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Unbalance Response Prediction for Accelerating Rotors With Load-Dependent Nonlinear Bearing Stiffness

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Rolling-element bearing forces vary nonlinearly with bearing deflection. Thus an accurate rotordynamic analysis requires that bearing forces corresponding to the actual bearing deflection be utilized. Previous papers have explored the transient effect of suddenly applied imbalance and the steady-state unbalance response, using bearing forces calculated by the rolling-element bearing analysis code COBRA-AHS. The present work considers the acceleration of a rotor through one or more critical speeds. The rotordynamic analysis showed that for rapid acceleration rates the maximum response amplitude may be considerably less than predicted by steady-state analysis. Above the critical speed, transient vibration at the rotor natural frequency occurs, similar to that predicted for a Jeffcott rotor with constant-stiffness bearings. A moderate amount of damping will markedly reduce the vibration amplitude, but this damping is not inherent in ball bearings.

Introduction

Rotordynamic response of all but very flexible rotors depends strongly on bearing properties. When bearings are nonlinear, accurate rotor response calculations require use of variable bearing properties that reflect the precise conditions encountered, rather than average properties. While fluid film bearings are often reasonably linear for small deflections (although there is usually a strong speed dependence) rolling-element bearings have a much less linear force-displacement relationship.

Previous papers by the first and third authors investigated the effect of nonlinear ball bearings on the transient response of a rotor to suddenly applied imbalance at constant speed [1] and the steady state response over a speed range [2]. Both papers made use of a rolling element bearing analysis recently developed by Poplawski et al. [3]. These papers showed that accurate results required the use of the nonlinear bearing properties; results could not be duplicated with any constant bearing stiffness.

The present work extends the investigation to the case of rotor acceleration. It is conventionally thought that rotor critical speed vibration can be reduced by rapid acceleration through the critical speeds. This was analytically and experimentally demonstrated by Hassenpflug et al. [4] for the case of a Jeffcott rotor on rigid bearings. Genta and Delprete [5] also studied the unbalance response of accelerating rotors taking into account their nonlinear behavior and geometrical or inertial anisotropy. However, they apparently did not attempt to use load-dependent bearing properties. In this report, response is calculated for the same rotor and conditions as [2] for various values of angular acceleration. Realistic bearing properties for a specific ball bearing are utilized.

Analytical System and Procedure

Figure 1 is a drawing of the shaft system. It depicts a fairly stiff shaft with concentrated masses (which may represent compressor or turbine wheels) at stations 3 and 4. In total, 6 stations and 5 elements were used in the model (with some of the elements having two or more subelements). Radial (deep groove) ball bearings of 25 mm bore diameter are at stations 2 and 5. The shaft material is steel with length of 300 mm and diameters of 38 mm in the central section and 25 mm at the ends. Total mass of the shaft system is 4.6 kg.

Figure 2 shows the first two system critical speeds as a function of bearing stiffness. For stiffness up to about 200 MN/m, critical speeds rise rapidly with stiffness, indicating significant bearing participation in the rotor motion. For higher stiffness values, critical speeds do not increase as rapidly, indicating that increasing amounts of motion are due to shaft bending rather than bearing deflection. Mode shapes calculated at critical speeds, presented in [2], confirm this.

The rotordynamics code DyRoBeS (Dynamics of Rotor Bearing Systems) [6] was chosen for this work. This versatile code has recently been enhanced to allow calculation of response when rotor speed varies in a preset manner, in the present case constant acceleration from 10,000 to 80,000 rpm.

The bearing code COBRA-AHS [3] was used to generate load versus deflection data for the bearings at a speed of 40,000 rpm and loads from 44 to 8800 N. A power series curve of the form $\boldsymbol{F = a + b\, e^{c}}$ was fitted to the data, where $\boldsymbol{F}$ is bearing force, $\boldsymbol{e}$ is bearing deflection, and $\boldsymbol{a}$**,** $\boldsymbol{b}$**,** and $\boldsymbol{c}$ are coefficients. The coefficients resulting from this curve fit were then supplied to the rotor response code. Figure 3 shows bearing load versus deflection. Both the bearing data and the fitted curve are shown; the curve fits the data very well, verifying that the mathematical representation of the code output is adequate. Note that the curve is decidedly nonlinear; this is representative of rolling-element bearings. Bearing data from only one speed were used because DyRoBeS cannot handle the actual bearing load variation with speed; however, the load varies much less with speed than with deflection.

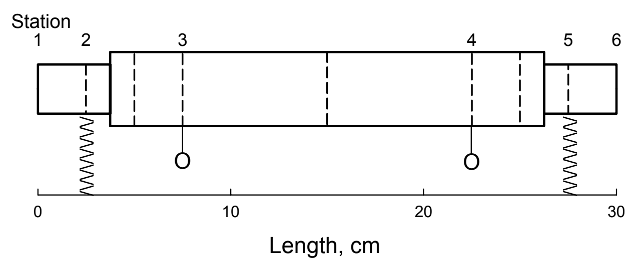

Figure 1.—Sketch of rotor.

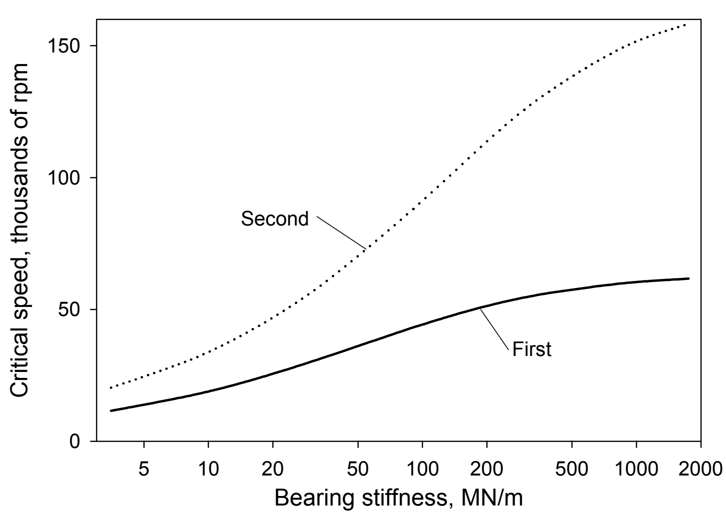

Figure 2.—Critical speed map.

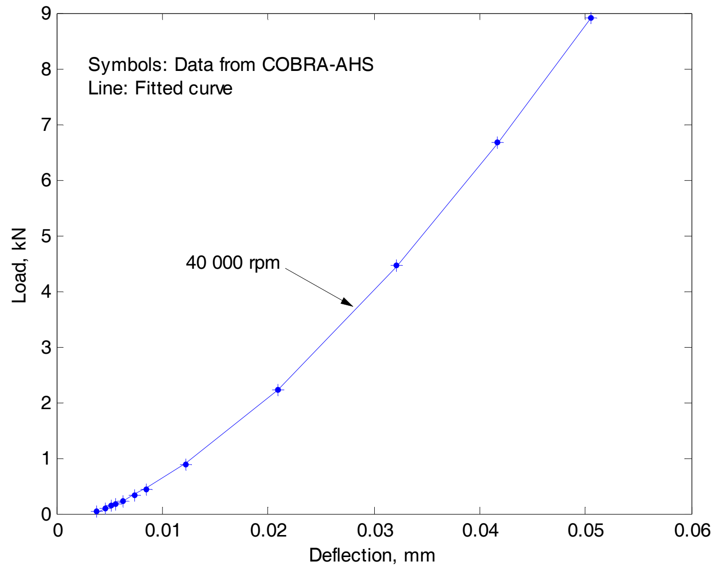

Figure 3.—Force-deflection characteristics for ball bearing at 40,000 rpm.

The median value of imbalance from [2], 1.2 g-cm, was used for the present study (the imbalance was incorrectly reported in the earlier paper as 12 g-cm). DyRoBeS was given times ranging from 0.01 to 5 seconds to accelerate from 10,000 to 80,000 rpm.

Results

Figure 4 displays the amplitude at station 5, the bearing station nearest the imbalance, for acceleration times of 0.01 to 1 second, plus the steady state amplitude. Response is plotted as a function of rotor speed. Not shown is the response for an acceleration time of 5 seconds, which is nearly identical to the steady state values, indicating that for low acceleration rates the response follows a steady-state response. These results show clearly that critical speed response amplitude is indeed reduced as acceleration rate increases (shorter acceleration time), and also that very rapid acceleration is needed for a meaningful reduction in critical speed response. The speed at which maximum amplitude occurs is greater for higher acceleration rates. This appears logical in that several revolutions are required for amplitude to build up; at higher acceleration rates, the rotor speed has increased by the time amplitude reaches a maximum.

Figure 5 shows bearing loads corresponding to the amplitudes of figure 4. The shapes of the curves are thus similar.

Vibration amplitude for station 4 (the location of the imbalance) is shown in figure 6. The appearance of this figure is similar to figure 4, but the amplitudes are higher as the critical speeds are approached because of rotor bending. As shown in [2], the first rotor resonance is a strong bending mode, and thus produces correspondingly larger amplitudes at the imbalance location.

A common feature of figures 4 to 6 is the sharp drop in amplitude as the critical speed is passed when the acceleration rate is low or nil. As discussed in [2], this is typical of systems with a hardening support, that is, where the support force increases at a rate higher than linear. This characteristic makes possible bistable operation, where for certain speeds either of two amplitudes can be stable. Thus in [2] there were different curves for rotor acceleration and deceleration. In figures 4 to 6, however, only positive acceleration was considered.

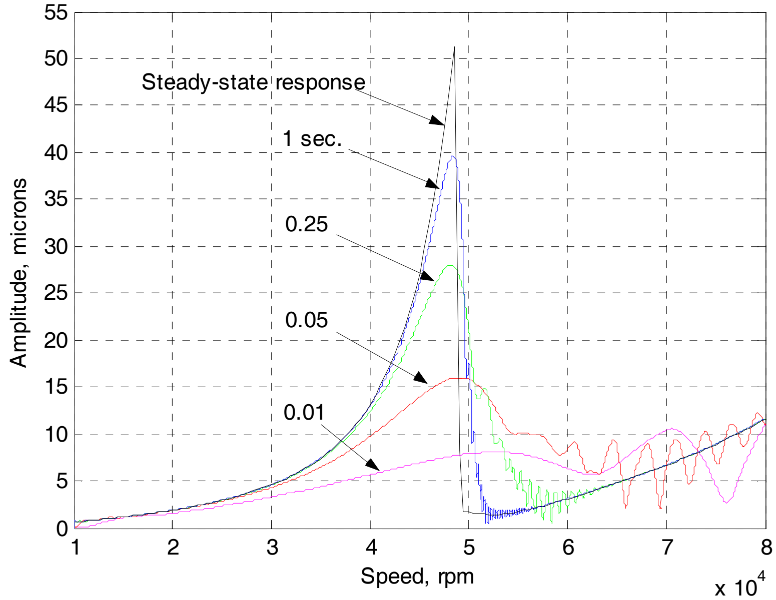

Figure 4.—Response amplitudes at station 5 for various acceleration rates; imbalance 1.2 g-cm.

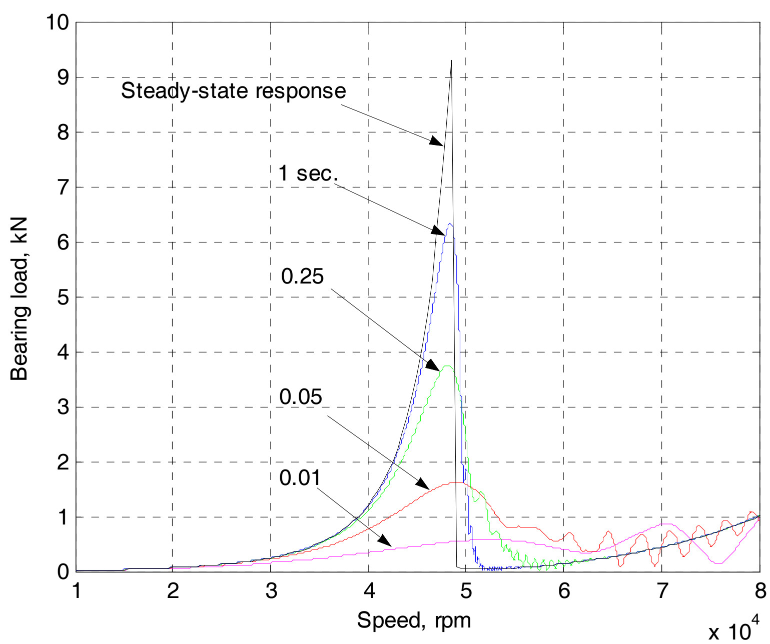

Figure 5.—Bearing loads at station 5 for various acceleration rates; imbalance 1.2 g-cm.

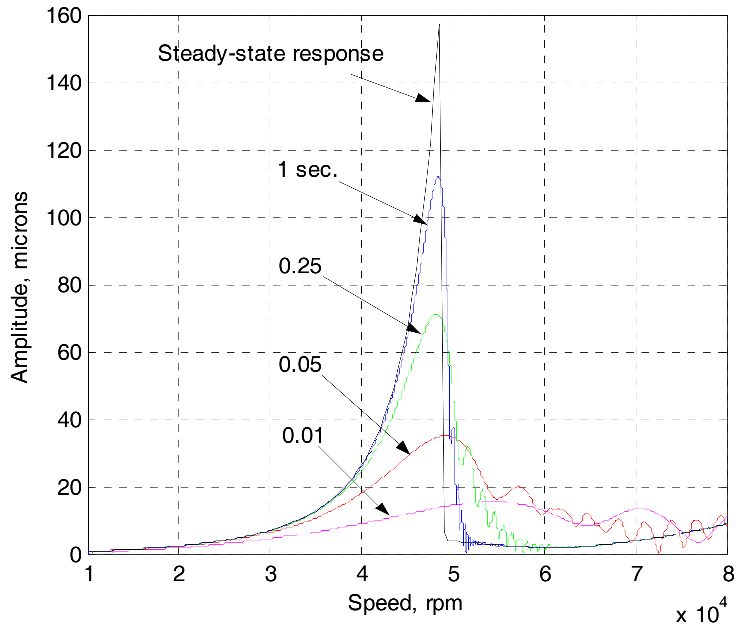

Figure 6.—Response amplitudes at station 4 for various acceleration rates; imbalance 1.2 g-cm.

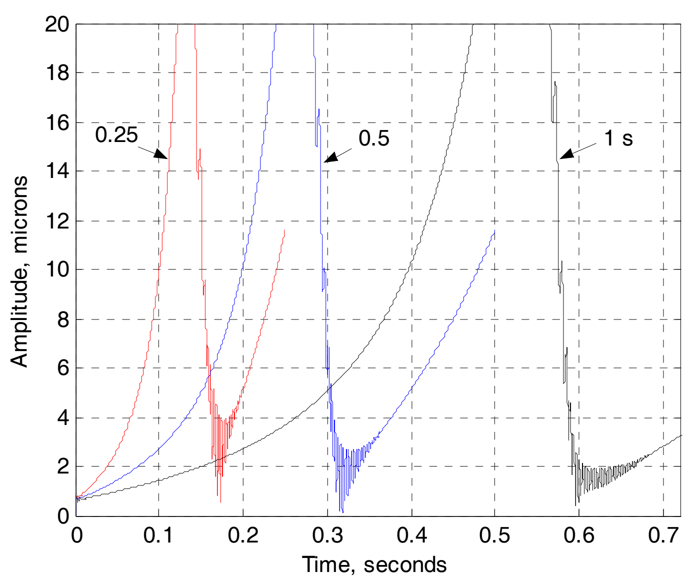

Figure 7.—Response amplitudes at station 5 versus time for various speed ramp times; imbalance 1.2 g-cm.

Another common feature of figures 4 to 6 is the oscillation in amplitude after critical speed passage. These oscillations eventually damp out. The oscillation seems to be at approximately the same frequency regardless of acceleration rate, and corresponds to the rotor natural frequency for the low bearing stiffness that would occur at the low supercritical rotor amplitudes. This is more easily discerned in figures 7 and 8, where the abscissa is time rather than rotor speed. Figure 8 uses the same data as figure 7, but with expanded time scales in the regions of the oscillations. Similar results were reported in [4] for a Jeffcott rotor on rigid supports.

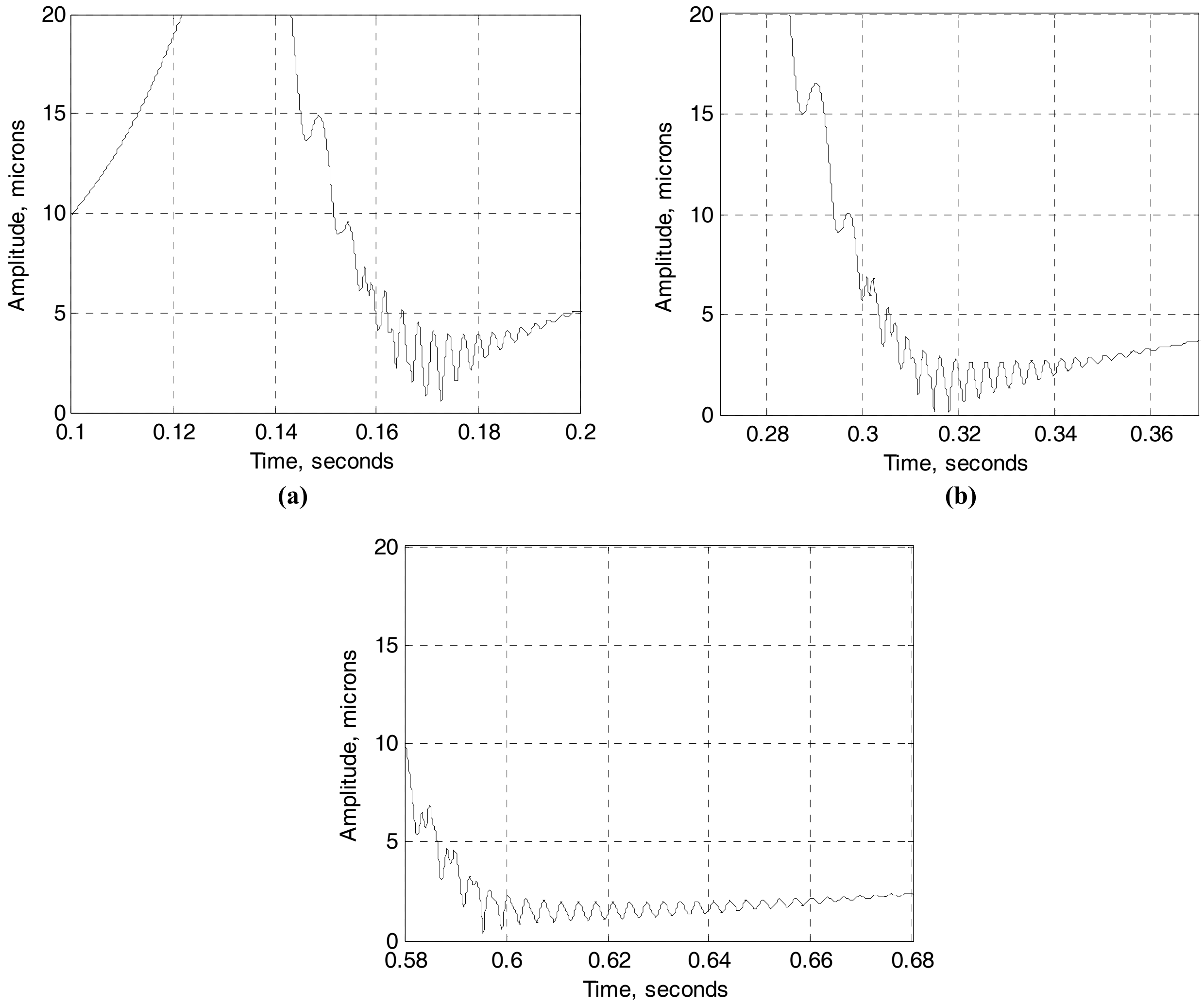

Figure 8.—Response amplitudes at station 5 with expanded time scale for various speed ramp times: (a) 0.25, (b) 0.5, and (c) 1 second; imbalance 1.2 g-cm.

Figure 9 compares results for the nonlinear bearing properties calculated by COBRA-AHS and two values of constant bearing stiffness corresponding to loads of 900 and 6,700 N. The selected speed ramp time was 0.25 sec. As was also the case in [2] for steady-state calculations, there appears to be no constant bearing stiffness that will duplicate the results for the nonlinear bearing.

Figure 10 explores results for various values of bearing damping for a speed ramp time of 0.5 sec. As discussed in [2], ball bearing damping is quite low, although precise values are difficult to determine. Damping values higher than the baseline value of 1.8 kN-sec/m result in considerably lower rotor amplitudes, as was also seen in [2] for steady state calculations. Greater damping also results in critical speed passage at a lower rotational speed, also seen in [2]. An additional effect of higher damping is the reduction or total suppression of supercritical rotor oscillations. All these effects of damping are beneficial; however, as discussed in [2], ball bearings have low inherent damping; thus in ball bearing supported machinery additional dampers such as squeeze films are used if vibration amplitudes cannot be kept low enough through careful balancing.

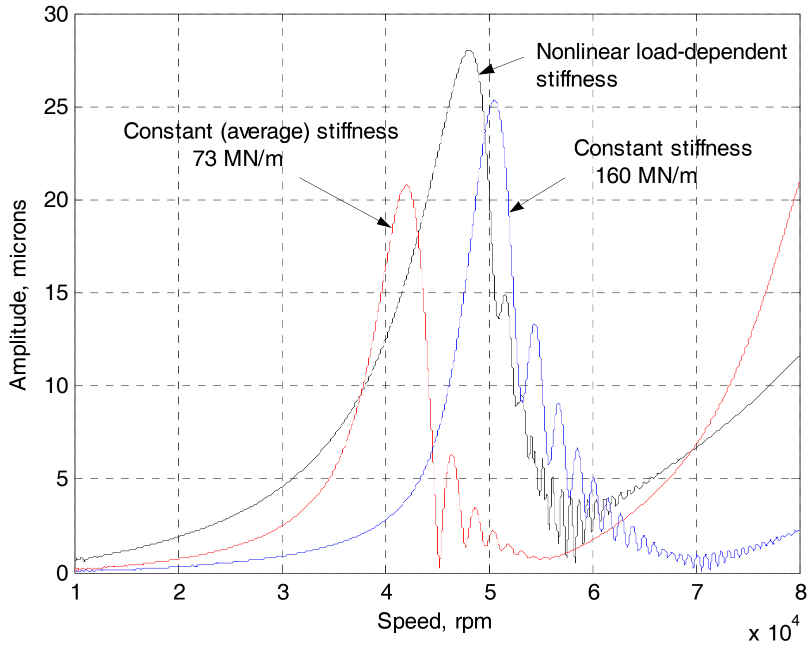

Figure 9.—Response amplitudes at station 5 for constant stiffness values and nonlinear load-dependent stiffness; speed ramp time 0.25 seconds, imbalance 1.2 g-cm.

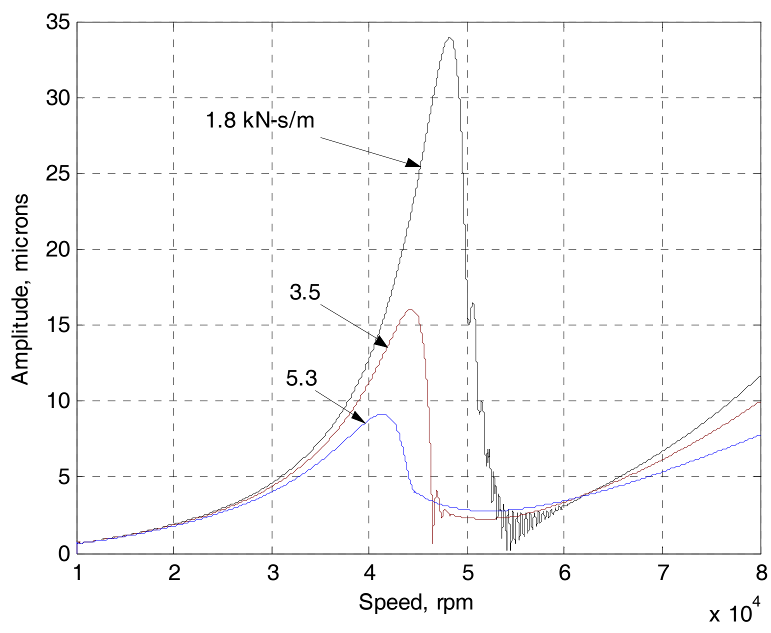

Figure 10.—Response amplitudes at station 5 for various damping values; speed ramp time 0.5 seconds, imbalance 1.2 g-cm.

Concluding Remarks

Unbalance response results were presented for an accelerating rotor supported on ball bearings with accurate bearing stiffness calculated as a function of bearing load. Reduction in critical speed amplitude was produced by rapid acceleration; however, very high acceleration rates were needed for significant amplitude reduction (ramp time of 0.05 seconds or less for acceleration from 10,000 to 80,000 rpm). Rotor transient oscillations at approximately the rotor natural frequency appeared in the supercritical speed region. A moderate amount of damping eliminated the oscillations, but this damping is not inherent in ball bearings.

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