Rotordynamics of Foil Bearing Supported High Speed Rotors

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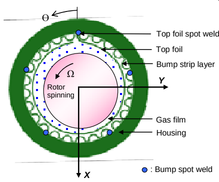
Abstract

Commercial microturbomachinery (MTM) for distributed power generation implements gas foil bearings due to their nearly friction-free operation, ability to operate at high temperatures, and tolerance to static and dynamic loads. The load capacity of a foil bearing depends mainly on the resilience of its support structure with mechanical energy dissipation from material damping. The foil bearing structure is highly nonlinear, with hardening characteristics as rotor displacements increase. For operation at rotor speeds well above the system critical speed, the bearings' nonlinearity determines multiple frequency rotor motions with large amplitudes at the system natural frequency. The large motions impair the efficiency and reduce the reliability of the MTM.

Gas Foil Bearings (GFBs)

Advantages of GFBs

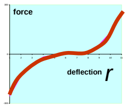
- Elimination of complicated lubrication and sealing systems.
- Reduction of drag power losses and heat generation.
- Reduction in weight and number of components.
- Improvement of overall engine efficiency & reliability.



Simple model of bump-type GFB

The foil bearing structure is hardening and, based on experimental data, its reaction force (F_{FB}) is a nonlinear function of the deflection (r)

$$F_{FB} = K_1 r + K_2 r^2 + K_3 r^3$$



Energy dissipation: Loss Factor γ

Measurements show foil bearings dissipate mechanical energy by dry-friction due to relative motion between the bumps and bearing cartridge and bumps and top foil. A loss factor defines the material damping and it is related to a viscous damping ratio ζ by

$$\zeta \approx \frac{\gamma}{2};\ \gamma = 0.14 \text{ from tests [1,2]}$$

Research objectives

Reproduce rotordynamic measurements of a test rotor supported on gas foil bearings

-Operation with various mass imbalances inserted at each end of the rotor : 55 mg, 165 mg, 330 mg, 660 mg (same angle)
- Model with and without the flexible coupling located at the rotor drive end
- Comparison of rotor amplitude of synchronous response with experimental data

Rotordynamics model

A FE rotordynamics program integrating a structural nonlinear model for the bearings predicts the time response of a test rotor (Fig. 1) supported on gas foil bearings for various mass imbalance conditions, as in prior experiments. A slender rod and flexible coupling connect the rotor to a drive motor.

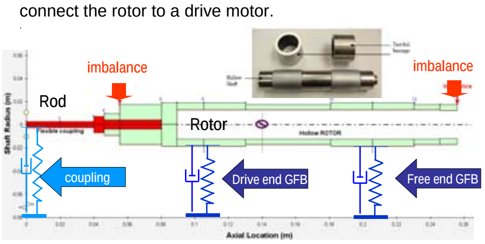


Fig. 1: Test rotor and foil bearings, and rotor structural FE model with flexible coupling at drive end (left)

Refs [2,3]:

Rotor Specifications			
Total Length	Total Mass	C.G. location	O.D. at bearings
m	kg	m	m
0.2559	1.0331	0.1398	0.381
Bearing Specifications			
K_1	K_2	K_3	γ
N/m	N/m^2	N/m^3	-
6.75E+04	-2.00E+09	1.00E+14	0.14

Numerical predictions

NORMALIZED ROTOR SYNCHRONOUS RESPONSE

As the applied imbalance mass increases, the synchronous rotor response becomes increasingly nonlinear (Figs. 2). The nonlinearity is exacerbated by the removal of the coupling (Fig. 2b), with larger vibrations of the flexible connecting rod. The rotor response amplitude in the vertical direction is similar, thus showing circular orbits. The responses below are normalized to the smallest mass imbalance. The predictions agree well with experimental data, Ref. [1]

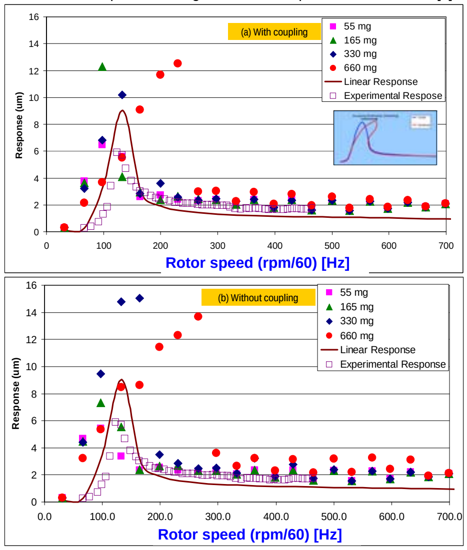


Fig. 2: Normalized amplitude of rotor synchronous response compared to experimental data (410 mg) [1]. Nonlinear predictions for (a) with coupling and (b) without coupling. Motions at rotor free end, horizontal plane. Linear response obtained with constant bearing force coefficients.

NORMALIZED SUBSYNCHRONOUS RESPONSE

The system response becomes increasingly nonlinear as the imbalance mass grows, with onset and persistence of subsynchronous whirl motions of large amplitudes. (Figs. 3a-b). More subsynchronous activity is evident with the removal of the coupling (Fig. 3b).

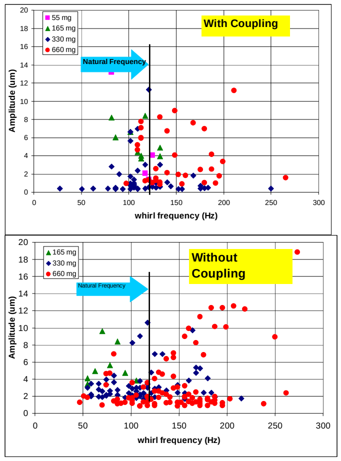


Fig. 3: Normalized amplitudes of rotor subsynchronous motions versus whirl frequency. Predictions for (a) with coupling and (b) without coupling. Motions at rotor free end, horizontal plane.

Predictions (cont'd)

WATERFALLS OF ROTOR RESPONSE

Waterfall plots of predicted rotor responses for shafts speeds from 2 to 50 krpm show complex motions with synchronous and sub synchronous frequencies (Figs. 4a-d). The larger the mass imbalance, the greater both the synchronous and subsynchronous responses. Removing the flexible coupling exacerbates the motion amplitudes, specially for whirl at the system natural frequency **(~133 Hz=7,980 rpm)**.

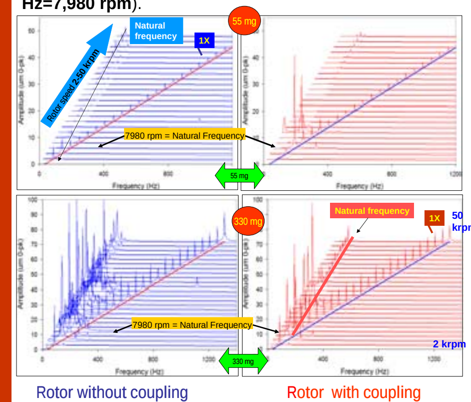


Fig. 4: Waterfalls of rotor response for 55 mg (top) & 330 mg (bottom) imbalance masses with coupling (red) and without coupling (blue). Motions at rotor free end, horizontal plane.

Conclusions

From the FE predictions, the magnitude of imbalance mass increases not just the amplitude of synchronous response but also the onset and severity of subsynchronous whirl motions. Larger imbalances force a greater nonlinear tendency, with the bearings exhibiting an increase in their stiffness, or "hardening," as the amplitude of response increases. Operation with rotor speeds larger than twice and three times the system critical speed induces the subsynchronous whirl motions with a frequency coinciding with the system natural frequency. Removal of the coupling intensifies the nonlinear subsynchronous whirl motions. The flexible coupling aids to improve system reliability. The nonlinear predictions agree well with test data.

References

1) San Andrés, L., and Kim, T., 2007, "Issues on Instability and Forced Nonlinearity in Gas Foil Bearing Supported Rotors," Paper AIAA-2007-5094, 43rd AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, Cincinnati, OH, July.
2) Rubio, D. and San Andrés, L., 2007, "Structural Stiffness, Dry-Friction Coefficient and Equivalent Viscous Damping in a Bump-Type Foil Gas Bearing," ASME Journal of Engineering for Gas Turbines and Power, 129, pp. 494-502
3) Rubio, D. and San Andrés, L., 2006, "Bump-Type Foil Bearing Structural Stiffness: Experiments and Predictions", ASME Journal of Engineering for Gas Turbines and Power, 128, pp. 653-660.

Acknowledgements

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