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THE ANALYSIS OF THE AUTOMOTIVE DIFFERENTIAL VIBRATIONS BY MEANS OF BIFURCATION DIAGRAMS

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Abstract: *This article is concerned with capabilities of bifurcation and chaos analysis methods used for analysis of the automotive differential operation and noise. The purpose of the article is to give an overview of practical use of the above mentioned methods. The potential advantages, disadvantages are discussed as well as expected contributions of application of the bifurcation analysis to solving the automotive differential journal bearing noise problem.*

Keywords: ***Automotive differential, sleeve bearing, chaos, bifurcation.***

1. Introduction

Fighting the noise emitted by a running vehicle into the environment and into the passenger area is a long-lasting problem. The development department of the Škoda Auto manufacturer has dealt with this phenomenon recently. The automotive differential was identified as one of the noise sources. The stub shaft is inserted into the differential case through sleeve journal bearing which requires some radial clearance to be operational. Therefore, there is a mechanical looseness in the journal bearing which in combination with vibrations of the engine may result in an undesirable noise emission. The VW Group attempted to eliminate the problem by adjusting the radial clearance of the bearing with a tapered centring ring and an axial compression spring, see Fig. 1. However, new kinds of undesirable noises of the differential were observed under a different set of operating conditions compared to the previous case. The noise phenomenon related to the sleeve bearing arrangement without compression spring was labeled as "wummern" (perceived as drumming noise) and the noise of the bearing with the compression spring was labeled as "schnarren" (perceived as rough chattering noise). Brno University of technology was introduced to the problem. Our main goal was to identify the causes of the noise and to develop a method for identification of phenomena types. Therefore, an analytic method capable of determining the type of noise phenomenon was created by Mazůrek et. al (2009). The method processed measured values of the stub shaft center displacement relative to the center of the bearing. Although the described method proved as functional in practical applications we were looking for other ways to improve it or to perform even deeper analysis of the bearing operation. Thus, the analysis of the bifurcation diagrams and chaotic behavior of the bearing was suggested. Myers (1984)

(a) (b)

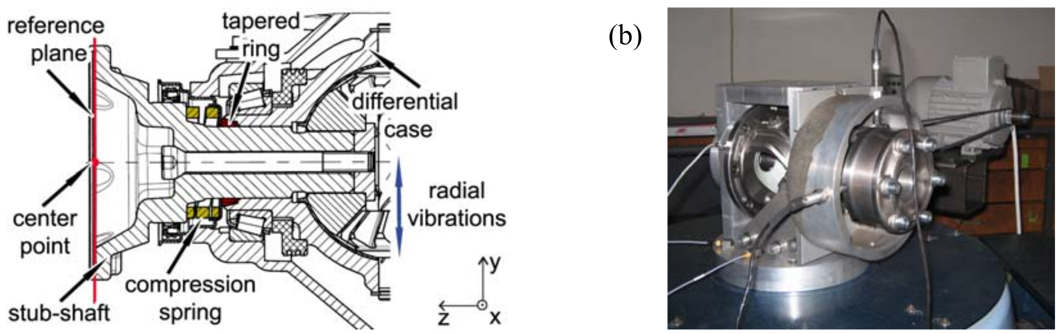

Fig. 1: a) Bearing arrangement with compression spring cut-away, b) Test rig with differential in experimental box.

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applied Hopf bifurcation analysis to journal bearing stability problem. Addiletta et. al (1996) used methods of chaos analysis for examination of the system stability for rigid rotor with sleeve journal bearings. Some other applications of chaos and bifurcation analysis in solutions to journal bearing and rotor stability problems were described by Muszynska and Goldman (1995), Wang and Wang (2004), Laha and Kakoty (2010) or Wang (2010). The main goals of this article are: overview of the knowledge about applications of the bifurcation and chaos analysis methods to the automotive differential sleeve bearing with radial clearance, discussion of the results, and potential contributions to the solution process. Results acquired by means of bifurcation analysis are compared with results of the stub shaft displacement power summation analysis described by Mazůrek (2009).

2. Examined bearing arrangements and operational conditions of the experiments

The bearing arrangement with adjustment of the radial clearance by tapered ring and compression spring (Fig. 1a) and an arrangement without the compression spring and tapered ring were experimentally examined.

The differential case was driven by controlled electromotor (see Fig. 1b) and was continuously slowed down from the speed of 3000 RPM to standstill. The speed of the differential case was proportional to the exciting vibrations which simulated vibrations of the vehicle engine. The exciting vibrations continuously decreased from 230 Hz to 10 Hz (corresponding to the maximum speed of 6900 RPM and the minimum speed of 300 RPM of the vehicle engine respectively, these values were also determined by technical parameters of the electrodynamic exciter). The amplitude of the exciting vibrations was 0.1 mm and the effect of decrease in excitation amplitude was observed with amplitude of 0.025 mm. The results of the experiments analyzed using the method of power summation of the stub shaft displacement are summarized in Tab. 1, where the greater value determines the type of noise emission.

Tab. 1: Parameters of the "wummern" and "schnarren" phenomena.

Design of the bearing	***wummern***	***schnarren***
without compression spring, excitation amplitude 0.1 mm	*0.474*	*0.357*
without compression spring, excitation amplitude 0.025 mm	*0.168*	*0.114*
with compression spring, exciting amplitude 0.1 mm	*0.222*	*0.553*
with compression spring, exciting amplitude 0.025 mm	*0.083*	*0.154*

3. Bifurcation diagrams

Bifurcation diagrams of the selected bearing designs with excitation amplitude of 0.1 mm and 0.025 mm are compared in Fig. 2. The frequency of the stub shaft center axis relative to displacement (in direction of exciting vibrations) passing through zero value as a function of the exciting vibrations frequency is depicted in the diagrams, the black line describes the ideal trajectory. The critical operating speeds are in the range from 2000 RPM to 6000 RPM. The difference between the bearing

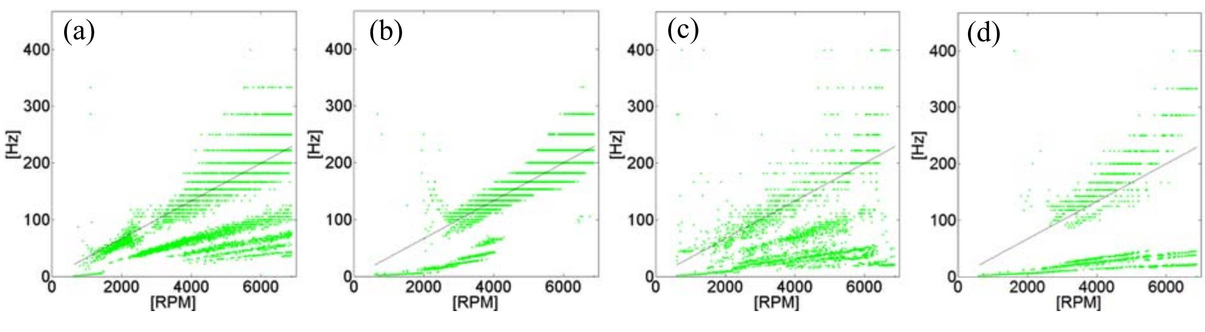

Fig. 2: The bifurcation diagrams of the bearing arrangement without compression spring (a), with the compression spring installed (b) both with exciting vibrations amplitude of 0.1 mm, and bearing arrangement without compression spring (c), with the compression spring installed (d) both with exciting vibrations amplitude of 0.025 mm.

design without compression spring and the one with the spring installed at the exciting vibrations amplitude of 0.1 mm is clearly visible from the comparison of the bifurcation diagrams, see Figs. 2a and 2b. There is a high amount of the bifurcations in the case of bearing without the spring. The sleeve bearing with radial clearance adjusted by the tapered ring and the compression spring has visibly lower bifurcation ratio and lower spread of points from the ideal trajectory of displacement frequency. Nonetheless, the behavior of stub shaft in the sleeve bearing is unstable in both of examined cases. It can, however, be claimed that the instability is somewhat lower in the case of bearing arrangement with compression spring. Interestingly, the resonance of the bearing at 4800 RPM has no significant effect on the bifurcation diagram pattern. The bifurcation diagrams of the stub shaft behavior loaded with 0.025 mm vibrations amplitude are shown in Figs. 2c and 2d. There are some significant changes in the pattern of the bifurcation diagrams, especially in the case of bearing design with compression spring. Observed differences between the diagrams arose most probably due to the reduced effect of exciting vibrations relatively to the effect of the differential case rotation and due to nonlinearity of the compression spring stiffness. From the results in Tab. 1 and bifurcation patterns in Fig. 2 it is possible to find that the higher amount of bifurcation (and spread of points from ideal trajectory) is characteristic for “wummern” type of noise. Lower amount of bifurcation (and spread of points from ideal trajectory) is characteristic for “schnarren” type of noise. Using subjective judgment it is possible to distinguish between the two observed types of noise. However, it would be more appropriate to find an objective criterion for this purpose. Therefore, an analysis of the phase trajectories was carried out.

4. Phase-trajectory diagrams

Analysis of dependence of the stub shaft center displacement in the direction of the exciting vibrations on the displacement speed was performed and phase-trajectory diagrams of the center point radial vibrations (see Fig. 1) were created for every single revolution of the differential case. From the phase trajectories in Figs. 4, 5, 6 and 7 it is obvious that the bearing subjected to combination of exciting vibrations and rotation of differential case is unstable in the whole range of observed operational speeds irrespective of the bearing arrangement. Therefore, mostly chaotic attractors were observed. Only the bearing arrangement with compression spring tends to stable behavior for speeds above 4000 RPM and exciting vibrations amplitude of 0.1 mm (see Fig. 6). Such behavior could be related to the resonance of the bearing assembly at 4800 RPM. It seems that the effect of displacement in direction of the exciting vibrations is strengthened by resonance against the effect of rotation and the system tends to be more stable. However, there are more steel-on-steel impacts which result in undesirable noise emission. The phase-trajectory diagrams analysis seems to be unsuitable for the objective of distinguishing between examined noise phenomena because only chaotic phase trajectories were observed. No significant differences are noticeable between phase trajectories of the bearing without the compression spring and the bearing with the spring. Therefore no explicit criterion can be expressed for differentiation between “wummern” and “schnarren” noises.

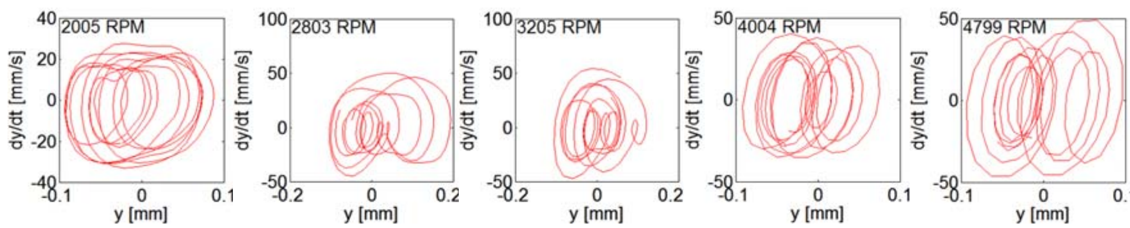

Fig. 4: Phase trajectories of the bearing arrangement without compression spring, amplitude 0.1 mm.

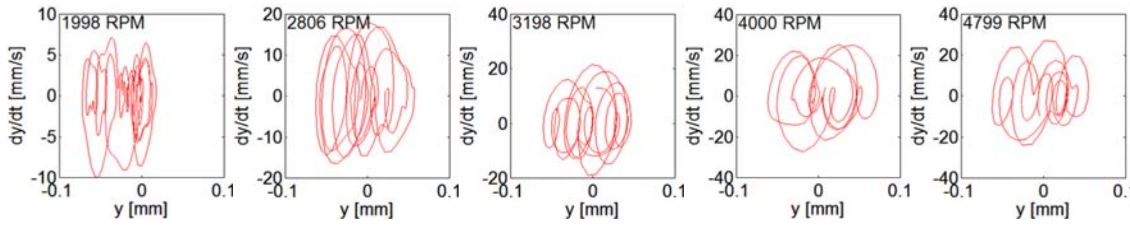

Fig. 5: Phase trajectories of the bearing arrangement without compression spring, amp. 0.025 mm.

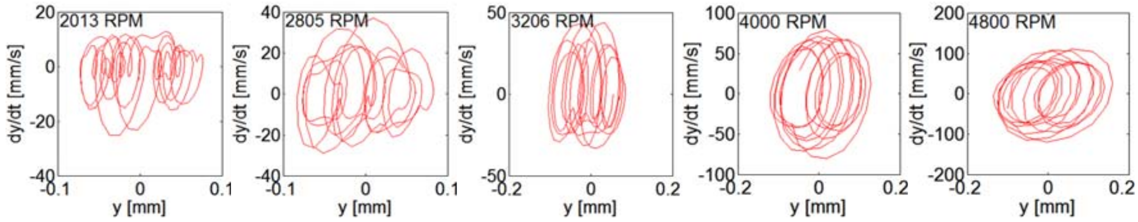

Fig. 6: Phase trajectories of the bearing arrangement with compression spring, amplitude 0.1 mm.

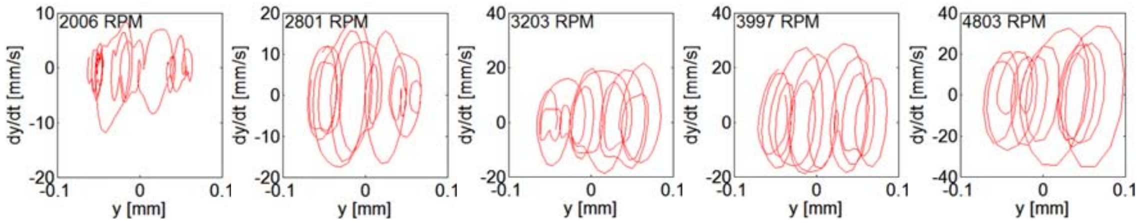

Fig. 7: Phase trajectories of the bearing arrangement with compression spring, amplitude 0.025 mm.

5. Conclusions

From the results of the bifurcation and phase diagrams analyses it is obvious that these methods can be used in combination with previously developed power summation analysis, mostly for better understanding of the bearing behavior. Generally, it is possible to subjectively differentiate the "wummern" and "schnarren" phenomena based on the bifurcation diagrams analysis (see Fig. 2), although no explicit criterion can be derived. The analysis of the phase-trajectory diagrams was carried out but the results were unsatisfactory. Neither objective nor subjective way of distinguishing between observed noises could be found from achieved results. Possibly, an estimate of the attractor dimension could be considered as an objective parameter if one assumes existence of explicit differences between attractor dimension values of the "schnarren" and "wummern" phenomena. Therefore, combined analysis using chaos and bifurcation methods could provide a lot of information about bearing behavior under various operating conditions. However, benefits of such complex data for design engineers in practical applications are questionable. Nonetheless, the combination of bifurcation and chaos analysis methods could be either valuable supplement of the stub shaft displacement power summation method or an alternative standalone analysis method for sleeve bearing behavior and noise assessment.

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