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ADVANCES IN NONLINEAR ROTORDYNAMICS OF PASSENGER VEHICLE TURBOCHARGERS: A VIRTUAL LABORATORY ANCHORED TO TEST DATA

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ABSTRACT

Passenger vehicle turbochargers (TCs) offer increased IC engine power and efficiency. TCs operate at high rotational speeds and use engine oil in their bearing support system comprising of inner and outer lubricant films acting in series. The hydrodynamic bearings induce instabilities, i.e. subsynchronous shaft motions over wide operating speed ranges [1]. Yet, the motions are well bounded limit enabling the TC continuous operation [2, 3]. Due to the lack of accurate and efficient predictive nonlinear tools, turbocharger rotordynamic design followed, until recently, costly test stand iteration [3]. Presently, a rotordynamics model coupled to a bearing lubrication model calculates the nonlinear motions of TCs and delivers predictions of TC shaft dynamic response for practical conditions [4-6]. The software emulates a virtual laboratory, effectively aiding to design better TC products with increased reliability in a shorter cycle time. Predictions of the nonlinear model compare well with recorded TC shaft motions, both in amplitude and frequency content. The benchmarking lends credence to the validity of the integrated computational model.

INTRODUCTION

Modern passenger vehicle TCs feature engine oil lubricated semi-floating ring bearings (SFRB) due to their low cost and reduced power losses [3]. References [4-6] report notable advances in the prediction of the rotordynamic response of TCs. Holt *et al.* [4] describe the integration of the SFRB flow model to a rotordynamics FE program. The computational model, driven by a GUI, predicts the TC system damped eigenvalues for linear stability analysis and the synchronous response using linearized SFRB stiffness and damping coefficients. The program features a time-marching numerical integration of the rotor-bearing nonlinear equations of motion and predicts the shaft limit cycle response for prescribed operating and maneuver conditions. The model includes thermal transport in the bearing supports for estimation of the lubricant viscosity and thermal growth of the rotor, bearing and floating ring [5]. Post-processing frequency domain analysis forwards the total orbital motion, amplitude of synchronous response, and magnitudes with frequencies of subsynchronous motions [6].

ILLUSTRATIVE TEST RESULTS AND PREDICTIONS

Fig. 1 depicts the structural FE model of a test TC rotor and semi-FRB support. The turbine wheel is welded to one end of a shaft. The other shaft end is threaded for installation of the compressor wheel, pushing the thrust collar and thrust spacer towards the shaft shoulder. The ring is a one piece cylinder; the oil film lands, inner and outer, are machined at the ring ends. Feed ports in the casing supply lubricant to each outer film, and small holes on the ring bring oil into the inner films. A pin locks the ring and prevents its rotation.

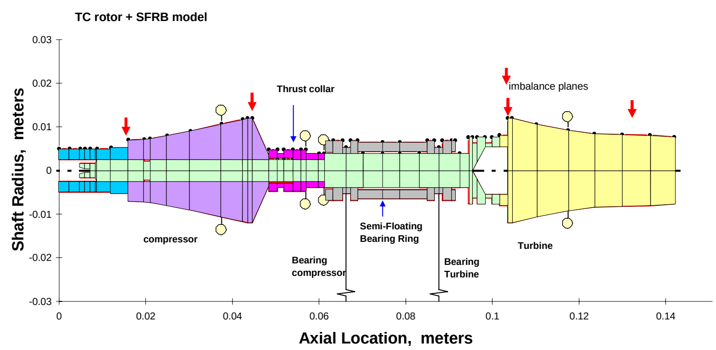

Fig. 1 Structural FE model of turbocharger rotor and semi-floating ring bearing

Hot gas tests recording the TC shaft motions at the compressor nose were conducted with 0W-30 lubricant of 10 cPoise at a nominal inlet temperature and pressure equal to 93 °C and 4 bar, respectively. Imbalances were located on four planes, at the nose and back face of the turbine and compressor wheels (see Fig. 1).

Fig. 2 displays a waterfall of the recorded shaft motions for speeds from 30 to 184 krpm, and with magnitude relative to the physical limit. The test data show the persistence of severe subsynchronous frequency motions over the entire speed range. The shaft subsynchronous amplitudes are notoriously large with a broad-band spectrum, in particular at low frequencies that represent the rigid body modes of the test unit.

Fig. 3 displays a waterfall of the nonlinear predictions. As in the tests, persistent subsynchronous motions are apparent over the entire shaft speed range; albeit without the broad spectra at low frequencies, presumably due in the tests to looseness of the anti-rotation pin in the SFRB.

Fig. 4 compares the measured and predicted total motion, i.e. size of shaft orbit. Fig. 5 depicts the amplitude of synchronous (1X) motion versus shaft speed. The 1X predictions correlate well with the measurements, in particular the nonlinear ones at low and high shaft speeds. The predicted total motion correlates well with the test values in the mid range of shaft speeds. Note that the predictions show a continuous reduction in total motion as speed increases, denoting a transition toward stable operation, i.e. free of subsynchronous motions. On the other hand, the test data shows no apparent trend and evidences severe subsynchronous motions at the top rotor speed.

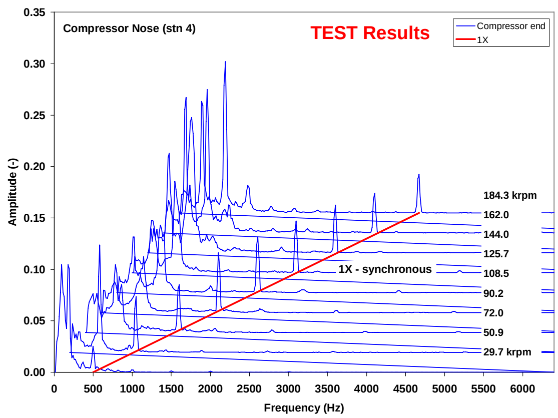

Fig. 2 Waterfall of measured shaft displacements at compressor end. Shaft speed range 30 to 184 krpm, 4 bar oil pressure and 93°C oil temperature

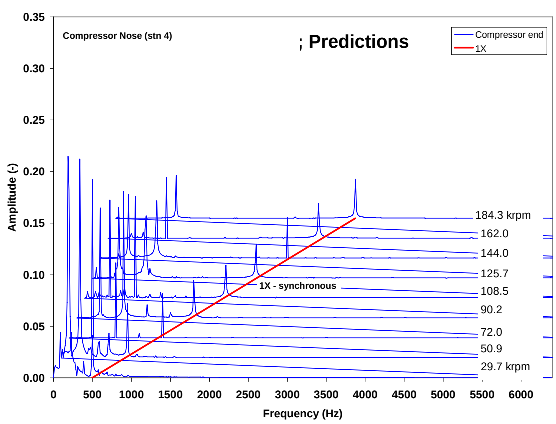

Fig. 3 Waterfall of predicted shaft displacements at compressor end. Conditions same as in Fig. 2

CLOSURE

A brief description of a computational analysis emulating a virtual laboratory for evaluation of TC nonlinear rotordynamics is presented. The predictive tool is currently in use to design high performance products with faster development cycle times and increased product reliability. There is good correlation between predicted and measured shaft motions, in particular for synchronous response amplitudes, and magnitude and frequency content of sub synchronous motions.

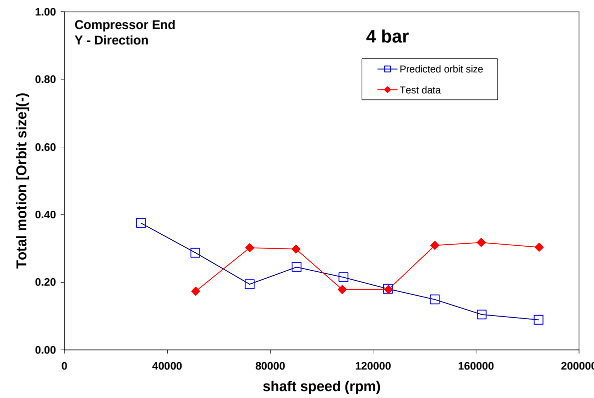

Fig. 4 Nonlinear prediction and test data of total shaft motion magnitude at compressor end versus rotor speed.

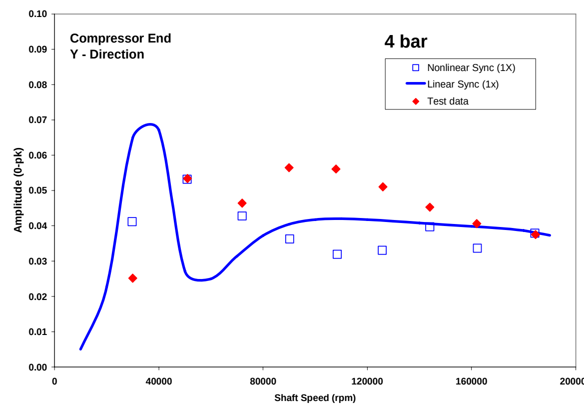

Fig. 5 Amplitude of shaft synchronous motion versus rotor speed. Test data, linear and nonlinear predictions

REFERENCES

[1] Tanaka, M., 1996, "A Theoretical Analysis of Stability Characteristics of High Speed Floating Bush Bearings," Proc. 6th Int. Conf. on Vibrations in Rotating Machinery, IMechE Paper C500/087/96, pp. 133-142.

[2] Holmes, R., Brennan, M.J., and Gottrand, B., 2004, "Vibration of an Automotive Turbocharger," Proc. 8th Int. Conf. on Vibrations in Rotating Machinery, IMechE Paper C623/011/2004, pp. 445-455.

[3] Sahay, S. N., and LaRue, G., 1996, "Turbocharger Rotordynamic Instability and Control," Proc. 8th Workshop on Instability in Rotating Machinery, Texas A&M University, May, pp. 1-11.

[4] Holt, C., San Andrés, L., Sahay, S., Tang, P., LaRue, G., and Gjika, K., 2005, "Test Response and Nonlinear Analysis of a Turbocharger Supported on Floating Ring Bearings," ASME Journal of Vibrations and Acoustics, Vol. 127, pp. 1-9.

[5] Kerth, J., and San Andrés, L., 2004, "Thermal Effects on the Performance of Floating Ring Bearings for Turbochargers", Journal of Engineering Tribology, Vol. 218, 5, pp. 437-450.

[6] San Andrés, L., Rivadeneira, J.C., Gjika, K., Chinta, M., and LaRue, G., 2005, "Nonlinear Rotordynamics of Automotive Turbochargers – Predictions and Comparisons to Test Data," ASME Paper GT 2005-68177.

