorized as a "pure" torsion mode, it includes two bending modes which involve significant amounts of torsional motion due to structural twist. It will be shown that this set of normal modes together with the pushrod damping values of interest result in only moderate increases in torsional frequency, and it is concluded that the error due to using undamped normal modes is insignificant (Ref. 13).

RESULTS

Subject aircraft

The remainder of the paper will examine the effects of varying pushrod stiffness and damping of a typical helicopter rotor, in this case the a Sikorsky S-76. A full scale S-76 main rotor was tested in the NASA Ames 40- by 80-Foot Wind Tunnel in the late 1970's (Ref. 8), providing experimental data for verification of the analytical results. The design characteristics are given in Table 2. In Ref. 8, four tip planforms were tested; the present investigation assumes the rectangular tip configuration in order to minimize modeling issues related to three-dimensional unsteady effects at the blade tip. Detailed data for the analytic model are available in Refs. 8 and 9.

The blade pushrod stiffness of the baseline aircraft is based on the control system stiffness given in Ref, 9, together with the assumption that this stiffness is entirely determined by the pushrod stiffness with no flexibility in the swashplate or servos. This assumption is adequate for the present feasibility study; nevertheless, the actual aircraft the control system stiffness may be affected by the swashplate and servo stiffness.

According to the normalization scheme used in the present analysis (Ref. 7), the nondimensional pushrod stiffness is defined as $\overline{k_p} = k_\theta \ / \ m_o \Omega^2 R^3 (a_p / R)^2$ with the reference mass distribution m_o defined as $m_o = 3I_\beta / (1-\varepsilon)^3 R$

This is the mass distribution which, if it were constant along the blade span, would give the actual flapping moment of inertia, I_β. Based on the mass data in Ref. 9, m_o for this blade amounts to 0.126 slug/ft, yielding a nondimensional value of k_p of 31.2 for the baseline aircraft.

Correlation with test data

To validate the analysis, the measured pushrod load time histories from Ref. 8 were compared with the analytic results for the same operating conditions. Figures 5(a) and 5(b) compare the measured and analytic results for advance ratios of 0.2 and 0.38 respectively. In each case, the rotor was trimmed to zero first harmonic flapping and the specified CT/σ,. The data have been adjusted to zero steady component to facilitate comparison of the vibratory components of the waveforms. Analytic results are shown using quasisteady aerodynamic modeling, and secondly, unsteady circulatory aerodynamic terms.

The Drees inflow model (Ref. 15) was used in this correlation study and the all of the results in this paper are based on this inflow model.

For each case, the overall correlation with the measured data is fair At both $\mu = 0.2$ and $\mu = 0.38$, the 1/rev vibratory components appear fairlv well matched. However, at $\mu = 0.2$, the test data exhibit a small higher harmonic component with a frequency near the blade first torsional frequency which is not present in the calculated results. At $\mu = 0.38$. the data differ noticeably in phase over the retreating portion of the rotor disk. Nevertheless, the main features of the torsional response are present in the predicted time histories, namely, the large excursion on the advancing side near 2/rev and the presence of response at the torsional frequency on the retreating side.

Only small differences are observed in the two sets of analytic results. Hence, quasisteady aerodynamics will be used for the remainder of this study. Inclusion of the stall model would probably not improve the overall correlation. Stall flutter is not a significant factor in the experimental data since the large "spikes" in pushrod load typically occurring in the aft retreating quadrant of the rotor disk (see for example Refs. 4 and 3) are entirely absent. Also, significant deviations between the measured and predicted time histories are observed near $\psi = 180^o$; this is not a region on the rotor disk where stall is expected to be a significant factor. That the data correspond to a fairly moderate values of CT/σ also suggests that the stall model is not essential in this case.

Jepson et. al. (Ref. 10) conducted an extensive correlation study using data from the tests documented in Ref. 8. One conclusion of this study was that the fuselage flow field can have a significant effect on blade and pushrod loads. Based on Ref. 10, including flow effects of the test stand body might improve the correlation in Figure 5.

In the present analysis the control inputs are assumed to take place about the blade undeformed axis; this corresponds to a hinge sequence with the feather axis inboard of both the flap and lag hinges. However, in the subject aircraft the pitch bearing flaps and lags with the blade spindle. Also, there is a small amount of pitch-lag coupling due to the pushrod kinematics which was neglected in the analysis. Instead of tabulated airfoil data (such as may be found in Refs. 8 and 9), the blade airfoil is aerodynamically represented in the analysis by the analytical expressions: $c_\ell = c_{\ell_0} + c_{\ell_\alpha}\alpha$, and c_d and $c_{m_{ac}}$ are constants derived from the tabulated data near $\alpha = 0$ deg. Finally, note that at this operating condition the cyclic and collective pitch settings are not prescribed. The differences in the predicted and measured loads may be attributed in part to differences in predicted and actual trim controls. Neither Ref. 8 nor Ref. 10 report these data making it practically impossible to arrive at any conclusions regarding correlation of the trim control predictions.

In summary, an actual case study of pushrod tuning for a specific aircraft would necessitate certain refinements to the analysis. However, the present investigation is considered a feasibility study; the major features of the blade torsional response, insofar as they may be affected by varying blade pushrod stiffness and damping, are well predicted. The qualitative results of the present study may hold after refinements to the analysis are implemented and remain valid for the subject aircraft.

Effect of pushrod stiffness and damping on blade dynamic characteristics

Figure 6 shows the effects of decreased pushrod stiffness on rotating blade natural frequencies. The blade dynamics are characterized by the close proximity of three strongly coupled normal modes. The modes are therefore labeled according to their mode shapes at the baseline pushrod stiffness. At very low values of pushrod stiffness, it is the low frequency branch that has the nature of a first torsional mode. As $\overline{k_p}$ is reduced to a nondimensional value of 5, this mode rapidly approaches 3/rev and the dynamics of the mode become increasingly dominated by the pushrod stiffness. Figure 7 compares the torsional mode shapes of the first torsional mode for two cases, the baseline stiffness and a reduced stiffness ($\overline{k_p}$ = 5). At the lower torsional stiffness the mode approaches the nature of a rigid body feather mode. This is to be contrasted with changes in torsional dynamics accompanying reductions in blade torsional stiffness. For the reduced torsional stiffness, the softness of the blade relative to the root torsional restraint causes the mode shape to approach that of a blade with infinitely stiff root restraint.

Figure 8 shows the effect of varying pushrod stiffness and damping on in-vacuo frequency and damping of the blade first torsional mode. As with Fig. 6, there are actually three modes that may be candidates for the first torsional mode. When generating Fig. 6, an attempt was made to select the mode most resembling a "pure" torsion mode. Hence, the various points on the $\overline{k_p}$-$\overline{c_p}$ map in the figure do not correspond to a single locus of frequency roots. At and above $\overline{k_p} = 20$, the damped natural frequency is very close to the baseline value of 5.5/rev. As expected, as the pushrod stiffness is decreased, the pushrod damping becomes more effective at increasing the damping ratio of the torsional mode.

As the mode shape changes in order to involve more displacement at the root (Fig. 7), increasing energy per cycle can be dissipated through blade root damper motion. The damped natural frequency also becomes more sensitive to pushrod damping at low values of $\overline{k_p}$. The figure shows that even at the lowest values of pushrod stiffness, below $\overline{c_p}$ of approximately 2 - 3, the calculated damped frequencies are well below the rigid pushrod case, indicating that the error due to using undamped normal modes may be neglected up to these damping values (compare with Figure 4).

Effect of pushrod tuning on vibratory hub loads

Figure 9 shows the effects of blade tuning on fixed system 4/rev loads in wind tunnel trim at $\mu = 0.38$ (this operating condition, together with the $\mu = 0.20$ condition in Fig. 11, was chosen to match the conditions in the wind tunnel test described above). The forces have been normalized by $m_o\Omega^2R^2 =$ 57,600 lb and the moments by $m_o\Omega^2R^3 = 1.27 \times 10^6$ ft-lb.

With zero pushrod damping, little effect is observed for a $\overline{k_p}$ reduced to approximately 10, at which stiffness value the fixed system loads tend to increase with a further reduction in $\overline{k_p}$, in some cases dramatically. For reference, the point at which the first torsional frequency crosses through 4/rev is indicated with a bold symbol ("•"). Although the 5/rev loads tend to increase most sharply at low pushrod stiffness, significant increases in 3 and 4/rev loads are also observed. At very low values of $\overline{k_p}$ the data were limited by difficulties in obtaining a trim solution. This is attributed to effects on control system effectiveness, discussed below.

With the introduction of a moderate amount of damping ($\overline{c_p} = 2$) the trends in the fixed system vibratory loads are reversed (for reference, a combination of $\overline{c_p} =$ 2 and $\overline{k_p} = 5$ yields a damping ratio of just over 25% for the first torsion mode - see Fig. 8). In the case of the longitudinal inplane shear and the hub pitch and roll moments, reductions ranging from 25% to 50% over the baseline case ($\overline{k_p} =$ 31.2, $\overline{c_p} = 0$) are observed. A further increase of damping to $\overline{c_p} = 3$ resulted in little improvement in the hub loads. Apparently most of the beneficial effects of damping are obtained with levels of damping sufficiently low that errors due to using undamped normal modes may be neglected.

The sharp increase in fixed system vibratory load in the zero damping case below $\overline{k_p} = 10$ may be directly seen from the rotating system loads in Figure 10. The figure also shows that the reduction in fixed system 4/rev load at $\overline{c_p}$ = 3 and $\overline{k_p}$ < 10 is associated mainly with reductions in rotating system 3/rev inplane shears. The vibratory moments and the $\overline{c_p}$ = 2 results were omitted for clarity.

Figure 11 shows fixed system 4/rev hub loads for μ= 0.20. Again at this advance ratio, favorable results may be obtained at combinations of low pushrod stiffness and moderate pushrod damping.

Based on the hub loads results, introduction of damping has almost no effect at the baseline pushrod stiffness. Also, the large peaks in amplitude observed in the rigid blade study when operating slightly off the optimum pushrod stiffness (Fig. 2) are not present in this elastic blade analysis results.

<u>Influence on pushrod loads</u>

In Figure 12, the 1 and 2/rev pushrod loads are shown as a function of pushrod stiffness for zero damping and $\overline{c_p}$ = 3. At μ = 0.38, the 1/rev pushrod load increases by approximately 50% as the pushrod stiffness is reduced from its baseline value to $\overline{k_p}$ = 5. The 2/rev is less significantly affected but still increases by approximately 20%. Addition of pushrod damping appears to reduce 2/rev loads slightly; the l/rev loads, however, remain virtually unaffected. These phenomena are much less pronounced at μ = 0.2. Here a slight improvement in 2/rev loads at low values of $\overline{k_p}$ is observed.

The large increase in vibratory pushrod loads at low pushrod stiffness at μ = 0.38 is significant. Although the pushrod would be replaced by an entirely new component sized to handle these vibratory loads, the pushrod loads have implications for loads in other control system components such as the swashplate and servos. It is envisioned that the present device may be retrofittable to existing aircraft; however, this advantage disappears if other control system components require redesign or "reduced time to replacement".

The hub loads shown in this study were calculated using a force summation method; no distinction is made between loads reacted through the hub and loads reacted through the pushrod. This distinction, however, is of potential interest. The vibratory pushrod load feeds into the fixed system through the non-rotating part of the control system. Changes in hub loads may have a different effect on fuselage vibration depending on whether they are reacted through the rotor shaft or through the pushrod. To properly capture this effect would require the fuselage and nonrotating control system to be modeled in some detail (see Refs. 11 and 12 for a discussion of the effects of control loads on fuselage vibrations).

<u>Effect on trim control settings</u>

Figure 13 shows the effects of pushrod tuning on trim control positions. At both μ= 0.20 and μ =0.38, reduced pushrod stiffness has little effect on collective pitch required. An increase in forward longitudinal cyclic is present, especially at μ = 0.38. Of particular interest is the trend of lateral cyclic at μ = 0.38. At very low values of $\overline{k_p}$, the lateral cyclic drifts by about 5 deg; in the neighborhood of $\overline{k_p} = 8$ it changes sign. This is probably why difficulties were encountered when finding the trim solution at very low values of pushrod stiffness. In the present analysis, the control positions were adjusted to yield zero first harmonic flapping using a tangential matrix obtained from a rigid blade model. Apparently, the reduced pushrod stiffness cuases a phase delay in the blade flapping response, changing the system response to cyclic pitch in such a way that the rigid blade tangential matrix no longer guarantees trim convergence. Introduction of pushrod damping alleviates this situation. Difficulties in finding a trim solution may also arise from actual instabilities introduced by reducing the pushrod stiffness (aeromechanical stability was not examined in this investigation). Pushrod tuning may or may not have a significant effect on trim controls of a free flying aircraft (the present investigation has been limited to the wind tunnel trim case in order to match the test operating conditions in Ref. 8).

Practical implementation

Regarding a practical implementation of a spring damper pushrod with desireable stiffness and damping values as identified in this study, consider the devices described in Ref. 4. They provide useful data points as to what dynamic properties are possible with such a device. The pushrods had a spring rate of 5000 lb/in and a damping rate of 90 lb-sec/in. These values may be nomdimensionalized in the present scheme with

$$\overline{k_p} = 60,000 \text{ lb/ft} \div m_o\Omega^2 R \cong 23$$
$$\overline{c_p} = 1080 \text{ slug/sec} \div m_o\Omega R \cong 13$$

Presumably the damping rate could be arbitrarily reduced to the desired value of $\overline{c_p} = 2$ to 3 by modification of the orifice size. More difficulty may be encountered achieving the low spring rate required ($\overline{k_p} = 5$). In Ref. 4, elastomeric elements were used to provide the required compliance. This suggests that an integrated elastomeric spring-damper pushrod may hold promise. Possibly the required damping can be provided by the elastomer itself, eliminating the need for a hydraulic damper.

CONCLUDING REMARKS

An analytical study on the reduction of helicopter rotor hub loads was conducted. The concept of dynamically tuned main rotor blade pitch links (also commonly referred to as pushrods) was analytically examined. The dynamically tuned blade pitch link is a passive device in which a rotor blade pitch link is replaced by a damper/spring element.

An analytical study was conducted using the four-bladed S-76 articulated rotor blade. For an advance ratio of 0.38 and a C_T/σ of 0.080, pushrod damping in

(20 to 25%) in 4/rev fixed system hub loads (longitudinal inplane shear and rolling and pitching moments), with no change in the lateral inplane and vertical shears. The 1/rev pushrod loads increased by approximately 50%. The design of a dynamically tuned blade pitch link may involve the redesign of a production pushrod in order to accomodate the required stiffness/damping; any additional fatigue considerations arising from the increased 1/rev loads can be included in this redesign phase.

The present reductions in hub loads signify that the dynamically tuned blade pitch link is a promising concept.

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Table 1: Blade Model Parameters for Loads Data in Fig. 2

Flapping frequency	ν_β	1.05/rev
Baseline torsional frequency	ν_θ	5/rev
Lock number	γ	8
Blade chord	c/R	.05 (solidity = .0159 per blade)
CG offset	e_{cg}/R	.0005 (fwd. of feather axis)
AC offset	e_{ac}/R	.00025 (fwd. of feather axis)
Moment of inertia about feather axis	I_θ/I_β	.001
Torsional damping	ζ_θ	0
Pitch-flap coupling	δ_3	0
Profile lift curve slope	c_{l_α}	6.28
Profile lift at zero angle of attack	c_{l_0}	0
Profile pitching moment coefficient	c_{mac}	0
Profile drag coefficient	c_{d_0}	.01
Fuselage equivalent flat plate area	$f_e/\pi R^2$	.0025

Table 2: Main rotor basic design data, Sikorsky S-76 (Refs. 8 and 9)

Number of blades	N_b	4
Blade tip		Rectangular (wind tunnel test only)
Blade torsional frequency	ν_θ	5.5/rev (calculated with baseline pushrod)
Solidity	σ	.0748
Lock number (nominal)	γ	10.8
Rotor speed	Ω	30.7 1/s
Tip speed	ΩR	675 ft/s
Flap and Lag Hinge Offset	ϵ	3.8%
Blade Pitch-Flap Coupling	δ_3	17°
Pitch Bearing Torsional Stiffness	$k_{\theta PB}$	680 ft-lb/rad
Control System Stiffness	k_θ	24000 ft-lb/rad
Pitch horn arm	a_{PH}/R	.0246

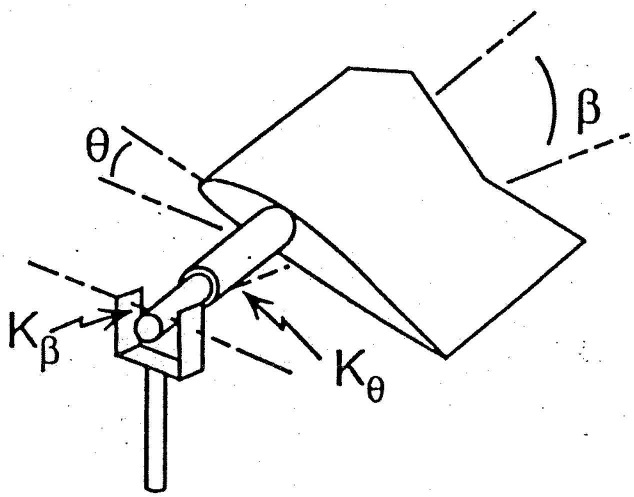
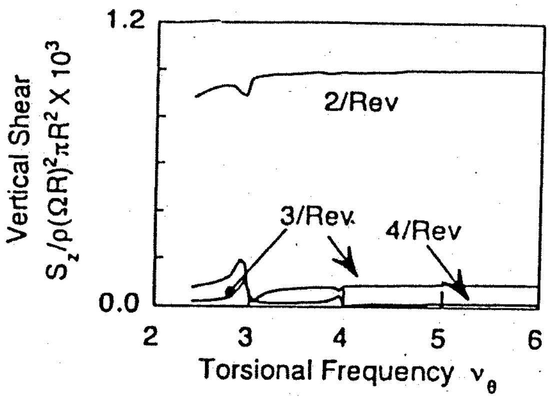
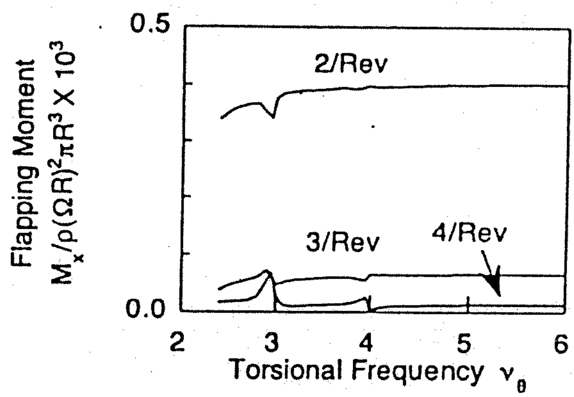


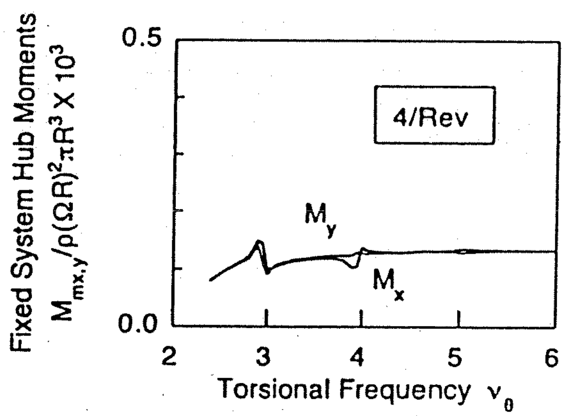
Figure 1: Rigid blade pitch-flap physical model.



(a) Blade root vertical shears (rotating system).



(b) Blade root bending moment (rotating system).



(c) Fixed system pitch and roll moments.

Figure 2: Rigid blade root and hub vibratory loads vs. blade torsional frequency. $\mu = .4$; $C_T/\sigma = .1$.

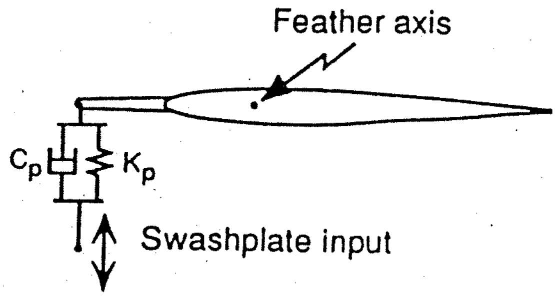


Figure 3: Spring-damper pushrod

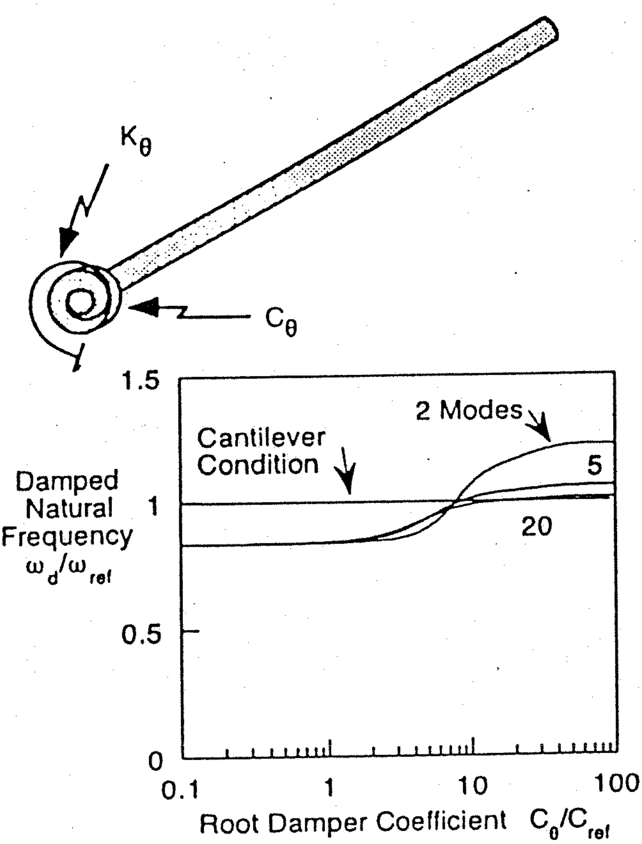


Figure 4: First torsional natural frequency of uniform rod with spring/damper boundary condition. $K_\theta GJ/l = 5$.

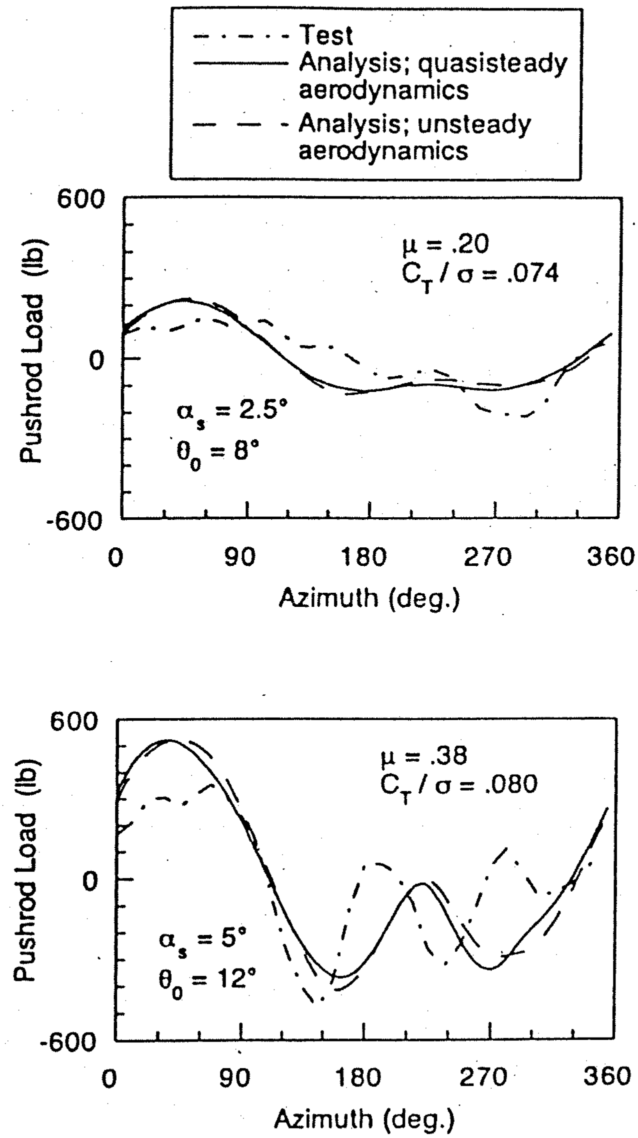


Figure 5: Measured and predicted pitch link loads.

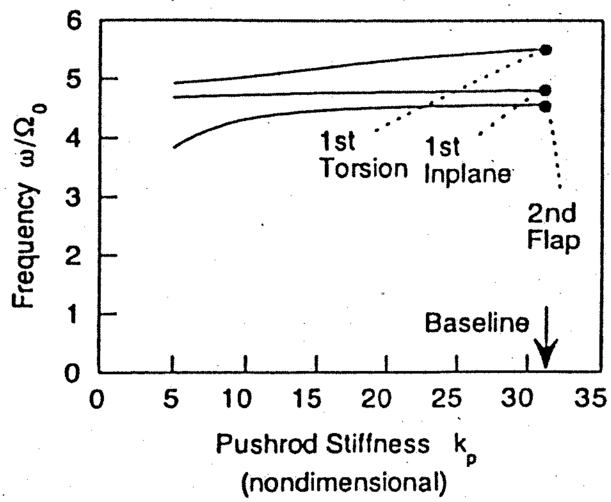


Figure 6: Effect of pushrod stiffness on blade *in-vacuo* rotating natural frequencies. Zero pushrod damping.

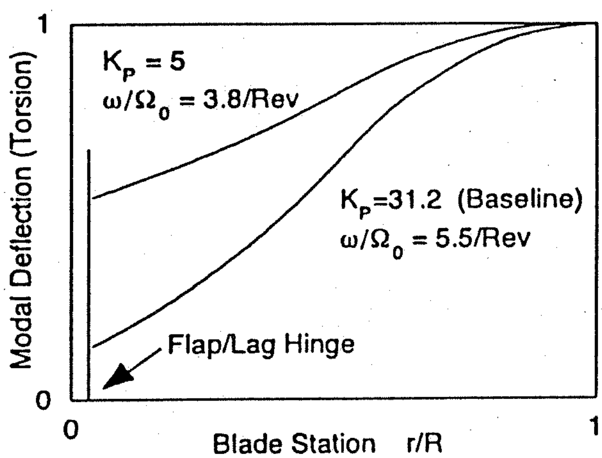


Figure 7: Effect of pushrod stiffness on blade *in-vacuo* rotating torsional mode shape. Zero pushrod damping.

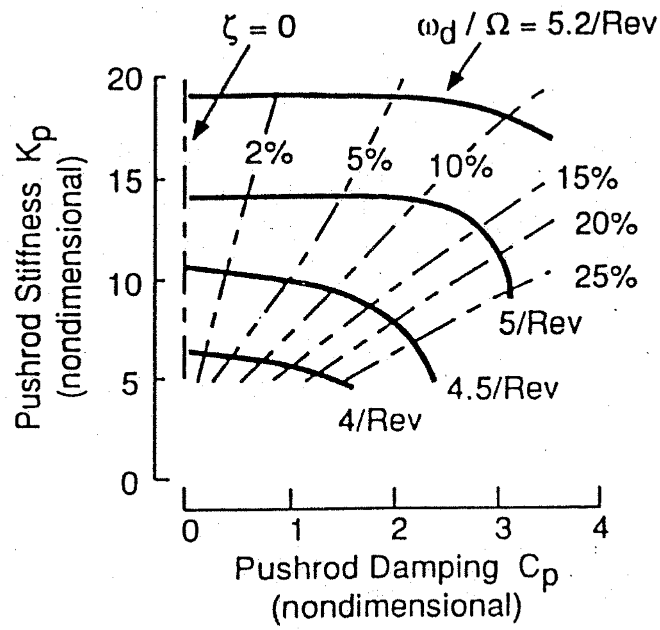


Figure 8: Effect of pushrod stiffness and damping on *in-vacuo* frequency and damping of blade first torsional mode.

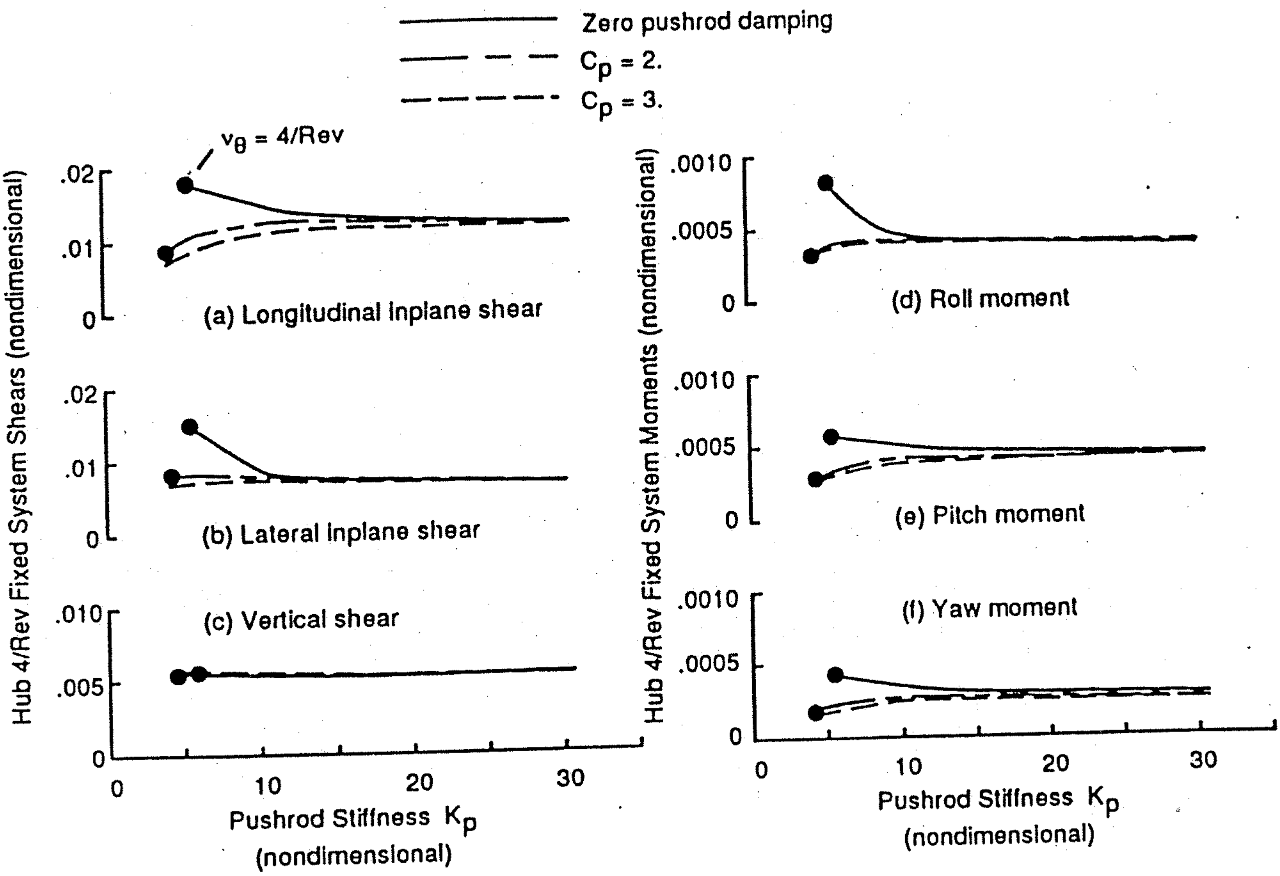


Figure 9: Fixed system 4/rev hub loads for $\mu = .38$, $C_T/\sigma = .080$, and $\alpha_s = 5°$

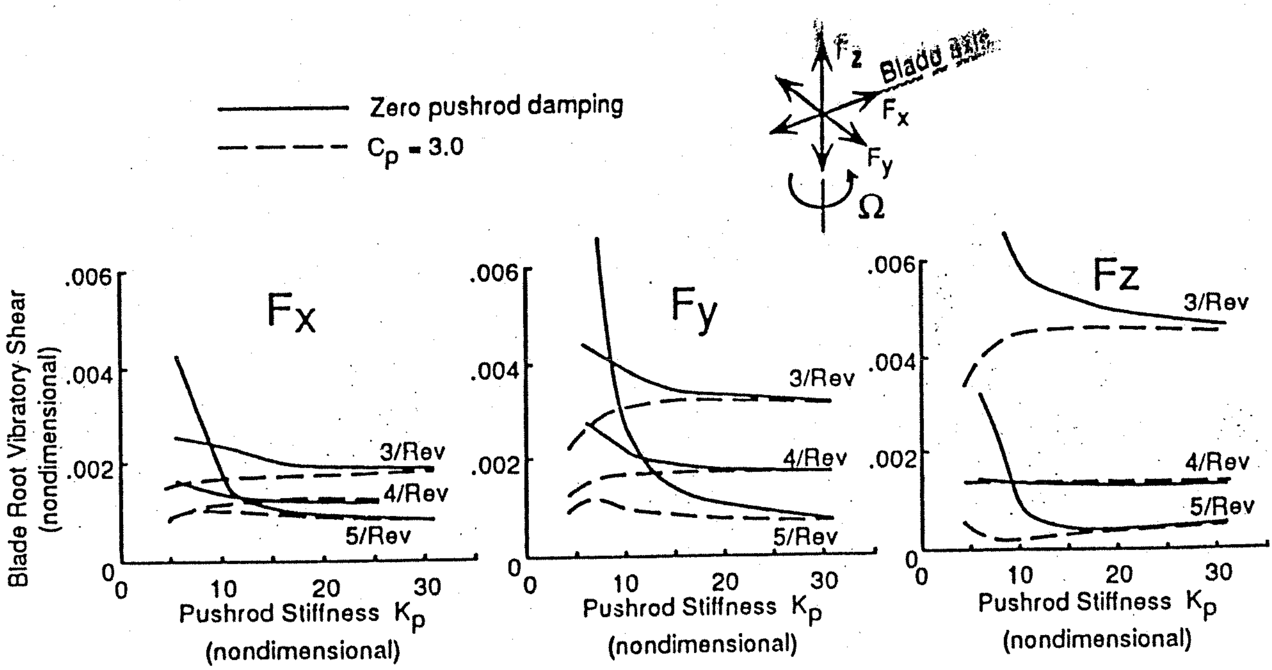


Figure 10: Rotating system 3, 4, and 5/rev blade root loads for $\mu = .38$, $C_T/\sigma = .080$, and $\alpha_s = 5°$

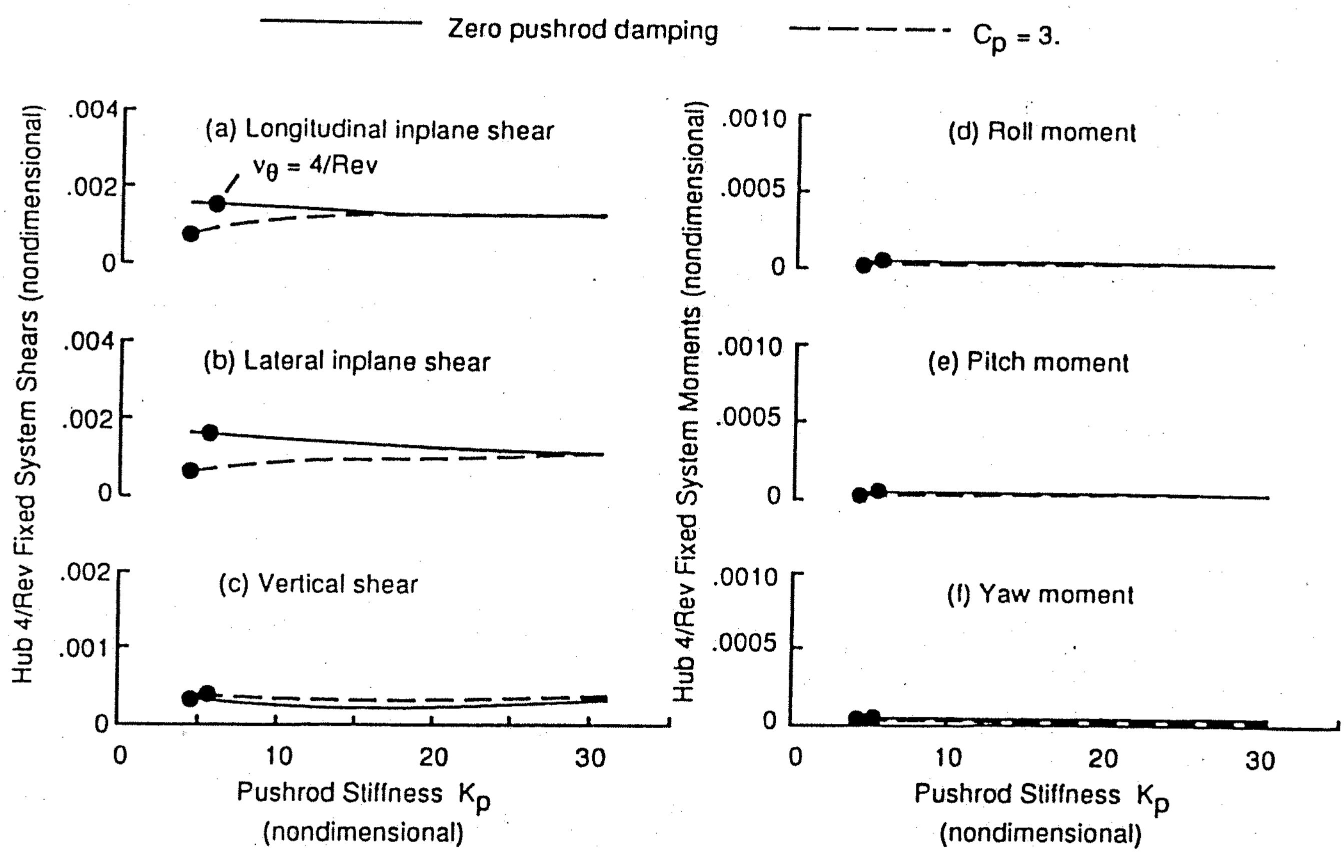


Figure 11: Fixed system 4/rev hub loads for $\mu = .20$, $C_T/\sigma = .074$, and $\alpha_s = 2.5°$

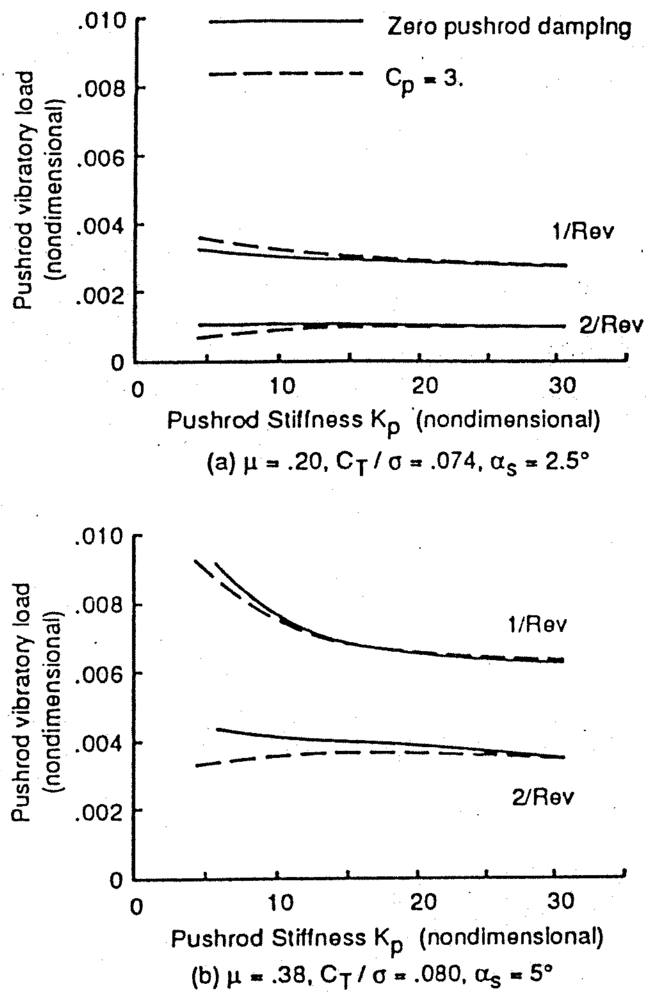


Figure 12: Pushrod loads.

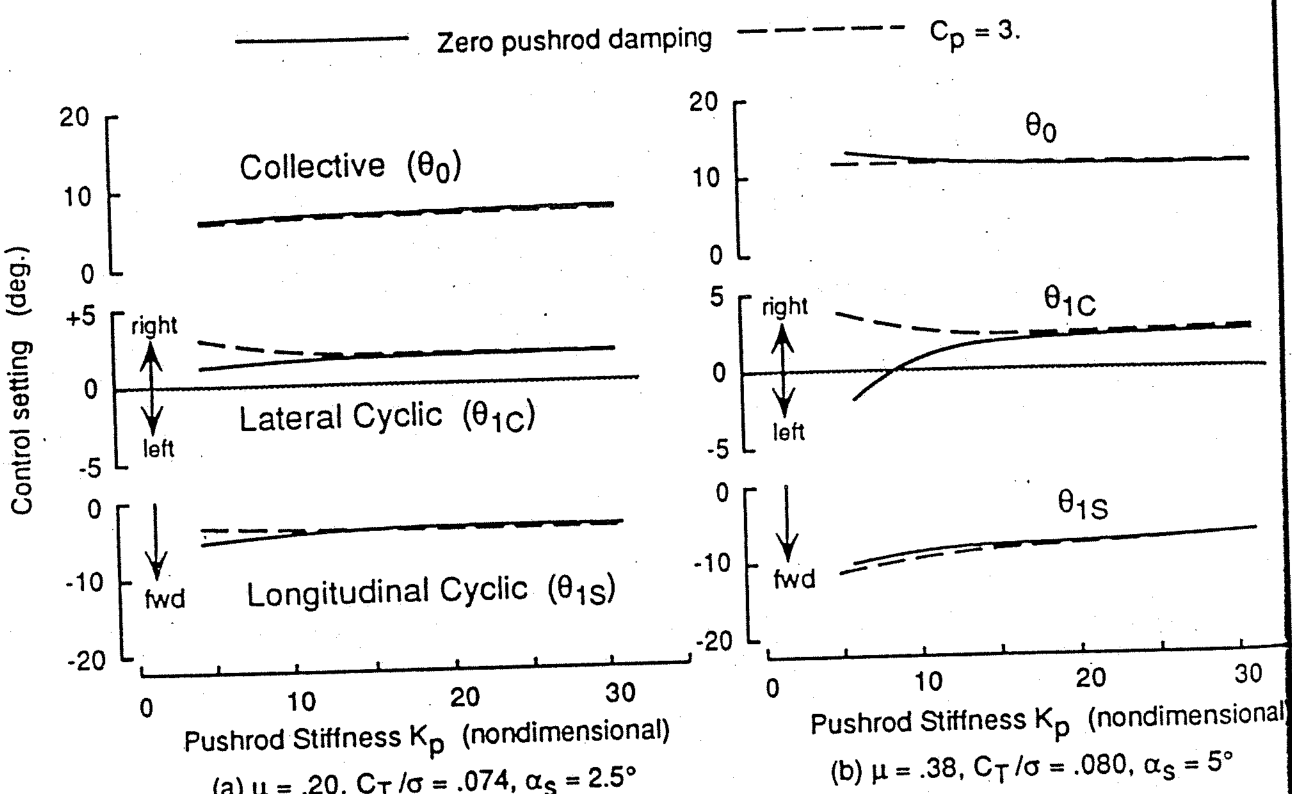


Figure 13: Trim controls.