Case History	Flexible Modal Balancing of AMB Rotor by Feed Forward Excitation	Active control
Resonance		

Object Machine

Rotating machinery with active magnetic bearing: centrifugal compressor

Observed Phenomena

Vibration amplitude increased when passing the critical speed of bending mode.

Cause Presumed

Under a rotor bending mode, the potential energy component fed by active magnetic bearings is small compared to a shaft strain energy. Similarly, the positive damping effect to be provided by active magnetic bearings is small compared to that provided by oil film bearings, so that it is considerably difficult to realize a desired modal damping ratio. As a result, the vibration amplitude increases when passing the bending critical speed. Thus, only a high precision bending mode balancing is to be performed.

Analysis and Data Processing

For the ordinary two bearing system rotor, as shown in Fig.1, it is assumed that the following two types of modes independent of bearing conditions are synthesized.

(a) Rigid body eigenmode under the free-free boundary condition, that is, two rigid body modes of δ_p (parallel) and δ_c (conical), and

(b) Pure rotor bending mode group (φ) , assuming that bearings are pin-pin supported. Thus it is called pin-pin eigen mode.

As shown in Fig.2, the actual resonance mode at the critical speed is considered by the superposition of a rigid body mode (δ) and a pure bending mode (φ). The superposed mode consists of the shaft bending deformation from the bearing center line between the left and right bearings. Thus, the rotor unbalance vector U acts as a modal unbalance force for the δ (parallel, conical) mode and the φ mode. As the balance cancels this unbalance force, it is equivalent to the N+2 plane balance. Fig.3 schematically illustrates these facts as an unbalance response.

First, the weight ratio of W_{c1}*, W_{c2}* and W_{c3}* was obtained. That is, the combination only for the bending mode is obtained where the rigid mode is already balanced.

For example,

$$\begin{array}{l} \delta_p W_c=0 \\ \delta_c W_c=0 \\ \Phi W_c=1 \end{array} \quad \Longrightarrow \quad \begin{array}{l} W_{c1}+W_{c2}+W_{c3}=0 \\ -W_{c1}+W_{c3}=0 \\ 0.78W_{c1}+1.0W_{c2}+0.71W_{c3}=1 \end{array} \qquad (?)$$

Consequently, the ratio was: $W_{c1}^* = 0.57$ $W_{c2}^* = 1.0$ $W_{c3}^* = -0.45$

Countermeasures and Results

As indicated in Fig.1, a synchronous harmonic shaker with the rotation was attached to apply FF excitation by way of active magnetic bearings. The magnitude and phase of the excitation force (signal of a synchronous two phase oscillator with rotation) was manually operated so as to reduce the unbalance resonance amplitude while watching a vibration monitor during critical speed operation. Then, based on these operation results, the rotor was scraped to reduce the modified weight equivalent to the excitation force.

Determination of a weight is described below. Let the excitation force be F,

$$\frac{\alpha \Phi W_c^* \gamma}{g} = (\Phi_{B1} + \Phi_{B2}) \cdot F$$

where,

Φ_{B1}=deflection of bearings (i=1, 2) = (0.41+0.51) = 0.92

γ: magnification, α=-0.72 contribution rate

As a result, $W_c=\gamma W_c^*$ can be obtained as $W_{c1}=-1.9$, $W_{c2}=3.2$, $W_{c3}=-1.4$

As mentioned above, a balancing operation was performed by combining the FF excitation and the N+2 plane balancing method, with the resultant response diagrams obtained as given in Fig.4 and Fig.5.

SHOT 1: Response of rotor bending resonance before balancing (dashed line)
SHOT 2: The FF excitation was applied while rotating the rotor around the critical speed, so as to find the magnitude and phase of the excitation force that would cause the excitation effective vector to turn toward the original point, for cancelling the initial unbalance. Then, the excitation was applied to allow the critical speed to be passed.
SHOT 3: The electromagnetic excitation force on the active magnetic bearings was converted to the N+2=3 plane correction weight, and was balanced as shown in Fig.6 as confirmation test.
In this example, complete conversion including rigid body unbalance left a substantial unbalance "□" as in Fig.6. However, as this unbalance needs not be modified, only the bending mode balance using three plane " □" was performed, which produced a significant effect.

Lesson Learned

High speed rotating bodies passing the bending critical speed require balancing appropriate for flexible rotors.

References

Transactions of the 69th National Convention of the Japan Society of Mechanical Engineers (Vol. C) (1991-10-16, at Nagoya)

Keyword

Active magnetic bearing, flexible rotor, balance, resonant magnification, control, FF(feed forward excitation), unbalance cancellation

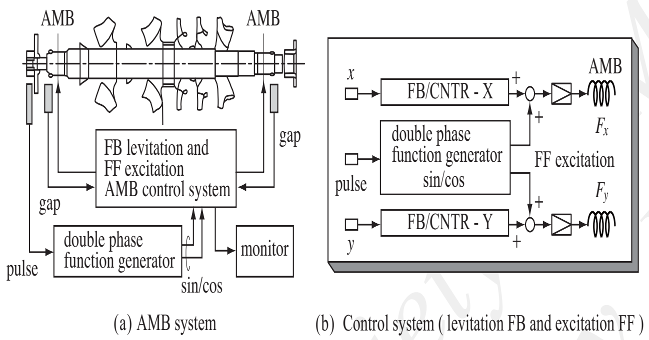

Fig. 5·40 AMB equipped compressor and bending modal balance[24)]

Fig.1 A magnetically borne flexible rotor with FF excitation

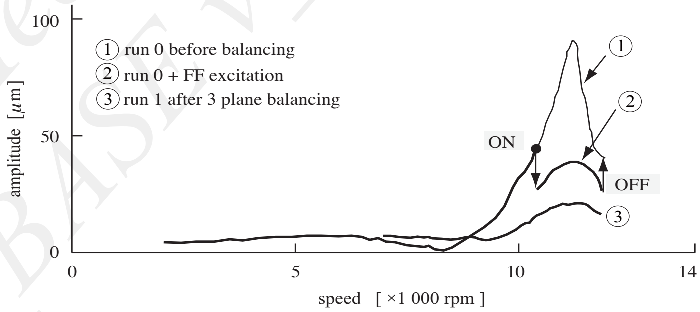

Fig. 5·41 Resonance curves

Fig.4 Feed forward excitation and balancing test results

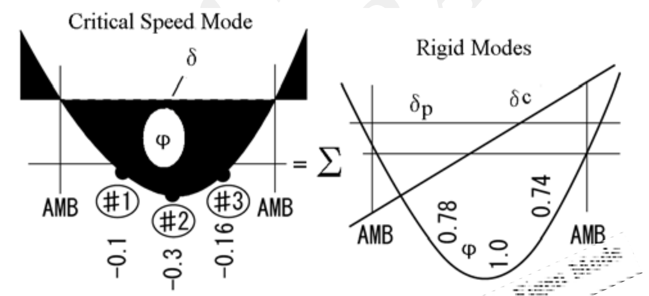

Fig.2. Modal synthesis

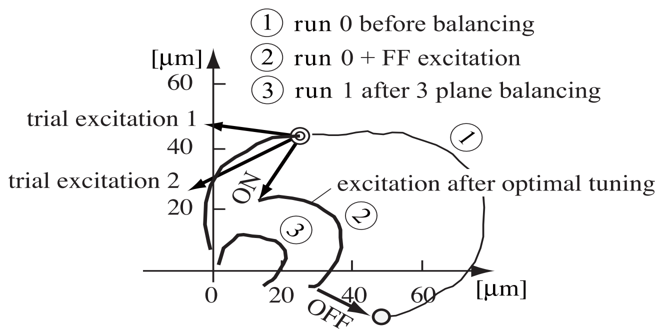

Fig. 5·42 Polar plots

Fig.5 Unbalance vibration response

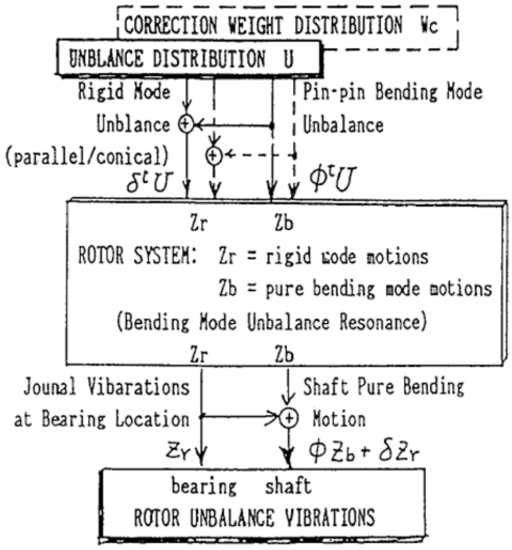

Fig.3 Unbalance/correction weight vs vibration response

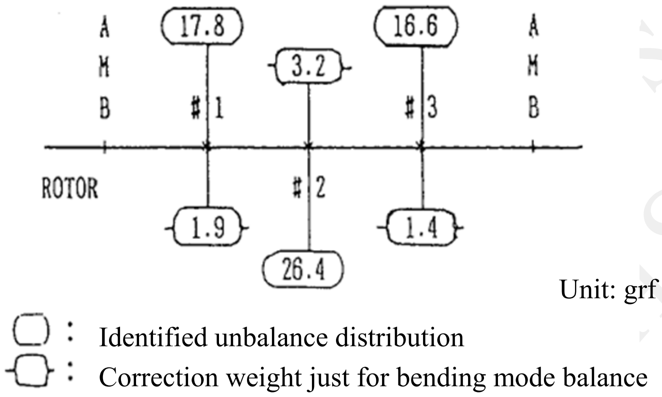

Fig.6 Unbalance identification and correction weight

★ Balancing at the mode bottom requires less modification weight, and seems more practicable.