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Effect of Bearing Dynamic Stiffness on Gear Vibration

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EFFECT OF BEARING DYNAMIC STIFFNESS ON GEAR VIBRATION

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ABSTRACT

Noise is a major consideration in the design of high performance geared transmissions, such as for helicopters. Transmission error, that is, the accuracy with which the driven gear follows the driver gear, is a common indicator of noise generation. It is well known that bearing properties have a strong influence on shaft dynamics. However, up to now the contribution of bearings to transmission error has received little attention. In this paper, a torsional-axial-lateral geared rotor analysis is used to determine dynamic transmission error as a function of bearing stiffness and damping. Bearings have a similar effect as found in shaft dynamics; transmission error can be reduced more than 10 decibels by appropriate selection of bearing properties.

INTRODUCTION

Much effort has been devoted to reducing the noise generated by high-performance geared transmissions such as those used in helicopters; see, e.g., [1, 2]. In [2], the proceedings of ASME's 2000 Power Transmission and Gearing Conference, 27 papers are devoted to the problem of gear noise. The author's experience as a passenger in a current-production helicopter verifies that this effort is warranted; the noise level in the helicopter interior is intense. In helicopter transmissions, input speeds are high (typically 15 000 to 20 000 rpm), requiring large reduction ratios to drive the main rotors at only a few hundred rpm. Minimizing weight is crucial, as weight directly affects aircraft payload, but a light-weight design reduces the noise-attenuating capability of the transmission. It is generally assumed that noise production is a byproduct of dynamic transmission error, which is the instantaneous amount by which the output gear speed deviates from the input gear speed times the gear ratio. Most of the work to reduce transmission error has been to determine appropriate gear tooth profiles, such that the theoretical gear ratio is maintained throughout the tooth engagement cycle. However, since tooth stiffness varies throughout the cycle as the contact point moves up and down the tooth flank, and varying numbers of teeth are in contact, at best the tooth profile would be ideal for only one value of transmitted load.

In the field of rotor dynamics (dynamics of rotating machinery), researchers have found that shaft vibration depends strongly on bearing support stiffness and damping properties [3, 4]. It is natural to inquire whether the proper choice of bearing properties might have a similar beneficial effect on dynamic transmission error.

This, then, is the goal of the present paper. A computer code that carries out a combined torsional-axial-lateral analysis of a geared transmission is used to calculate dynamic transmission error for varying values of bearing stiffness and damping in a simple gearset model.

ANALYSIS AND COMPUTER CODE

Blanding [5] analyzed a geared shaft system and proposed a frequency-branched transfer matrix solution. With frequency branching, a solution is obtained for only a select set of frequencies. The frequencies of importance in a geared transmission include shaft rotational speed, mesh (tooth passing) frequencies and multiples thereof, and sideband frequencies (mesh frequencies plus or minus rotational speed). Frequencies other than synchronous and tooth-passing are produced due to the nonlinear stiffness of the gear mesh, and the fact that stiffness varies throughout the engagement cycle. David [6] and David and Mitchell [7] showed the importance of including dynamic coupling of the system elements and postulated that a tractable solution could be obtained using a harmonic balance method. Park [8] developed a computer code using the techniques of Blanding, David, and Mitchell. In the code developed by Park, the user selects the frequencies for which results are desired. It is this code that will be used to investigate the effect of bearing stiffness and damping on transmission error.

TRANSMISSION MODEL

The code used can analyze complex geometries consisting of multiple flexible shafts, bearings, and gear meshes. A simple model, however, suffices to determine whether bearing properties can, in fact, have a beneficial effect on transmission error. The model is shown in figure 1. Two identical shafts carrying the gears are supported by rigid bearings at the ends farthest from the gears (i.e., the bearings are rigid in the radial direction, but have no moment stiffness). Identical "test" bearings of varying stiffness and damping support the shafts adjacent to the gears. Two 30-tooth spur gears are overhung from their respective shafts. The program input includes static transmission error and mesh stiffness as these quantities vary over the engagement cycle; the values used were those of [8]. One would normally calculate static transmission error and tooth stiffness from knowledge of the gear tooth profile, contact ratio, etc. A transmitted gear load of 2200 N was assumed. The computer code assumes massless elastic shafts with system mass contained in gears and disks attached to the shafts. For the model employed herein, the only system mass is in the gears.

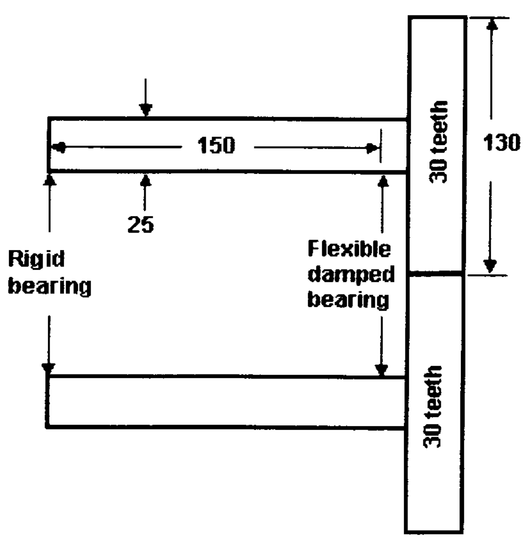

Figure 1. Gear system model (dimensions in mm).

RESULTS AND DISCUSSION

The analysis was carried out for three frequencies: shaft synchronous, mesh (i.e., 30 times shaft speed), and twice mesh. Initially, side bands of the mesh frequency were included, but for the model analyzed, sideband response was much less than for the other frequencies. Consequently, sideband response was not calculated for most of the analyses undertaken. Results showed that transmission error at synchronous frequency was independent of both bearing properties and gear speed; this is most likely due to the lack of dynamic forcing functions at synchronous frequency. Lack of synchronous excitation may also explain the low response at sideband frequencies. Transmission error at twice mesh frequency was generally greater than that at mesh frequency. Therefore bearing properties to minimize transmission error at twice mesh frequency were sought.

Figure 2 shows dynamic transmission error as a function of shaft speed for bearing stiffness of 350 MN/m and damping of 350 kN sec/m. Transmission error at mesh and twice mesh frequencies depends strongly on shaft speed; for twice mesh frequency, transmission error peaks near 114 Hz. For comparison, the static transmission error is approximately $10^{-4.2}$ for both frequencies shown in figure 2 (i.e., log TE = −4.2). Thus transmission error is amplified by system dynamics at some speeds and attenuated at others.

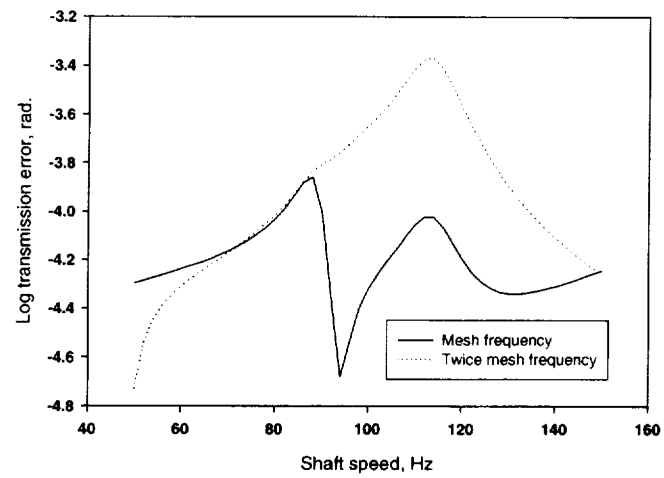

Figure 2. Dynamic transmission error. Bearing stiffness, 350 MN/m; damping, 350 kN/m.

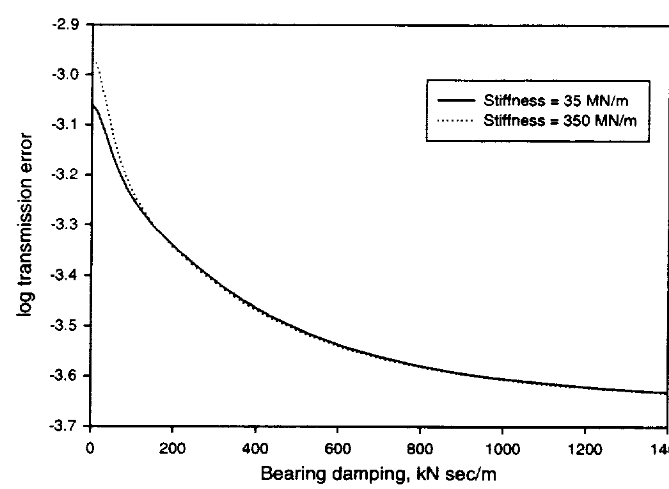

Figure 3. Dynamic transmission error at twice mesh frequency; shaft speed, 114 Hz.

Next, transmission error was calculated over a range of bearing stiffness and damping for several fixed frequencies. Figure 3 shows results for a shaft frequency of 114 Hz. The figure shows that transmission error is insensitive to bearing stiffness except at very low values of damping. However, damping does have a strong effect; transmission error drops as damping increases out to about 1400 kN sec/m where it becomes asymptotic. From this figure it appears that using the maximum damping possible will result in the minimum transmission error. However, when results are calculated over a range of speeds, one sees that this is not the case; each speed has its own optimum damping value.

Figure 4 is a contour plot of transmission error for a range of speeds and damping values; a constant stiffness of 175 MN/m was used. From this figure a bearing damping of about 350 kN sec/m is seen to be optimum. The maximum transmission error for this damping value, shown in figure 3, is about $10^{-3.4}$ (i.e., log TE = –3.4), while the maximum from figure 4 for very low bearing damping is $10^{-2.6}$. This represents a 16 dB decrease in transmission error when damping increases from the very low value typical of rolling-element bearings [9] to the optimum 350 kN sec/m. (dB = 20 times the difference in log values.) This is comparable to the vibration reduction calculated for rotor systems [3].

While the particular frequency at which transmission error peaks depends on the specific parameters of the geared system, it is expected that transmission error can be reduced in most transmissions by appropriate choice of bearing properties.

Gear lateral vibration is also of interest. Figure 5 plots the radial gear vibration, along the centerline of the gears, over a range of speeds. Maximum amplitude occurs at the same speed as does maximum transmission error (see fig. 2), and again, the highest amplitude is for twice the mesh frequency. At speeds below 90 Hz, vibration at mesh frequency is greater than at twice this frequency, but values are considerably lower than at 114 Hz. However, if this lower speed range is where the gearset normally operates, one would naturally carry out the optimization there. Vibration at shaft rotational speed, while low in amplitude, increases slowly with speed; this is expected for excitation from shaft imbalance. Figure 6 shows amplitude at 114 Hz as a function of bearing damping. Behavior is similar to transmission error (fig. 3); bearing stiffness has little effect except at low damping values, and higher damping leads to lower vibration amplitude. Although not shown, results at other speeds show that, similar to transmission error, there is an intermediate value of damping that minimizes amplitude over the speed range.

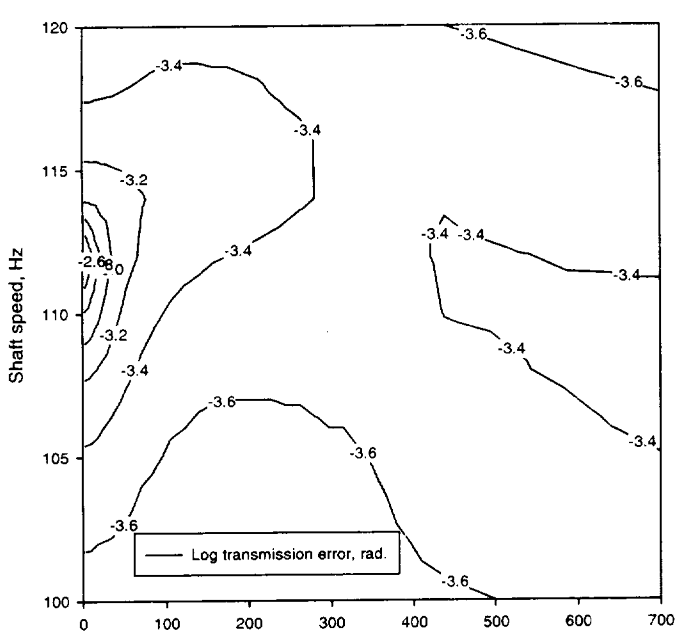

Figure 4. Dynamic transmission error at twice mesh frequency; bearing stiffness, 175 MN/m.

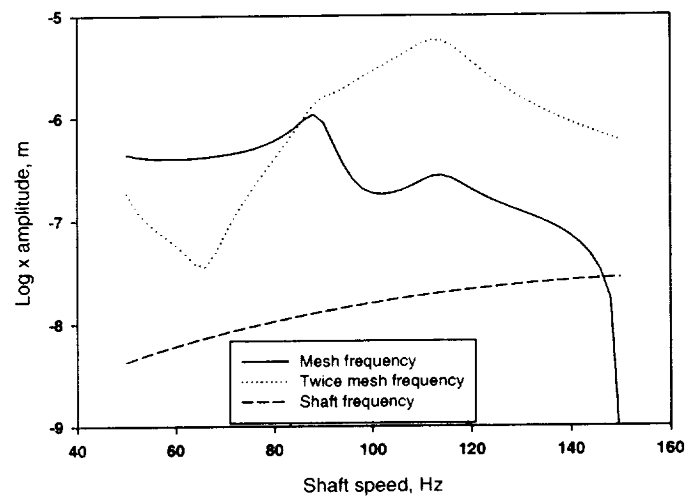

Figure 5. Lateral amplitude. Bearing stiffness, 350 MN/m; damping, 350 kN sec/m.

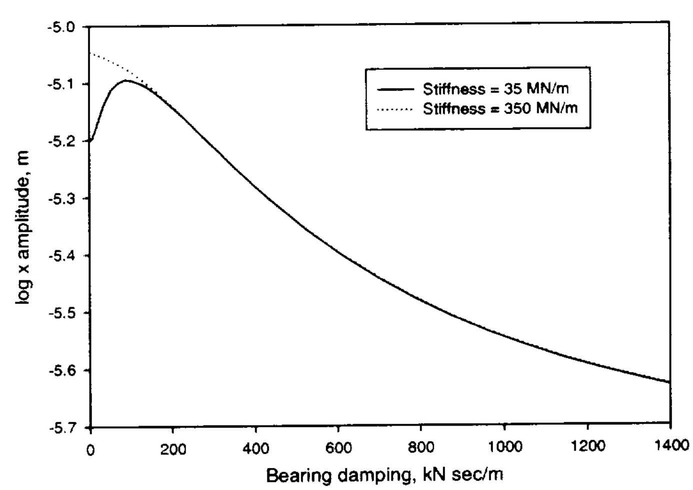

Figure 6. Lateral vibration for twice mesh frequency; shaft speed, 114 Hz.

Comparing figures 2 and 5, or alternatively figures 3 and 6, one sees that there is a correspondence between transmission error and lateral vibration. This is not a surprising result, as lateral vibration causes the gear centerline distance to change over the engagement cycle. This, in turn, changes the contact point on the gear teeth from the ideal location for which the tooth profile was manufactured. Static transmission error and contact ratio are thereby affected, with corresponding changes in mesh stiffness.

IMPLICATIONS FOR TRANSMISSION DESIGN

Most high-performance transmissions employ rolling-element bearings which inherently have very low damping, of the order of 3 kN sec/m [9]. As figures 3 and 4 show, transmission error is quite large for this low value of bearing damping. Transmission error can be substantially reduced if the bearings have more damping. This can be accomplished, in some cases, by replacing rolling-element bearings with fluid film bearings; up to 10 dB reduction in transmission noise is cited in [10]. In cases where rolling-element bearings must be used, the bearings can be mounted in dampers such as squeeze films, which can readily be designed to provide adequate damping values.

RELATION TO GEAR TOOTH PROFILE RESEARCH

Producing the optimum gear tooth shape is analogous to shaft balancing in the field of rotordynamics, while the work discussed herein corresponds to selecting optimum shaft supports to minimize shaft vibration for a given balance quality. These areas, tooth profile and shaft support, are complementary; both must be addressed to produce transmissions having minimum noise and vibration.

CONCLUDING REMARKS

Appropriate bearing properties, especially damping, can reduce transmission error in geared transmissions. When bearing damping was increased from the low value typical of rolling-element bearings to the optimum value, transmission error reductions of 16 decibels were calculated for the model analyzed. This represents a significant reduction in noise generation. The optimum damping values are readily achievable with fluid film bearings or squeeze film dampers.

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