NOISE REDUCTION OF THE DIFFERENTIAL SLEEVE BEARING

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This article describes a technique for noise reduction of the differential bearings. Noise is excited by the mechanical looseness of the system and by the vibrations of the engine. The mathematical model enables the optimization of the radial stiffness of the differential bearings. A test rig has been designed for the simulation of the phenomenon outside the vehicle. Analytical software has been created that is capable of classifying the types of noise from the measured data. The optimal solution is tested on the rig at the end, thus proving that the technique works.

Keywords: *rolling bearing, vibration, radial stiffness, virtual model*

1. Introduction

The Institute of Machine and Mechanical Design has recently co-operated with the development department of a world car manufacturer on a problem of noise emission. The phenomenon was related to the irregular chattering noise which spread from the front axle. The source of the noise in the nearest joint of the half axle in the differential was found using the noise source location method. The noise spread along an acoustic bridge to a large brake disk which spreads the noise. However, the real mechanical source of the problem is the sleeve bearing which supports the half axle. The complex mechanism can only work with accurately set mounting clearances. On the other hand, these clearances could increase the noise level of the sleeve bearing and the connected parts due to possible metal-to-metal collision. This phenomenon is widely discussed in papers [1, 2]. The application to the car differential is solved in detail in [3].

2. Problem definition

The differential gear (Fig. 1) works under vibration load due to an insufficiently balanced engine which is flexibly mounted onto the chassis. Some car manufacturers therefore redesign the assembly to prevent the chattering noise.

These modifications are mainly based on setting clearance limits by using flexible bindings (Fig. 2). This type of bearing modification was used in some types of passenger cars. Thus the problem with noise emission was noticed. Noise always occurs at certain engine rotational speeds independently of the speed of the car (rotational speed of the half axle) – therefore it was audible even in a non-moving vehicle. This noise typically occurs when driving with unloaded transmission unit. In contrast, the engines equipped with balancing shafts were never noticed to have such a noise. Therefore it is evident that the noise must be excited by the engine vibrations and inappropriately balanced shafts.

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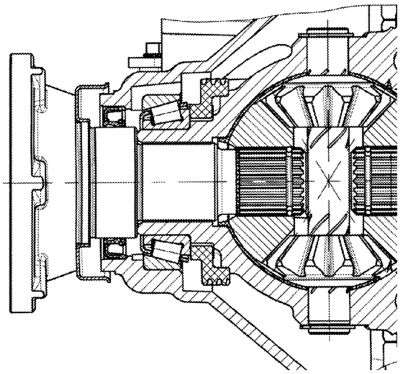

Fig.1: Cut-away of the bearing of the stub-shaft in the common differential

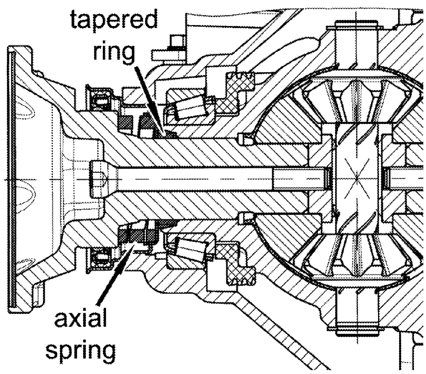

Fig.2: The bearing arrangement with axial spring and tapered ring

Motion of the stub-shaft in plane perpendicular to the stub-shaft centerline was monitored in order to evaluate the noise generated by the differential assembly. The plane of the stub-shaft to half-axle attachment was chosen as a nominal plane. The so-called orbits, i.e. the trajectories of the shaft centerline motion per revolution, are recorded. Figure 3 shows the measured relation of the orbit shape when driving at various engine rotational speeds (in RPM). A resonant-type noise is typical for differential assemblies with flexible bindings. Noise always occurs at the same engine rotational speed.

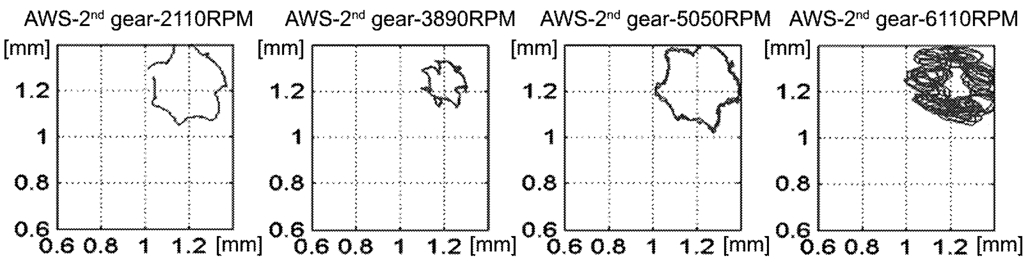

Fig.3: Motion of the stub-shaft with axial spring (anti-wummern) at various engine rotational speeds

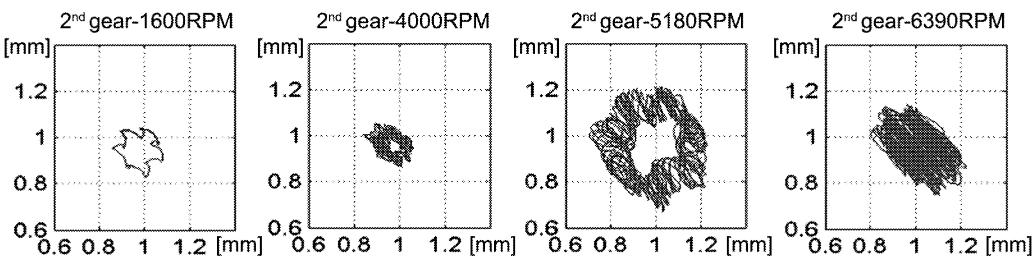

Fig.4: Motion of the stub-shaft without axial spring at various engine rotational speeds

Owing to the above-mentioned facts we presume that the differential gear was suffering from resonant problems. Therefore, the measurement was also carried out without the components used for flexible clearance limit settings in the assembly. Similar measurements are presented in the following figures (Fig. 4). Interestingly, the behaviour of the assembly in the configuration where the suspicious flexible binding has been removed is apparently resonant, too. Force, which is proportional to the square of engine rotational speed, has a fundamental effect on the noise intensity According to a subjective evaluation, the type of noise spectrum was completely different, and it occurred at different rotational speeds.

3. Mathematical model

It is obvious that the testing of new designs using a trial-and-error method would take a very long time. Creation of a virtual prototype is advantageous because it makes testing of the variants significantly more productive, and particularly, it allows a system analysis. Thus an optimal arrangement can be found very quickly. Selection of the area to be simulated is fundamental in terms of the number of simulated assembly elements and the required level of accordance between the model and reality. The pursuit of bringing as many physical phenomena as possible into the parametrical model often results in the devaluation of that model. If there are too many parameters, they cannot be properly measured, and that is why they are often estimated. There is a schematic sketch of the modeled 'differential – half-axle – wheel' driving assembly in figure 5.

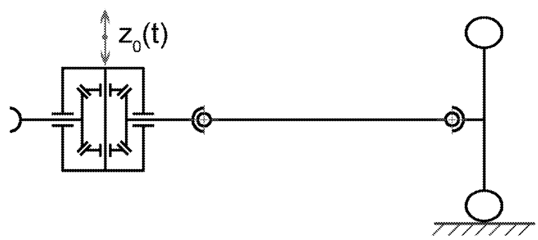

Fig.5: Schematic sketch of the driving assembly

To design the lumped parameter model the rigid bodies of the stub-shaft and half-axle were divided into three concentrated masses bound by so-called 'mass constraints' (Fig. 6). The kinetically activated model with five DOF (degrees of freedom) only operates with displacements in a z-axis direction which is its main advantage.

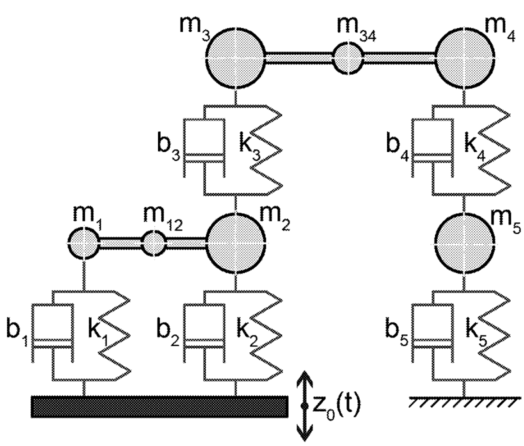

Fig.6: Linear mechanical model with five DOF

Kinetic, dissipative and potential energy of the vibrating system are defined by common equations, which are substituted into the LagrangeŽs equations of the second class:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial E_\mathrm{K}}{\partial \dot{z}_j}\right) - \frac{\partial E_\mathrm{K}}{\partial z_j} + \frac{\partial E_\mathrm{D}}{\partial \dot{z}_j} + \frac{\partial E_\mathrm{P}}{\partial z_j} = 0 \;, \qquad j = 1,\ldots,5 \;. \tag{1}$$

After adjustment and conversion into matrix form we obtain the following equation:

$$\mathbf{M}\,\ddot{\mathbf{z}} + \mathbf{B}\,\dot{\mathbf{z}} + \mathbf{K}\,\mathbf{z} = \mathbf{B}_0\,\dot{\mathbf{z}}_0(t) + \mathbf{K}_0\,\mathbf{z}_0(t) \;, \tag{2}$$

where for single matrices the following hold:

$$\mathbf{M} = \begin{bmatrix} m_1 + \frac{m_{12}}{4} & \frac{m_{12}}{4} & 0 & 0 & 0 \\ \frac{m_{12}}{4} & m_2 + \frac{m_{12}}{4} & 0 & 0 & 0 \\ 0 & 0 & m_3 + \frac{m_{34}}{4} & \frac{m_{34}}{4} & 0 \\ 0 & 0 & \frac{m_{34}}{4} & m_4 + \frac{m_{34}}{4} & 0 \\ 0 & 0 & 0 & 0 & m_5 \end{bmatrix} ,$$

$$\mathbf{B} = \begin{bmatrix} b_1 & 0 & 0 & 0 & 0 \\ 0 & b_2 + b_3 & -b_3 & 0 & 0 \\ 0 & -b_3 & b_3 & 0 & 0 \\ 0 & 0 & 0 & b_4 & -b_4 \\ 0 & 0 & 0 & -b_4 & b_4 + b_5 \end{bmatrix} , \qquad \mathbf{B}_0 = \begin{bmatrix} b_1 \\ b_2 \\ 0 \\ 0 \\ 0 \end{bmatrix} ,$$

$$\mathbf{K} = \begin{bmatrix} k_1 & 0 & 0 & 0 & 0 \\ 0 & k_2 + b_3 & -k_3 & 0 & 0 \\ 0 & -k_3 & k_3 & 0 & 0 \\ 0 & 0 & 0 & k_4 & -k_4 \\ 0 & 0 & 0 & -k_4 & k_4 + k_5 \end{bmatrix} , \qquad \mathbf{K}_0 = \begin{bmatrix} k_1 \\ k_2 \\ 0 \\ 0 \\ 0 \end{bmatrix} .$$

This system of linear differential equations could have a relatively simple solution. However, the real system is significantly non-linear: elastic constraints have relatively large clearances and the whole system is damped mainly using sliding friction. Therefore there could be problems with the numerical solution in terms of required integration step. Thus high sampling frequencies are used, even though the solution could be unstable. In addition, the identification of a large number of system parameters is almost impossible. The individual joints have a wide range of clearances, stiffness and friction. However this range generally has no influence on noise.

Therefore the proposed model was considerably simplified. Simulated responses of the simplified models were always compared with the real signals from the test rig. The non-linear radial stiffness $k_2 = k_2(z)$ proved itself as a determining parameter. The model was reduced to a kinetically activated system with one DOF (Fig. 7) after numerous measurements. The damping was very low and it remained linear (viscous).

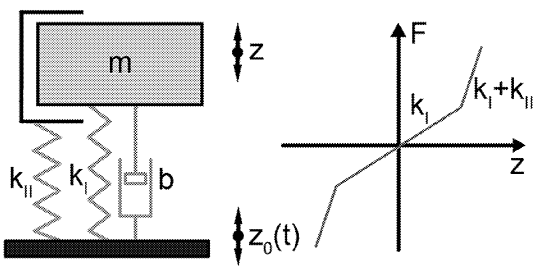

Fig. 7: Nonlinear mechanical model with one DOF and approximation of the nonlinear stiffness

Non-linear radial stiffness is a problem when simulatiing real bearing configurations. For this reason it cannot be simulated with a single parameter, therefore the function $F_\mathrm{k}(z)$ has to be applied. This function describes the force acting on the bearing flexible binding depending on the eccentricity of the shaft. Dynamic behavior of the kinematically excited nonlinear system (Fig. 7) is described by the following equation:

$$m\,(\ddot{z}+g)+b\,(\dot{z}-\dot{z}_0(t))+F_\mathrm{k}(z-z_0(t))=0\ . \tag{3}$$

Radial stiffness $F_\mathrm{k}(z)$ in the reference plane is the easily identifiable parameter [4]. The stiffness of each tested variant was measured after the differential was assembled in an experimental testing rig (Fig. 8). For the purposes of the simulation, the real measured stiffness was substituted by a kinked line (Fig. 7). Angular coefficients of both lines and the position of the intersections are the most important parameters of the simplified kinked-line stiffness. The lowest slope area of the simplified stiffness k_I is determined by the stiffness of the flexible element (spring, polygonal insert, rubber element). The steeper line $k_\mathrm{I}+k_\mathrm{II}$ represents either sudden or smooth fit of the sprung element in the limiting position.

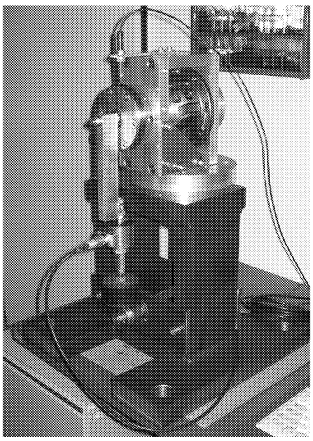

Fig.8: The test rig for measurement of bearing radial stiffness

4. Description of the experiment

Driving tests are often slightly subjective and difficult. Moreover, monitored noise is very low in comparison to a relatively high level of noise from the background in the case of the differential gear. In order to take further measurements, the whole monitored assembly was removed from the vehicle. A test rig for the problematic assembly was created. Vertical vibrations of the combustion engine were simulated by a laboratory electrodynamic vibration exciter. The differential gear was placed into a simple dural housing with one completely assembled side of the differential assembly including side gear and differential pinions. The connected parts of the half-axle were substituted by a cylindrical element. The calculated reduced mass of the element was 2.3 kg. There is a timing belt and an electromotor placed outside the vibration ramp (Fig. 9) to ensure controlled rotation of the differential. Stub-shaft motion in both axes was measured by two non-contact displacement sensors of the Schenck Vibronecs IN-081 type with a resolution of 0.001 mm.

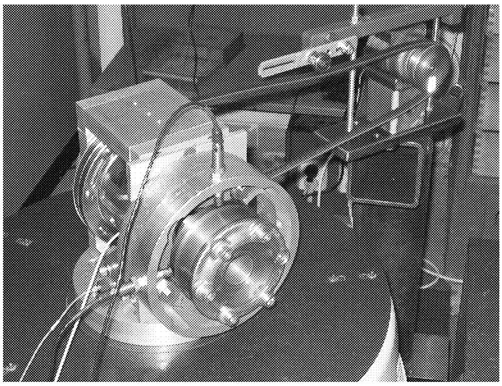

Fig.9: The test rig

The sweep test serves as a basis to describe the relation between the emitted noise and the engine rotational speed. Although the illustration of orbit shapes at various rotational speeds is visually clear, a much simpler parameter is needed for a larger number of experiments to describe the vibration responses and to allow the distinguishing of various types of vibrations. By recording the RMS (root mean square) value of vibrations in the direction of the engine cylinder axis it should be sufficient to obtain a vibration time response and evaluation of the relation between engine rotational speed and level of the differential gear vibrations (Fig. 10).

```
for P = 1:number_orbit
  Xrms(P)=sqrt(std(X(start_o:finish_o))^2+mean(X(start_o:finish_o))^2);
  Yrms(P)=sqrt(std(Y(start_o:finish_o))^2+mean(Y(start_o:finish_o))^2);
End
```

Fig.10: Calculation of the root mean square (RMS) values of vibration in individual orbits (MATLAB)

At this point it has become necessary to introduce an evaluation method, as well as a criterion determining the dominant noise produced by the differential gear. To achieve this, a tracking analysis at engine rotational speed according to gear ratio multiples was used. Furthermore the tracking analysis of the middle band of the excitation frequency is especially important. The time response of the analyzed signal then describes the response at excitation frequency ('forced'), and it is typical of inappropriately tuned modal systems without clearance. The time response of the signal at the centre of the band pass filter at the rotational frequency of the differential gear is also important for the correction of results. The signal filtered in this manner is useful mainly for the evaluation of radial run-out of the stub-shaft that appears after mounting. However, run-out has no influence on the noise emission of the whole differential assembly. Then the timings of the signals mentioned above are subtracted from the time response of the complete measured vibration signal. The remaining information represents vibrations due to unclear causes. It proved that this information describes the intensity of the chaotic differential gear vibrations ('drumming') fairly precisely. In order to describe the behaviour of the differential assembly with a single value, the power summation method was applied to the given filtered signals obtained from the whole sweep test. Consequently, a program was created for the analysis and evaluation of the above-mentioned measured data. The program was developed using a Matlab 2007b environment and it is able to process data files obtained from a Dewetron DEWE 2010

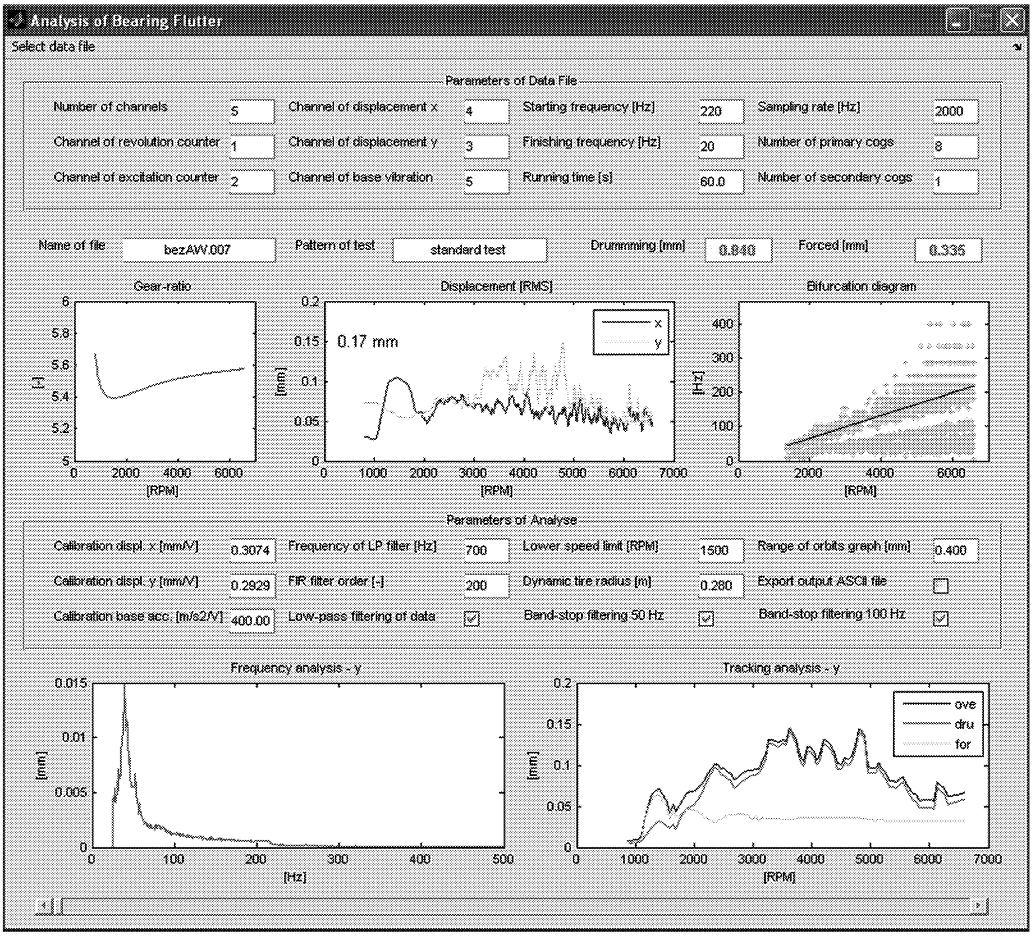

Fig.11: Interface of the analytical program in a Matlab environment

analyzer. Figure 11 shows the program interface. It represents the results (both graphical and numerical) and enables the setting of analysis parameters.

5. Results and discussion

The existing bearing design shows the resonant effect that occurs in a range of operational speeds. Thus, the node needs to be 'retuned'. One way this can be achieved is by increased stiffness. For example in the case of the experimental replacement of the spring (Fig. 2) by a steel ring, the natural frequency of the assembly is above the upper limit of the operational speed. Therefore the differential gear causes some noise problems, both in the virtual model and by subjectively assessed driving tests. Unfortunately, this improvement can only be used with actual driving which reduces its practical importance to zero. Another possible way to eliminate the noise problem is to decrease stiffness. Thus the natural frequency of the differential gear is below operational engine rotational speed. However, if the basic stiffness of the node is too low, the clearance in the bearing has to be sufficient to prevent its 'impact delimitation' during the resonance of the node. For example, a stiffness that corresponds to the shaft seal stiffness requires a clearance of about 3 mm which is unacceptable for the proper functioning of the seal. Therefore, it is important to find a ‘gap’ with suitable and usable stiffness which enables proper operation of the shaft seal. Such a task is perfectly suited to the virtual model of the node.

Simulated time responses (amplitude – frequency) for three kinds of stiffness (as a function of displacement) are compared in Fig. 12. The time response represents the amplitude of vibration proportionally to engine rotational speed. The hardest version of differential gear assembly ($k_{\mathrm{I}} = 3900\,\mathrm{N/mm}$) is the mass produced one with a helical spring. The standard version with the flexible binding removed is the softest one ($k_{\mathrm{I}} = 32\,\mathrm{N/mm}$). The middle version ($k_{\mathrm{I}} = 205\,\mathrm{N/mm}$) chosen from the analyzed range (150 to 400 N/mm) seems to be optimal.

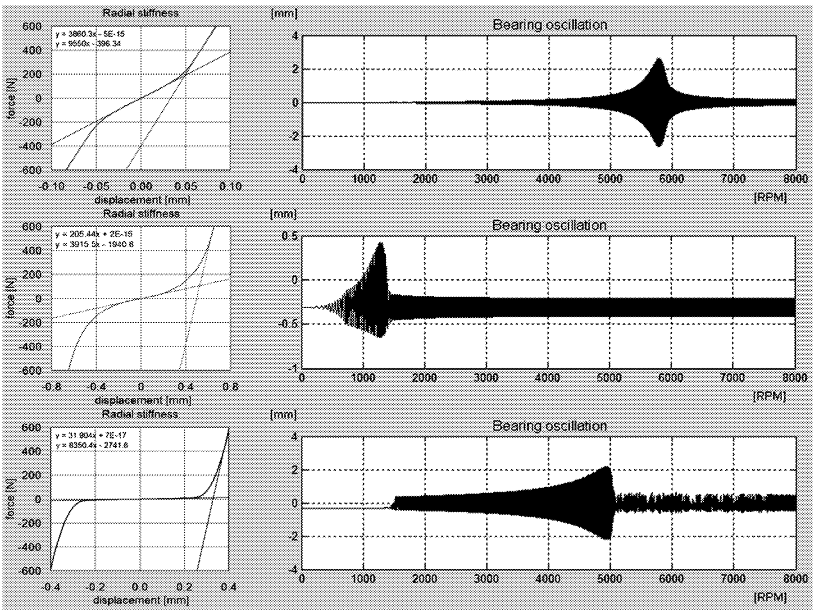

Fig.12: Simulated time responses of the virtual model with various kinds of (measured) stiffness

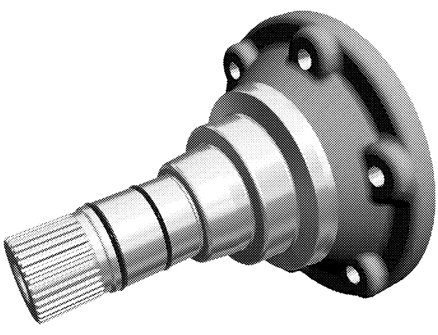

Fig.13: The provisional design modification of the stub-shaft

Hence we have performed a provisional realization based on this analysis [5]. A simple design improvement of the node was completed by adding two O-rings into the grooves on the stub-shaft (Fig. 13).

The results of the measurements which were carried out with the real design arrangements corresponding to previous virtual models (Fig. 12) are shown in Fig. 14 for comparison purposes. The best description of the intensity of impacts is through the summary values of the 'drumming' and 'forced' oscillation types. These values are 0.22/0.56 mm for the standard design with a spring, and 0.86/0.31 mm for the version with the spring being removed. Thanks to the optimized solution the value of vibrations has significantly dropped down and reached a value of 0.15/0.15 mm. Equal results were obtained in subjective assessment of the noise emitted by the node in the test rig.

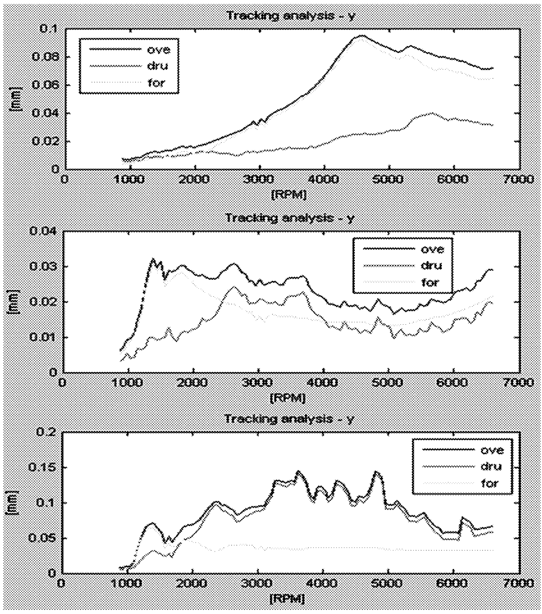

Fig.14: Amplitude-frequency plotting (ove – overall, dru – drumming, for – forced)

6. Conclusions

The outlined approach to the solution of undesirable noise problem in the differential bearing enables to optimize a relatively simple construction solution. The proposed arrangement of the differential gear is only the first step. Obviously there must be further tuning in the future. This paper deals with the analysis of the required stiffness and the ways to reach it by suitable arrangement of the bearing in the differential gear assembly. We presume that by optimizing the mounting overlap it will enable the use of the special roller polygonal bearings with the required stiffness. The quality of the roller bearings will provide the necessary reserve in the bearing lifetime for this particular solution.

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