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A MODEL FOR WINDTUNNEL ROTORCRAFT RESEARCH
- GROUND RESONANCE INVESTIGATIONS -

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by

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List of Symbols

- a - distance between lagging hinge and rotor axis [m]
- b - distance between c.g. of blade and lagging hinge [m]
- c_E - spring stiffness of the equivalent blade [kg s⁻²]
- $$\hat{c}_E^* = \frac{2}{z} c_E^* = \frac{2}{z} (c_E + a M_E \omega^2)$$
- c_x, c_y - spring stiffness of the equivalent rotor support system [kg s⁻²]
- I_E - moment of inertia [kg m²]
- $$\hat{I}_E = \frac{2}{z} I_E$$
- k_E - damping value of the blade [kg s⁻¹]
- $$\hat{k}_E = \frac{2}{z} k_E$$
- k_x, k_y - damping value of the equivalent rotor support system [kg s⁻¹]
- m_x, m_y - generalized mass (rotor hub included) [kg]
- M_E - static mass moment [kg m]

n	-	index, n-bladed rotor (n = 1, 2...z)
x	-	longitudinal axis (horizontal)
y	-	lateral axis (horizontal)
z	-	number of blades [-]
ζ	-	lagging angle [rad]
ψ	-	azimuth angle [rad]
σ, τ	-	$= \sum_{n=1}^{n=z} \zeta_n \sin \psi_n ; = \sum_{n=1}^{n=z} \zeta_n \cos \psi_n$
ω_{RO}	-	circular frequency of the rotor [rad s ⁻¹]
ω_{ζ}	-	lagging frequency [rad s ⁻¹]
v	-	eigenvalue [s ⁻¹]

1. Introduction

At the Institute of Flightmechanic of the DFVLR a model for rotorcraft research has been built which will be available for scientific research from Sept. 1976 (Fig. 1).

In order to avoid any dangerous situations due to ground resonance of the test stand this problem had to be considered in particular before the first test runs were conducted in March '76.

As it is quite difficult to describe the rotor/rotor support system mathematically an equivalent system was defined that has the same vibration characteristic at a defined region of frequency as the test stand itself [1].

In this case this equivalent system can be separated into two systems:

- a spring, mass, damping system with a different oscillation characteristic in longitudinal - and lateral-direction to simulate the rotor support.
- an equivalent rotor with an equivalent lagging hinge and equivalent spring to simulate the hingeless rotor.

The definition of the equivalent system yields the parameters that have to be determined to conduct the ground resonance calculation (Fig. 2).

Summarizing this figure shows the block diagram of the work that has to be done for ground resonance calculation.

The equivalent values of the rotor support were determined experimentally by ground vibration tests at DFVLR in Göttingen [2].

2. Ground Vibration Technique

Ground vibration technique is used when mathematical methods are too sophisticated to determine the vibration characteristic of a system.

Therefore the main objective is to determine the natural frequencies, the eigen-forms, the generalized masses and the damping values of the physically unknown system under operation.

2.1 Application to the Rotor Support

The identification of the above mentioned values were determined in the following way:

- Approximated determination of the natural frequencies of the rotor support by sweep tests.

Thereby the structure was excited separately in x-and y-direction with slowly and continuously changing frequencies at the rotor hub (Fig. 3).

The signals of the accelerometers installed on fuselage and rotor support are fed to a plotter where high amplitudes show the resonance frequencies in the first approximation.

During these sweep tests the exciting frequency was changed up to 80 Hz (Fig. 4).

For ground resonance calculation it is sufficient to evaluate frequencies not higher than the rotor frequency minus lagging frequency ($\omega_{Ro} - \omega_{\zeta}$) which means to analyse the lowest natural frequency of the system.

2.1.1 Identification of a Resonance-Peak

After the conduction of the sweep-tests an isolation of a natural frequency was made in the vicinity of the resonance peak. Frequency and forces of the exciter configuration were adapted to the phase-resonance-principle.

2.1.2. Determination of the Eigenform and the Equivalent Values of the Vibration System

The eigenform of the testing stand was measured by results from 40 accelerometers fixed on the stand. The signals of this devices were split into real parts and imaginary parts with reference to the phase of the exciting forces. A process-computer calculated the eigenform for each accelerometer (Fig. 5).

The generalized mass, i.e. the mass which has the same value as that one of a spring-mass-damping system, is determined by frequency shifting caused by fixing additionally little masses on the test stand. This yields the spring stiffness, too.

Damping values are determined by switching off the excitation of the shakers and by recording the decay of the oscillation of the structure by a speed indicator.

Applying the described methods to the test stand yields the characteristic values for the equivalent rotor support.

2.2. Configurations

As the test stand is not axially symmetrical but nearly symmetrical with respect to the x-and y-axis the above mentioned method had to be applied for these two directions.

As there were no informations about trouble with ground resonance before the ground vibration tests six test-configurations were tested yielding a lot of data about the behavior of the rotor support.

The results from the ground vibration tests gave in particular information about the vibration characteristic of the rotor support. But ground resonance calculation also needs the elastomechanical characteristic of the rotor, because ground resonance is a self induced coupled oscillation between rotor and air frame, i.e. rotor support system.

3. Vibration Characteristic of the Rotor

Ground resonance can occur, when after any disturbance the rotor blades oscillate in the lagging direction in such a way that the c.g. of the rotor leaves the axis of rotation. This unbalance leads to a rotation with a frequency equal to the difference between rotor frequency and natural lagging frequency of the blades.

If the eigenfrequency of the rotor support system is the same as this frequency of the unbalanced rotor a self induced dangerous oscillation happens.

To determine the frequency of the unbalanced rotor the vibration characteristic of the rotor must be known.

3.1 Determining the Lagging Frequency of the Rotor

The method of matrix transfer [3] was used to calculate the natural lagging frequency of the rotor.

If dimension, mass-and stiffness distribution of the rotor blade are known, the natural frequencies and also higher harmonics can be calculated for the whole range of rotor frequencies (Fig. 6).

The figure shows the natural lagging frequencies ω_{ζ} due to ω_{RO} for the hingeless rotor. These frequencies are important for ground resonance calculation. This figure can be used as a "quick look" for ground resonance considerations although no coupling of the rotor - rotor support system and no damping of the rotor was taken into account.

The points of intersection of $\omega_{RO} - \omega_{\zeta}$ with the frequencies of the test stand are points of resonance of the rotor- and the rotor support system. The graph doesn't show whether a disturbance yields a damped or undamped oscillation.

3.2 Calculation of an Equivalent Hinge and Spring for a Hingeless Rotor Blade

Once the lagging frequency was known the state variables of the equation of motion had to be calculated. For simplification an equivalent rotor system was chosen (Fig. 7). It had to be defined in such a way, that in operating condition the first lagging harmonic of the hingeless rotor is equal to the first harmonic of the equivalent rotor: or, to define it mathematically, - the Southwell coefficient of the equivalent system must be equal to the basic vibration of the original system [4].

4. Ground Resonance Calculation

As explained in the introduction the rotor and rotor support are described by an equivalent system. The equivalent oscillating system of the longitudinal (x) and the lateral (y) motion of the rotor support forms together with the equivalent rotor a two-dimensional asymmetrical ground resonance system (Fig. 8).

In opposition to the rotor support, which can oscillate only in longitudinal and lateral direction, the rotor vibrates in the horizontal plane.

4.1 The Uncoupled System

It will be enough to consider the system to be uncoupled for a rough estimate of the ground resonance. In figure 6 you see the uncoupled system. The curve of the rotor frequency minus the resonance frequency of the lagging motion and the resonance frequencies of the rotor support system are plotted for different configurations. Especially "configuration 2" (without fuselage) and "configuration 12" (with fuselage) are important.

The points of intersection of the curve $\omega_{RO} - \omega_r$ with the horizontal frequency of the equivalent rotor support are mostly located within the two regions of instability of the coupled and damped system. Accordingly both regions of instability of the asymmetric two dimensional equivalent system move to the right when the resonance frequency of the rotor support increases.

4.2 The Coupled System

Using the synthetic method the equation of motion for the coupled ground resonance system results from the forces of equilibrium in x and y -direction and from the four equations for the equilibrium of moments at the four lagging hinges of a four bladed rotor (Tabulation 1).

The matrix system of equations includes the eigenvalues ν . The determinant of the system has to be zero for non-trivial solutions. Under this condition the solution of the determinant yields a polynomial of eighth order. The numerical calculation of the eigenvalues was made by a computer, giving the real parts and the imaginary parts of ν with respect to different angular velocities of the rotor.

4.3 Curves of Frequency

The main configurations of the test stand for ground resonance calculation are:

- braced, without fuselage
- braced, with fuselage

For all test configurations without fuselage there are no problems due to ground resonance (Fig. 9).

It turned out that there is a positive real part (that means instability) only at higher speed of rotation.

The curves change when the fuselage is included into the calculation (Fig. 10).

The mass of the fuselage produces a shifting of the regions of instability to a lower speed of rotation. One of these regions lies now at the operation speed so that ground resonance can occur. However, the positive real part is so low that hazardous ground resonance can be avoided in time. According to the low damping characteristic the curves of the real parts can not be regarded as an exact proof of stability because it is nearly impossible to derive an exact mathematical model of the whole test stand.

In the region of low instability it is not advisable to proof the theoretically predicted instability by experiments.

Therefore it was necessary to change the construction of the testing stand slightly. These changes consisted of struts between the stand and the floor.

5. Conclusions

The ground resonance investigations show that for a safe operation it is important to conduct a comprehensive program to know the vibration characteristic of the rotor test stand .

Low instabilities or stabilities can not be determined by simple investigations. Considering the coupled system of rotor and support the results will yield a good basic knowledge of the ground resonance characteristic. It could be shown that a mathematical - experimental method is able to predict any problems due to ground resonance of the test stand with high confidence.

References

- [1] R.P. Coleman, A.M. Feingold: Theory of self-excited mechanical oscillation of helicopter rotors with hinged blades.
NACA Rep. 1357 (1958)
- [2] F. Kießling: ground vibration test. A tool for rotorcraft dynamic and aeroelastic investigations.
Presented on "First European Rotorcraft and Powered Lift Aircraft Forum" 22-24 September 1975, Southampton.

- [3] S. Benz: Berechnungen der ungekoppelten Schwingungen von Hubschrauberblättern mit Hilfe der Übertragungsmatrizenmethode, DLR-Forschungsbericht 70-63, 1970.
- [4] G. Schilling: Berechnung von Ersatzgelenk und Feder für ein gelenkloses Rotorblatt, DFVLR IB Q27-71/10, 1971.

$m_x v^2 + k_x v + c_x$	$-M_E v^2$	o	o	$\begin{bmatrix} x_o \\ \phi_o \\ y_o \\ \tau_o \end{bmatrix} = 0$
$-\frac{z}{2} M_E v^2$	$I_E (v^2 - \omega^2) + k_E v + c_E^*$	o	$-(2I_E \omega v + k_E \omega)$	
o	o	$m_y v^2 + k_y v + c_y$	$M_E v^2$	
o	$+(2I_E \omega v + k_E \omega)$	$\frac{z}{2} M_E v^2$	$I_E (v^2 - \omega^2) + k_E v + c_E^*$	

$c_E^* = (c_E + aM_E \omega^2)$

$m_x v^2 + k_x v + c_E$	$-M_E v^2$	o	o	$\hat{D} = 0$
$-M_E v^2$	$\hat{I}_E (v^2 - \omega^2) + \hat{k}_E + \hat{c}_E^*$	o	$-(2\hat{I}_E \omega v + \hat{k}_E \omega)$	
o	o	$m_y v^2 + k_y v + c_y$	$M_E v^2$	
o	$+(2\hat{I}_E \omega v + \hat{k}_E \omega)$	$M_E v^2$	$\hat{I}_E (v^2 - \omega^2) + \hat{k}_E + \hat{c}_E^*$	

TAB. 1: Equation of Motion for the Coupled Ground Resonance System



Fig. 1: Rotor Test Stand

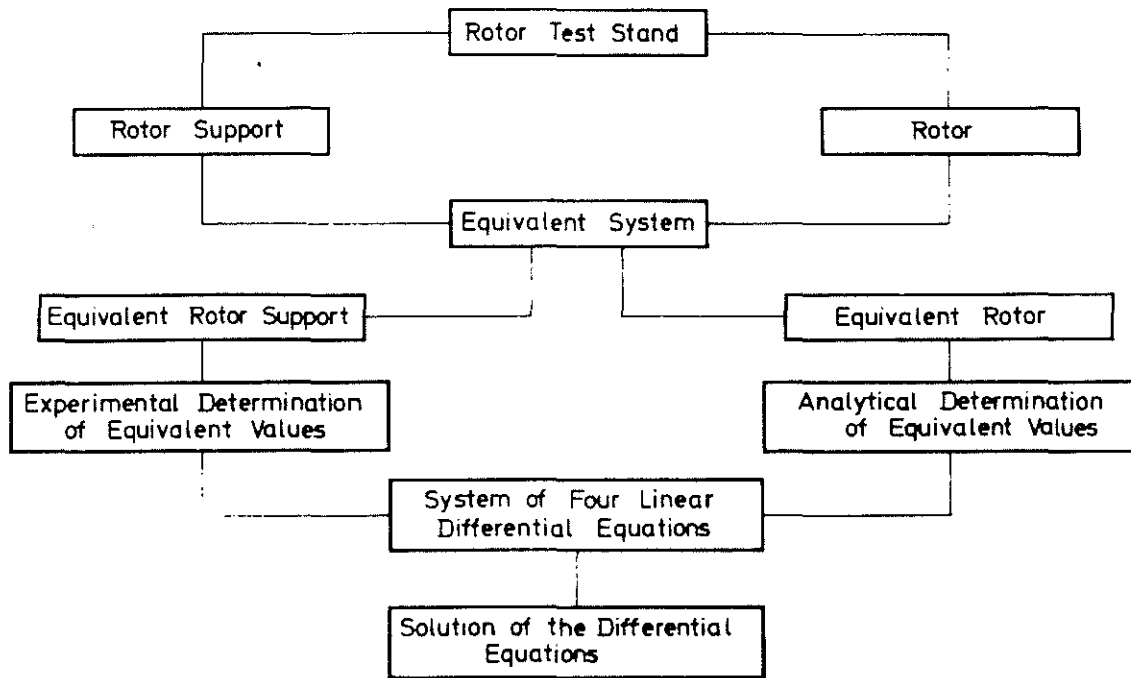


Fig. 2: Ground Resonance Calculation

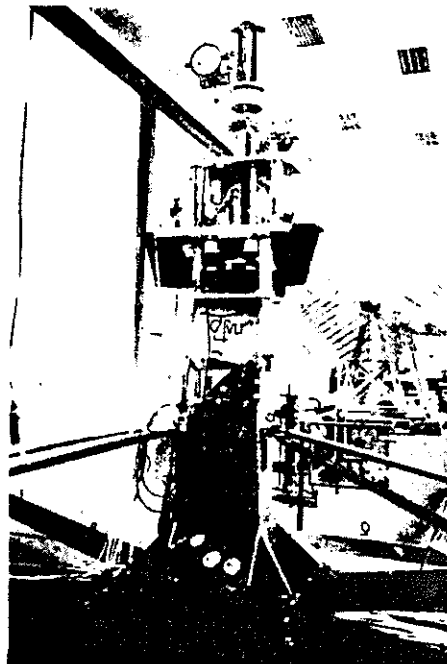


Fig. 3: Ground Resonance Test Configuration

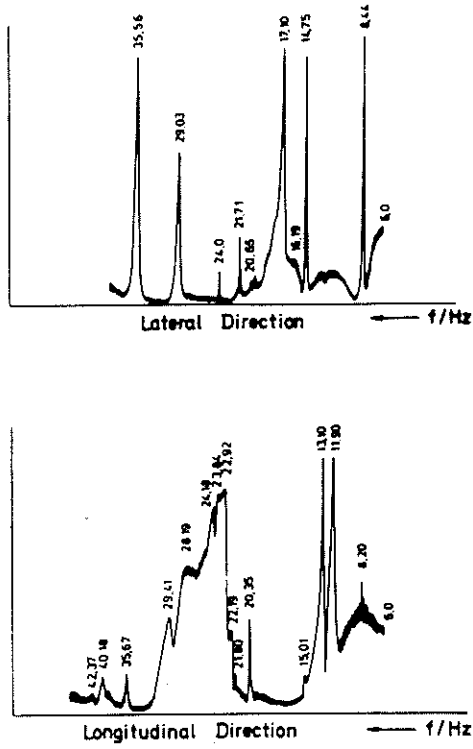


Fig. 4: Sweep Tests

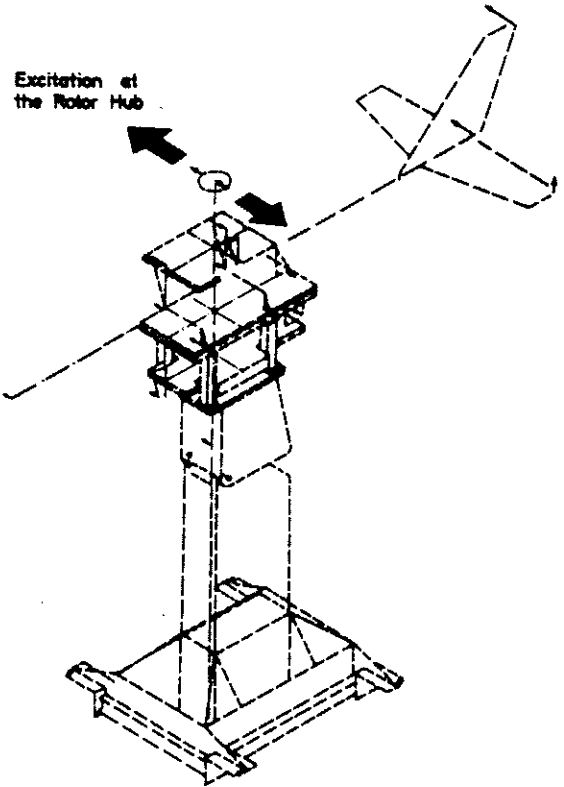


Fig. 5: Eigenform of Configuration "With Fuselage"

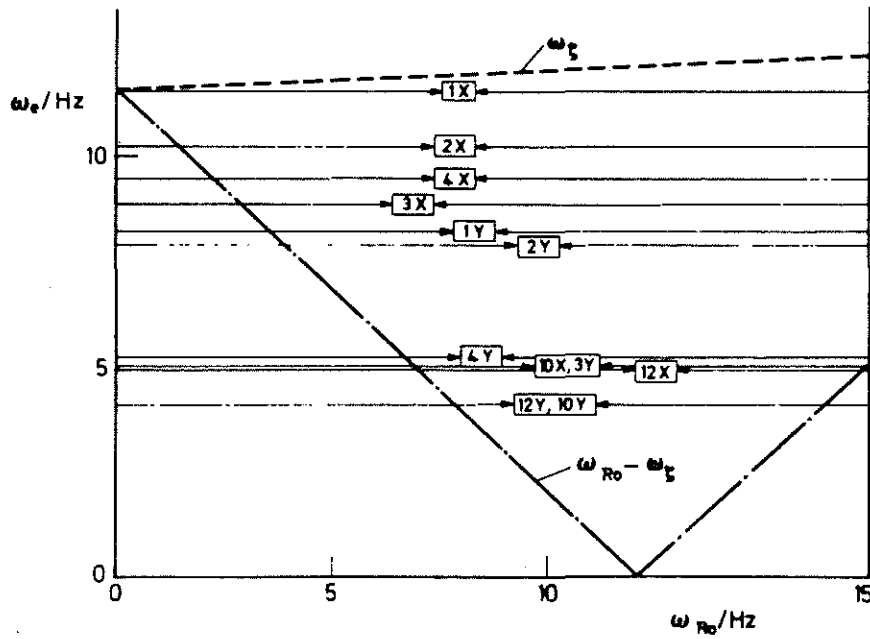


Fig. 6: Natural Frequencies of the Rotor Support and Ground Resonance Diagram; Uncoupled, Undamped

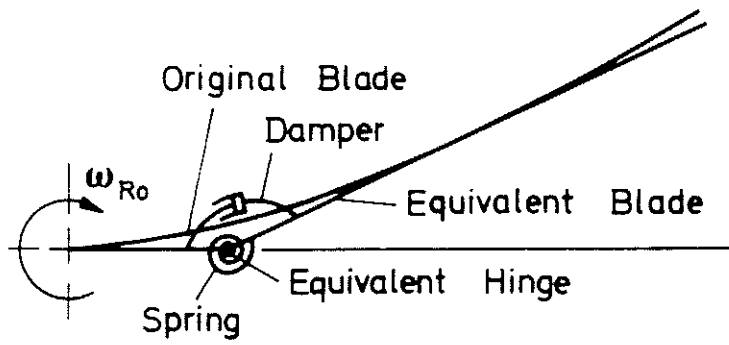


Fig. 7: Hingeless Blade and Equivalent Blade

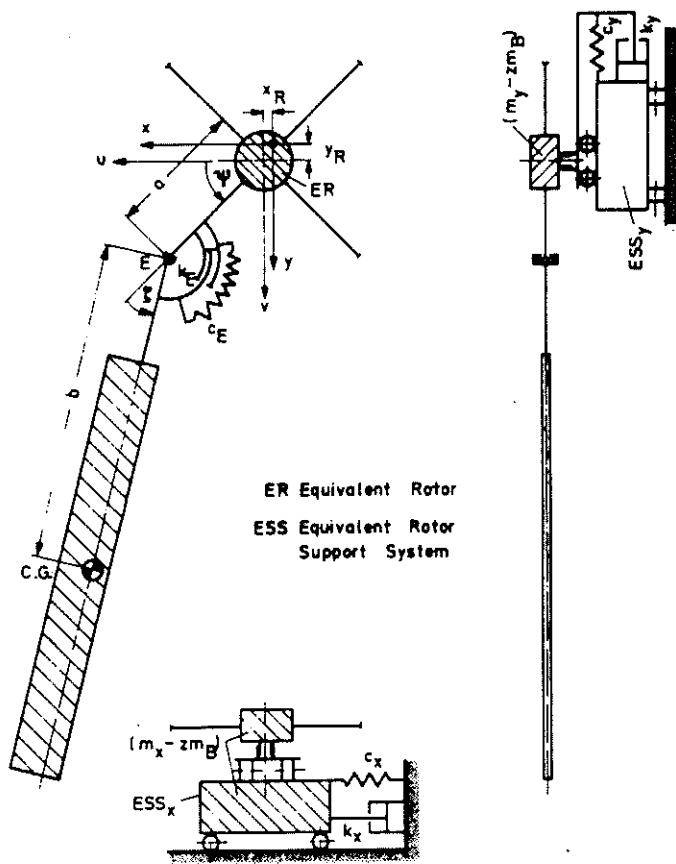


Fig. 8: Two-Dimensional, Asymmetrical Equivalent System

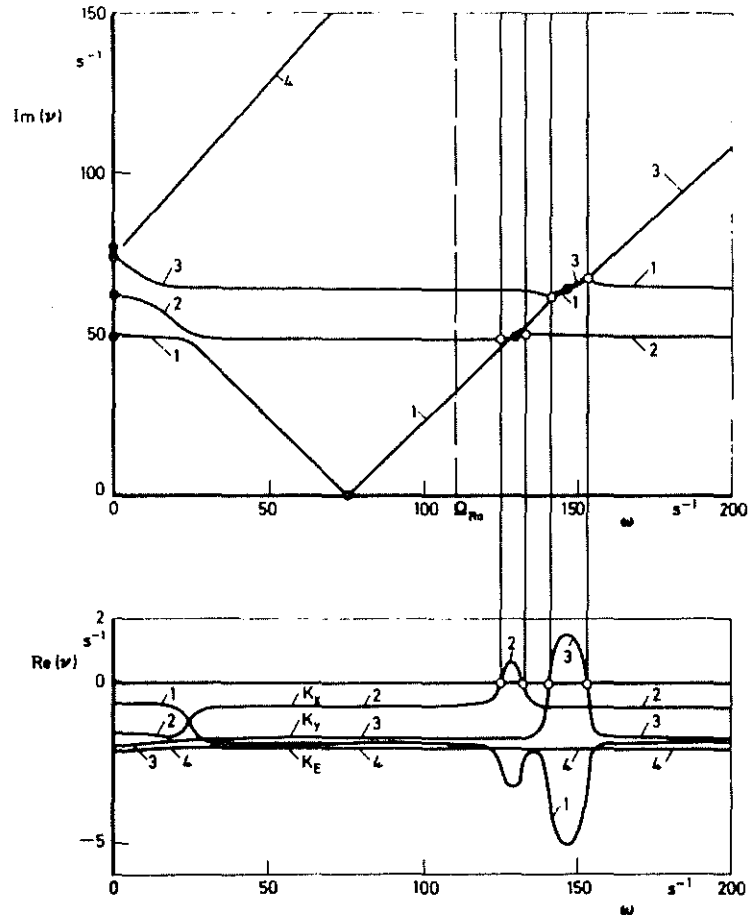


Fig. 9: Imaginary Parts $Im(v)$ and Real Parts $Re(v)$ of the Two-Dimensional Damped System (Configuration "Without Fuselage")

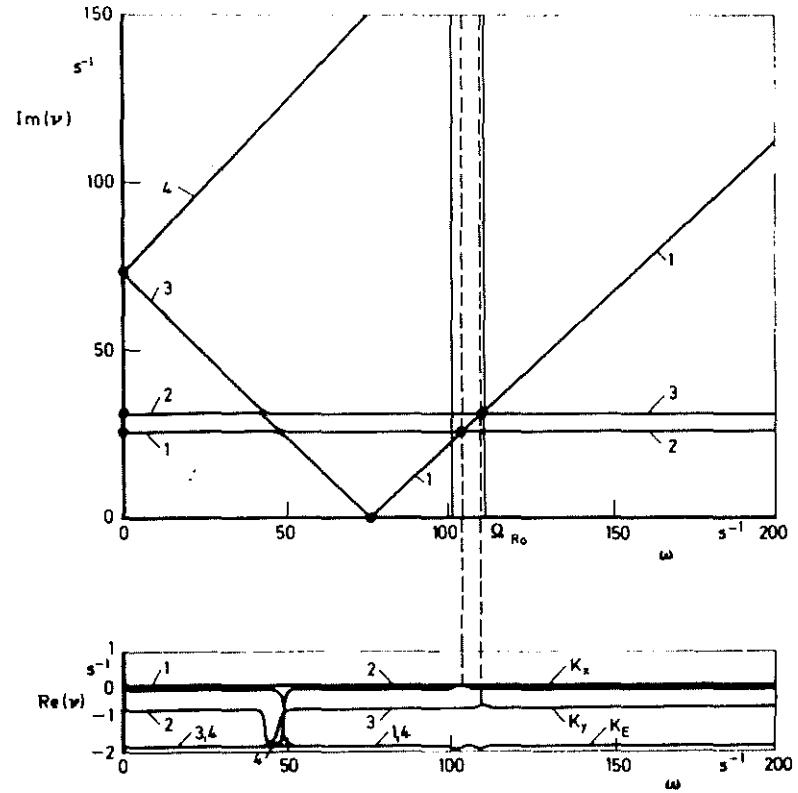


Fig. 10: Imaginary Parts $Im(v)$ and Real Parts $Re(v)$ of the Two-Dimensional Damped System (Configuration "With Fuselage")