SOME COMMENTS ON TAIL ROTOR GROUND RESONANCE PROBLEM

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ABSTRACT

The ground resonance problem for either a rotor or a helicopter with rigid blades articulated by a hub has been considered in many papers [1 - 4] and developed rather thoroughly. However some specific features being particular for the tail rotor blades as well arise a number of difficulties for the application of the results obtained. These specific features can be observed in blades, oscillation frequency of which is determined primarily by their own elasticity and elasticity of elastomeric element. They involve firstly the lack of precise representation of a form and oscillation frequency for an elastic blade by some rigid articulated model being elastic in this particular articulation. Other characteristics namely the dependence of blade oscillation form on angular speed of rotation as well as the dependance of damper elasticity and of its damping properties on oscillation frequency are also of major importance.

Taking into account the significance of the facts stated above the paper gives the solution of an oscillation stability problem using a model with elastic blades. This solution enables to account all the specific properties stated above.

AN ANALYSIS OF THE ELASTIC BLADE OSCILLATION WITH A DAMPER

An analysis of elastic blade oscillations with a damper can be carried out on the basis of a dynamic stiffness method applying the traditional solution for blade oscillations without a damper. A damper can be figured here as in general case having a dynamical response to an angular displacement involving elastic and damping components:

$$M = -(\bar{C} + ip\bar{\bar{C}})\vartheta \qquad (1)$$

The equations of the displacement method as an analog of dynamic stiffness method must be satisfied during the oscillations of the blade model shown on fig.1.

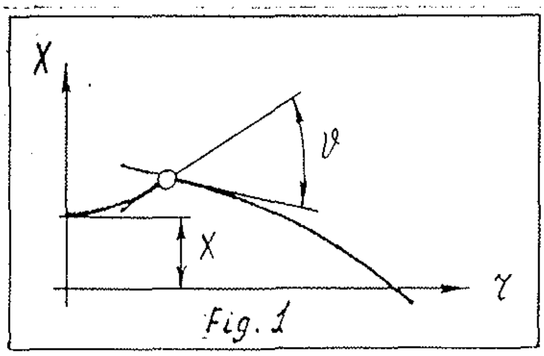

Fig. 1

$$\begin{aligned} X &= D_{xx}X + D_{x\vartheta}\vartheta \\ M &= D_{\vartheta x}X + D_{\vartheta\vartheta}\vartheta \end{aligned} \qquad (2)$$

Where X and M are lateral force and bending moment in certain blade sections. The coefficients in these equations are the values of corresponding dynamic stiffness defined in the analysis of an isolated blade without a damper.

The substitution of (1) into (2) enables to get the expression for a dynamic response of a blade with a damper in particular section. This expression has the following form:

$$X = (\bar{D}_{xx} + i\bar{\bar{D}}_{xx})X \qquad (3)$$

Where

$$\bar{D}_x = D_{xx} - Q(D_{\vartheta\vartheta} + \bar{C})$$

$$\bar{\bar{D}}_{xx} = Q\bar{\bar{C}}p$$

$$Q = \frac{D_{x\vartheta}^2}{(D_{\vartheta\vartheta} + \bar{C})^2 + \bar{\bar{C}}^2 p^2}$$

In the solution given the values of strains and loads are presented in complex form.

Such an approach allows to consider a large variety of dampers used in linear representation

1. $C_d = \bar{C}$; $\bar{\bar{C}} = 0$ – elastic element;
2. $C_d = i\bar{\bar{C}}_p$; $\bar{C} = 0$ – hydraulic damper;
3. $C_d = \bar{C} + i\bar{\bar{C}}_p$ – spring-hydraulic or elastomeric damper

Here $\bar{C} = \bar{C}(p)$ and $\bar{\bar{C}} = \bar{\bar{C}}(p)$.

The analysis of the oscillations with damper having friction characteristics can be carried out similarly as it was in the case of dampers with linear relation of resistance against velocity if the essential property of linearizing will be used:

$$\bar{\bar{C}} = K_{eq} = \frac{2}{\pi}\frac{M_0}{\omega\xi} \qquad (4)$$

To define the values of Dxx and Dxv as functions of oscillation frequency the motion where =0 is considered. In compliance with the additional mass method the mentioned functions are deduced when a blade is attached by rigid mount. To calculate the value of Dvv the blade motion with X=0 is considered. For this purpose a large enough mass about 10 kilos is placed in a certain point of a blade whereas an elastic hinge is simulated by a beam model being one of the blade sections comprised by finite elements.

The desired accuracy is obtained when the length of section selected is sufficiently small.

As an example let's consider the dynamic properties of Mi-8 helicopter main rotor blade. Fig. 2 – 3 shows two sets of curves representing real and imaginary part of dynamic stiffness in a particular range of frequencies. The damping coefficient of linear hydraulic damper is a parameter. As its value increases the gradual transition of blade dynamic properties from articulate to rigid attachment conditions is occurring

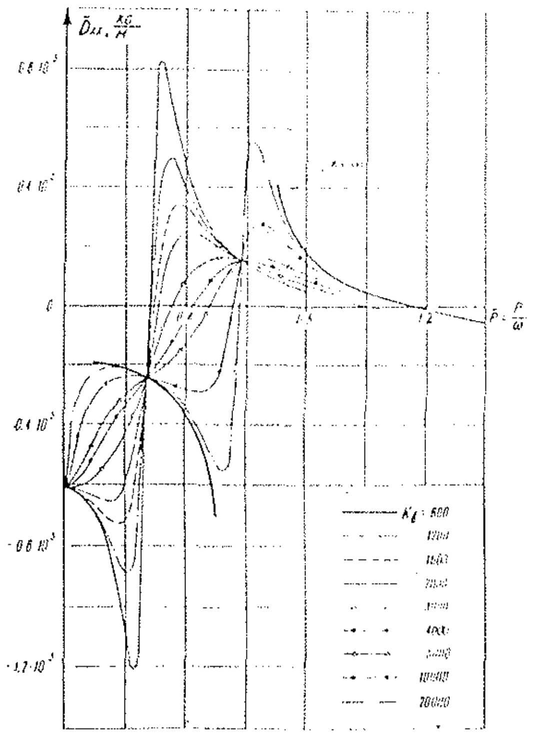

Fig.2

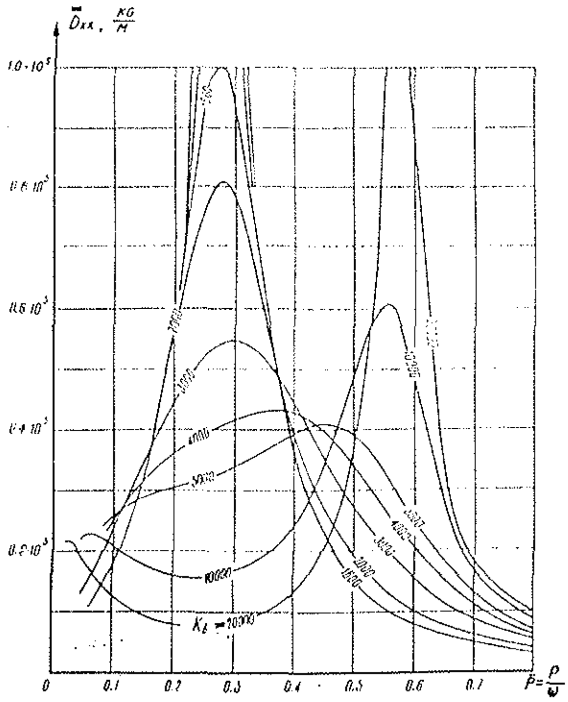

Fig.3

The other example (fig.4) shows the dynamic coefficient changes with the increase of damping coefficient. Dynamic coefficient is defined as

$$\lambda = \frac{1}{\sqrt{\bar{D}^2 + \bar{\bar{D}}^2}}$$

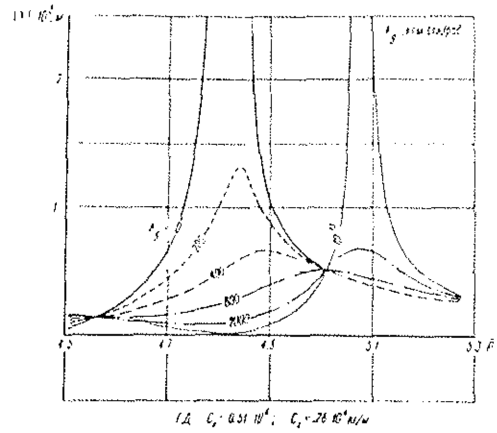

Fig. 4

for the same blade but in the other frequency range. This relationship is an analog to resonance peak which serves to identify natural oscillation frequency in practical operation.

The results presented allow us to make some remarks useful in practice. The application of hydraulic dampers can considerably shift the resonance peak position towards natural frequency of rigidly attached blade oscillation. The increase of damping coefficient can result in an increase of dynamic coefficient and the amplitude of excited oscillations.

Similar effect can be found in other devices, for example, in helicopter landing gear [5]. The increase in shack-absorber damping above some fixed value, when the maximum available damping is reached, causes its subsequent drop.

INSTABILITY OF OSCILLATIONS - GROUND RESONANCE

The suggested method gives the opportunity to define the ground resonance instability boundaries for a rotor with elastic blades.The method enables to consider linear dampers possessing any combination of elastic and damping properties.

Rotor instability is examined in case of support displacement in the rotation plane only as it is the most typical design case. The steady-state harmonic oscillations correspond to the system instability boundaries. For damped systems vanishing of the real part of any eigenvalue or equally vanishing of the value of the determinant of system dynamic stiffness coefficients matrix is the criteria of stability :

$$|C+D|=0 \qquad (5)$$

where C is stator dynamic system coefficient matrix and D is rotor dynamic system coefficient matrix [6].

To simplify the computation this condition is presented in the following form:

$$Re\,|C+D|=0$$
$$Im\,|C+D|=0 \qquad (6)$$

Making calculations within a particular frequency range "p" with Δ p subinterval we define the p(R) and p(I) roots of imaginary and real parts of determinant. The equality of two values

$$p(R)=p(I)$$

defines the boundary of instability.

The required computation accuracy and Δ p subinterval value in determining p(R) and p(I) roots as well as instability boundaries were obtained while comparing of the results obtained by the suggested method with the results of characteristical polynomial solution for traditional ground resonance equations [1].

The calculation of dynamic stiffness matrix coefficients is performed by two methods. The former assumes the calculation to be similar to that for an elastic blade, but with blade stiffness value "EI" being several orders of magnitude greater. The latter assumes coefficients to be calculated from the following analytical functions:

$$D_{xx} = -(p^2 + \omega^2)\, M$$

$$D_{x\vartheta} = D_{\vartheta x} = -(p^2 + \omega^2)\, S_e$$

$$D_{\vartheta\vartheta} = -p^2 I_e + \ell S_e \omega^2$$

where

- M – is the blade mass;
- S_e – is the static moment of blade mass with respect to lag hinge;
- I_e – is mass moment of inertia with respect to lag hinge axis and
- ℓ – is the distance between rotor axis and lag hinge axis.

Further some examples of the calculations performed by the suggested method for Mi-8 helicopter hydraulically damped main rotor blades and for two variants of elastic support are given:

1. One degree of freedom of a support

$$C_{x1} = -m_0 p^2 + C_{x0} + i K_0 p$$

$$C_{z1} \to \infty$$

2. Two degrees of freedom of a support (isotropic case)

$$C_{z2} = C_{x2} = C_{x1}$$

Fig. 5 shows the results of calculations for rigid blades where figures indicate the values of instability boundaries resulting from the solution of characteristical equation.

Fig. 6 shows the solution for elastic blades possessing the same mass characteristics.

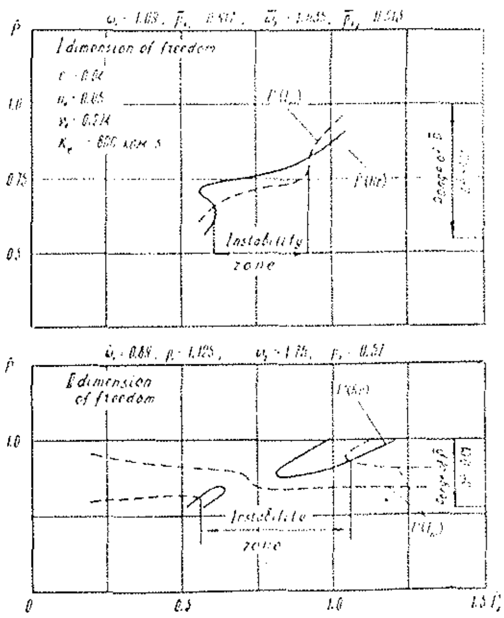

Fig. 5

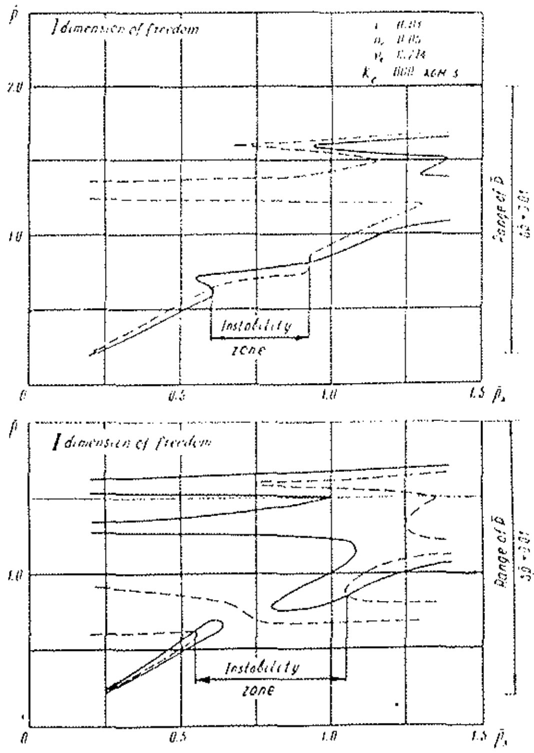

Fig. 6

The data resulting from investigations of ground resonance stability boundaries depending on the properties and values of damping coefficient for two types of dampers - hydraulic H and elastic-hydraulic EH (the last is a tandem of elastic and damping elements) are shown on fig. 7-8.

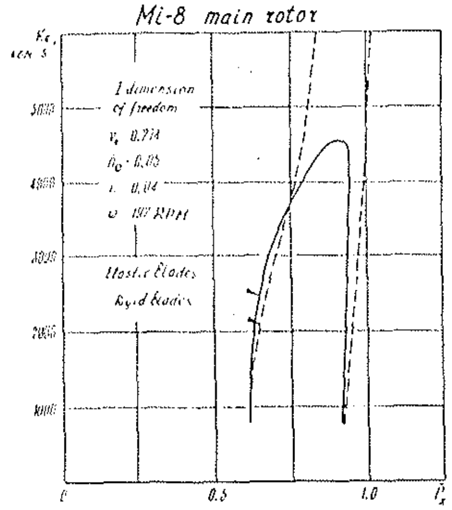

Fig. 7

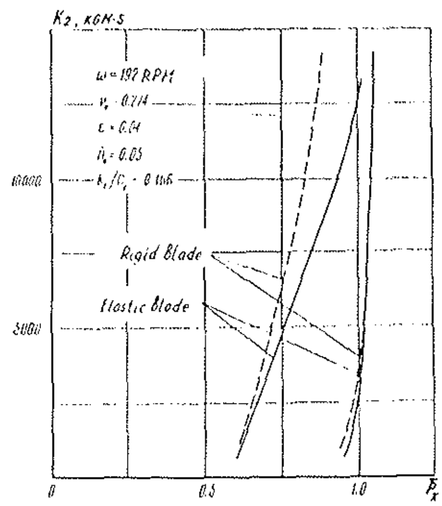

Fig. 8

For comparison the instability boundaries obtained for a rigid blade are shown by a dotted line.

An increase of damping in 2 - 3 times, with respect to damping applied in series production, creates considerable differences in instability boundaries position for rigid and elastic blades. Moreover, this characteristic is quite different for two damper types. For instance, the above- mentioned effect of blade nip can be seen in case of hydraulic damper when its damping increases. The result is the appearance of a certain equivalent value of non-rotating blade oscillation frequency. As it is known from the ground resonance theory this effect reduces the value of damping required.

The interaction of elastic blade and E-H damper is characterised by a large variety of features as both elastic and damping portion of its reaction depend on the oscillation frequency.

Fig.9 shows equivalent damping variations of such a damper as a function of blade oscillation frequency. One can see three families of curves each satisfying one of the criterions: C - const, K - const and K/C - const.

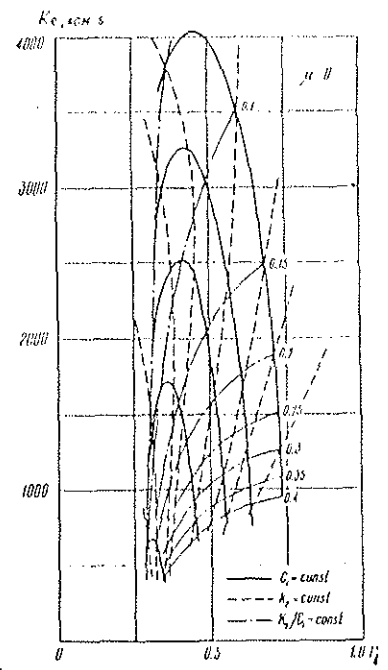

Fig. 9

TAIL ROTOR GROUND RESONANCE

A single rotor helicopter fuselage as an elastic support of a tail rotor possess several natural oscillation frequencies which are within the rotor angular velocity range and therefore are of interest when solving the stability problem. The experience of developing and testing of the experimental rotors with lag hinges for Mi-8 and Mi-6 helicopters has shown that it is required to provide stability for two potential fuselage oscillation forms. It is essential to have in mind that damping features of support oscillations are very limited and reduced mass of support has relatively smaller values that augments rotor energy importance in interaction with support.

Our experience shows that an EH-damper in lag hinge should have high enough stiffness to provide required value of blade oscillation frequency ( ) and sufficient move away from an instability boundary.

The results of stability problem analysis for alternative version of the tail rotor for Mi-38 helicopter project are presented here. Basing on experimental and design work and bearing in mind the succession of structure the fuselage properties are considered to be the same as for Mi-8 helicopter.

Fig.10 shows the position of two instability zones depending on rotor angular velocity and lag hinge elastic element stiffness.

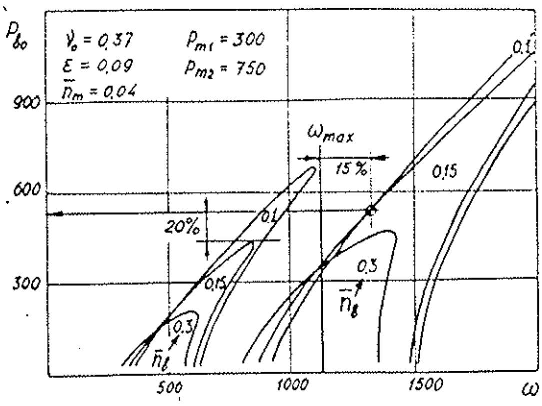

Fig. 10

The safety condition for the first oscillation tone is the excess of available damping over the required one as shown on fig.10. "C" and "n" parameters which satisfy this condition are shown on fig.11. For the second tone it is necessary to provide the margin of instability zone over the maximum possible operational value of rotor angular velocity (taking into account the possible rotor RPM increase in manoeuvres. This margin is to be about 15%. In view of such a margin it is possible to define the required combination of blade oscillation frequency and fuselage second tone frequency satisfying these conditions. Fig.12 shows this relationship and corresponding value of elastic element stiffness – C. Taking into account the limitation of blade oscillation frequency ($p < 0.7$) to avoid resonance with first harmonic, the fuselage second tone frequency should not be lower than 700 cycles/min. and C = 500 Nm/deg. This requirement creates certain difficulties because the frequency of this oscillation tone for Mi-8 helicopter is about 650...680 cycles/min.

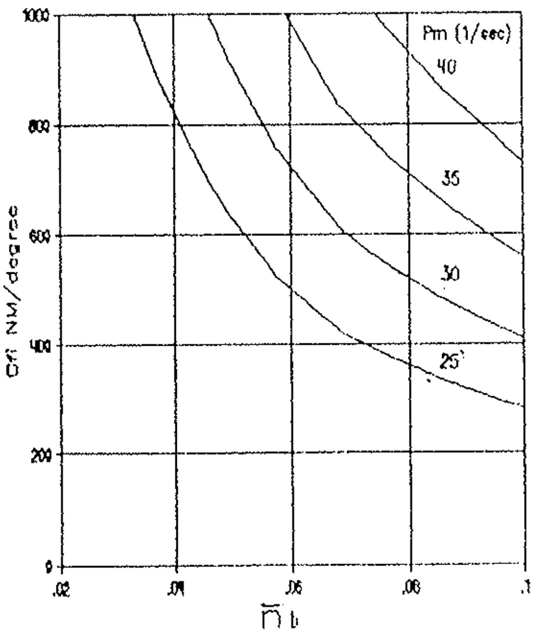

Fig. 11

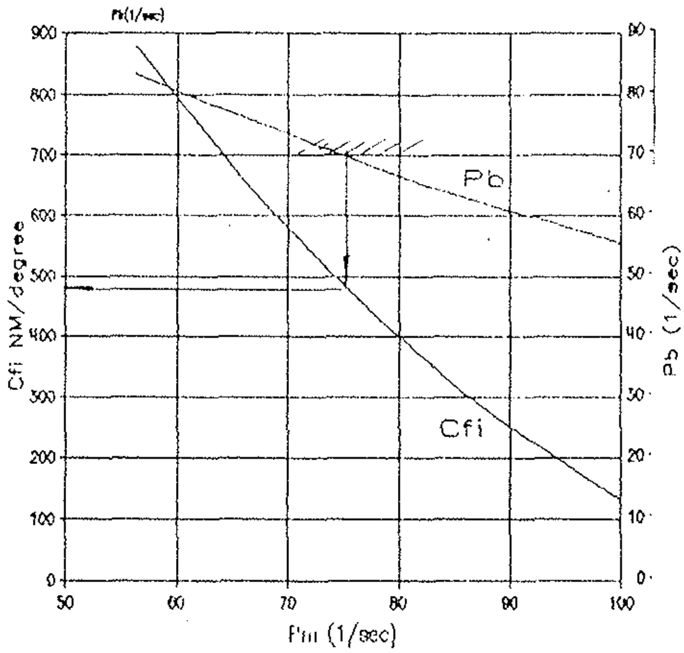

Fig. 12

As calculations and experiments show this form of fuselage oscillations contains strains caused by bending in two planes and by twisting. Oscillation coupling occurs due to tail rotor positioning out of the symmetry plane. As a result the tail boom design requirements ensuring specified frequency and mass parameters may be very complicated.

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