Vibration	Simplified Determination of Stability Limit of Rotating Speed on Journal Bearing Rotor	Rotating Machinery
Self-Excitation		

Object Machine

Journal bearing rotor involving stability problem, called oil-wheel/oil-whip (Fig.1)

Observed Phenomena

There is no method to simply determine the Stability limit number of revolutions for instable vibrations, such as oil whirl & whip to occur during high speed rotations

Cause Estimation

Analysis and Data Processing

By citing the reference literature (1) as to oil whirl / whip, examinations were made for three cases of the bearing gap C/R = 0.001, 0.003 and 0.010, with the results of experiment and of calculation given in Fig.2 and Fig.3, respectively. Fig.3 shows the results of complex eigenvalue analysis using 8 parameters of dynamic bearing characteristics, indicating that both the experiment and calculation have the Stability Limit Speed (SLS) of about 90rps. A simplified method is considered for determining this SLS. The equation of motion is given from the reference literature (2) as follows:

$$\begin{bmatrix} m_t & m_c \\ m_c & 1 \end{bmatrix}\begin{bmatrix} \ddot{z}_b \\ \ddot{\eta} \end{bmatrix} + \begin{bmatrix} c_d(s-j\lambda\Omega)+k_d & 0 \\ 0 & \omega_z^2 \end{bmatrix}\begin{bmatrix} z_b \\ \eta \end{bmatrix} + \begin{bmatrix} k_{b+}c_b s & 0 \\ 0 & 0 \end{bmatrix}\begin{bmatrix} \bar{z}_b \\ \bar{\eta} \end{bmatrix} = 0 \qquad \cdots(1)$$

The first term is the mass matrix of a mode synthesis method model, k_d and c_d are oil film forward components, k_b and c_b are oil film backward components, while z_b is journal displacement, η is relative bending deflection of the shaft center of the first mode possibly occurred, and ω_z is the number of natural frequency of shaft bending (simply supported), respectively. As oil film damping is very large ($\zeta_d \gg 1$), the approximate characteristic roots would be $s = j\lambda\Omega$ and $s = j\omega_z$. Thus, the following equation gives the stability of $s = j\lambda\Omega + \Delta$, where Δ is the difference between precise roots and approximate roots.

$$\Delta = -\begin{vmatrix} -(\lambda\Omega)^2+\omega_d^2 & -\mu_c(\lambda\Omega)^2 \\ -\mu_c(\lambda\Omega)^2 & -(\lambda\Omega)^2+\omega_z^2 \end{vmatrix} / 2\zeta_d\omega_d[-(\lambda\Omega)^2+\omega_z^2] \qquad (2)$$

As the determinant is the eigenvalues of an undamped system, if ω_z and ω_d are given as in Fig.4, the primary natural frequency ω_1 becomes a little below than the smaller one. Thus, this simplified determination "Exceeding $\lambda\Omega > \omega_1$, then instable" is obtained.

Countermeasures and Results

Fig.5 illustrates the results of applying this simplified determination to the reference literature [1]. Now, look at the process of SLS calculation for C/R = 0.010. From the number of natural frequency of shaft bending ω_z = 47.3Hz, oil film spring constant k_d = 8MN/m and rotor mass m_t = 51.8kg, the whirl natural frequency ω_d = 62.6Hz. These two frequencies provide the undamped natural frequency ω_1 = 42.2Hz. The intersection point of $\omega_1$③and λΩ=0.493Ω in Fig.5 corresponds to the SLS, that is, 42.2/0.493 = 85.6rps, which agrees well the experimental data of 88.4rps. For comparison, in case of C/R = 0.003, ω_1 = 44.1Hz, and λ = 0.503, thus the SLS is 86rps (experimental data 86.8rps). And in case of C/R = 0.001, ω_1 = 45.8Hz, and λ = 0.5, thus the SLS is 93rps (experimental data 93.7rps).

As one more example, the results of applying this simplified determination to the reference literature [3] are shown in Fig.6. The left figure corresponds to a case where the disk is on the left end, and the number of revolutions for whirl generation is high, that is , an unstable condition soon liable to shift to whip. The right figure corresponds to a case where the disk is on the right end, where the system is unstable in the low speed side, liable to generate whirl. Further rise in rotation results in shift to whip, while the frequency is saturated at the shaft bending natural frequency ω_z. In the left figure, the lowest degree natural frequency ω_1 = 2,880cpm, and λ = 0.44, thus the SLS is 6,545rpm (experimental data about 7,000rpm). In the right figure, the lowest degree natural frequency ω_1=1,661cpm, and λ = 0.44, thus the SLS is 3,775rpm (experimental data about 3,600rpm). As mentioned above, application of this simplified determination to the reference literatures [1]and [2], is valid for estimation.

Lesson

In place of complicated complex calculations, this simplified determination is found valid. The shaft bending natural frequency ω_z can be measured, but the oil spring natural frequency ω_d is difficult to determine, which remains problematic.

References

[1] KIKUCHI Katsuaki et al., "Safety analysis of multi-bearing multi-disk rotating system", Transaction of the Japan Society of Mechanical Engineers, Vol.42. No.368, 1977

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[4] MATSUSHITA Osami, "Vibrations of rotating machinery-Fundamentals of practical vibration analysis, (2009), Corona Publishing
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Keywords

Slide bearing, magnetic bearing, oil whip, oil whirl, approximate calculation of determinants

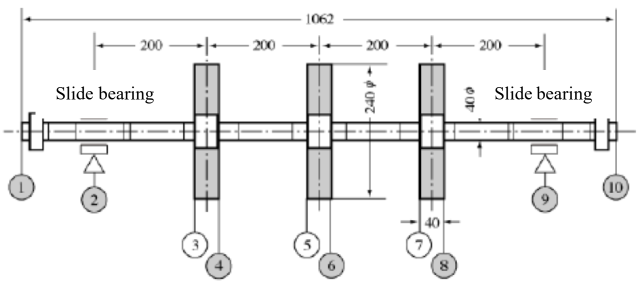


Fig.1 Schematic diagram of rotor [1)4)]

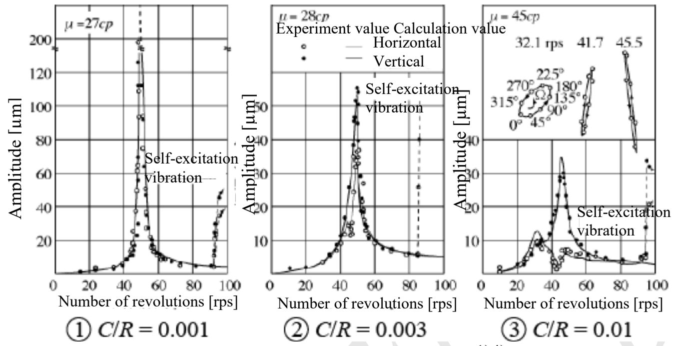


Fig.2 Experimental results [1)4)]

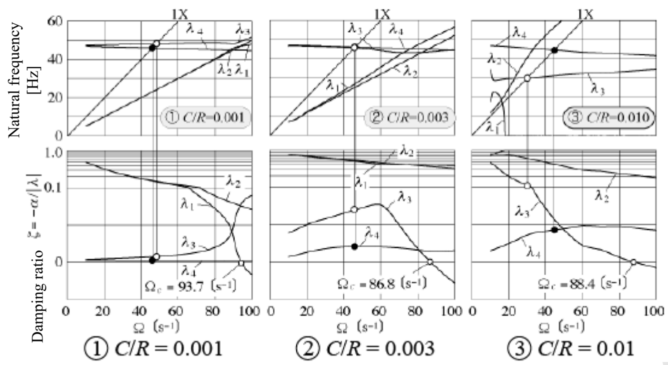


Fig.3 Results of complex eigenvalue calculation [1)4)]
(dynamic bearing characteristics: 8 parameters)

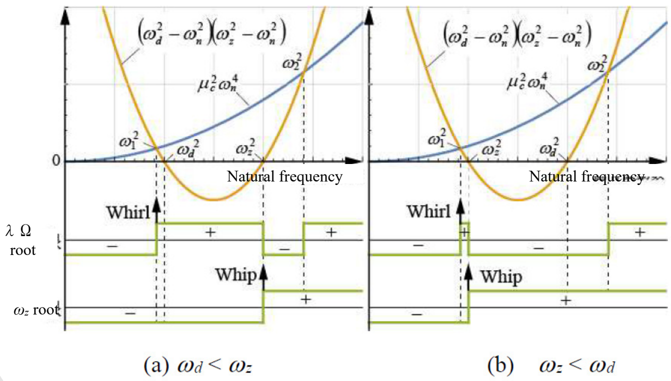
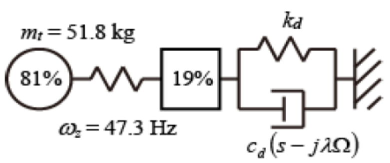


Fig.4 Characteristic roots and stability



C/R	①0.001	②0.003	③0.01
k_d	48×10^6	17×10^6	8×10^6
c_d	2.94×10^6	0.13×10^6	0.02×10^6
λ	0.5	0.503	0.493
ζ_d	29	2.2	0.49

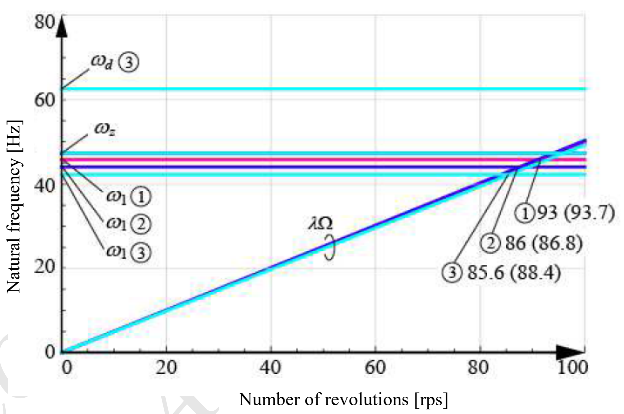


Fig.5 Results of simplified determination applied to the reference literatures (1)

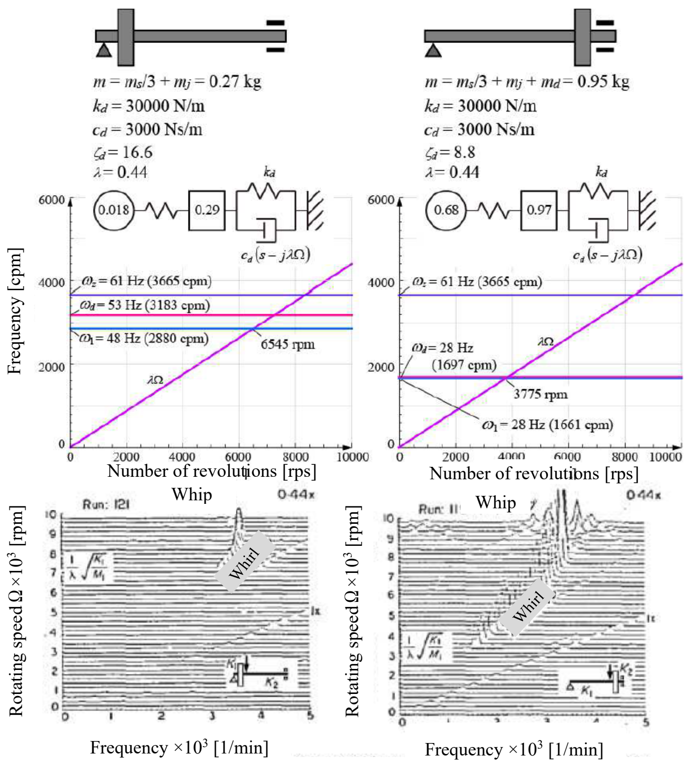


Fig.6 Results of simplified stability determination [1)4)] applied to the reference literatures (3)