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A Novel Test Procedure to Determine the Behaviour of A Center Bearing of A Multi-Piece Driveshaft under Real Road Load Conditions

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Abstract: Driveshafts are one of the most important members of the drivetrain that transmits the power and rotational movement from the engine to the rear axles. In light duty commercial vehicles, the multi- piece driveshafts have elastomeric parts called center bearing. The driveshaft center bearing usually consist of a rolling element bearing isolated in rubber and with a bracket configuration for attaching it to the vehicle chassis structure. The duty of the center bearing of the multi-piece driveshafts on the vehicle is the vibration damping function thanks to the elastomeric structure. Center bearing is an important sub-component of the multi-piece driveshaft which determines frequency response characteristic of driveshaft. The manner in which it is attached to the vehicle, whether directly to the under body or to the chassis frame, it can have a considerable effect on the center bearing's capability of providing optimum isolation and reducing the transmission of disturbances to an acceptable level in a given application. Due to both real road conditions and operation time, the elastomeric structure and the rolling ball bearings are damaged in time and the level of vibration and noise can cause disturbance on the vehicle. In this study, a test systematic has been developed to determine the change in noise characteristics of center bearing sub-assembly during the real time operation of the driveshaft. The noise characteristics of center bearing sub-assembly before and after the tests were compared and investigated.

Keywords: Driveshaft, Novel test, Drivetrain

Introduction

Vibration and noise, or NVH is the sound and vibration characteristics of ground vehicles. Sound and vibration can be measured, but noise can only be evaluated subjectively. All machines with moving parts produce dynamic forces under normal operating conditions. The mechanical behavior of the machine may change due to wear, changes in operating conditions and load changes. Understanding the dynamics of the machine and how the forces create vibration is the most important key to understand the sources of vibration.

The NVH of driveshaft is a type of NVH that causes the transmission of sound and vibration from the drivetrain system to the customer in a wide range of conditions. An error that may occur during the detection of the formation of the driveshaft NVH can create customer dissatisfaction and can lead to costly downtime and can cause fatal failures. Vibration in the automotive sector is undesirable. Because vibration means wasting energy and unwanted noise. For example, the vibration movement of the motor, vibration in electric motors or vibrations that occur in a mechanical device fall into the unwanted vibration group. Such vibrations may be caused by unbalanced rotating objects, irregular friction, collision of corresponding gears in the gearbox and other reasons. Sound and vibration are very close concepts. Sound, pressure waves are composed of vibrating structures, pressure waves are the vibration of structures. Therefore, we try to reduce the vibration level while simultaneously reducing the noise level.

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When the rotational speed of the driveshaft is gradually increased, bending occurs in the center of the driveshaft tube and can gradually increase in deflection and reach an amplitude that causes the shaft to break at the critical speed of driveshaft. In order not to reach the critical speed limit during real time operation, driveshafts are designed as multi-pieces (Figure 1).

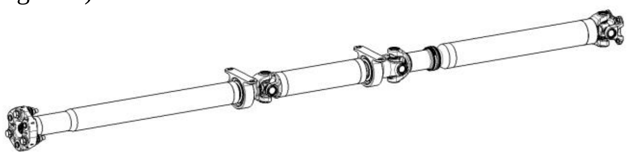

Figure 1. Multi-Piece driveshaft

In its simplest form which is depicted in Figure 2, center bearing sub-assembly consists of a rolling ball bearing in the center, a damping elastomer element around the roller ball bearing outer ring and the bracket which is attached to the chassis frame. The center bearing of driveshaft is designed to absorb vibration and noise generated by the driveshaft.

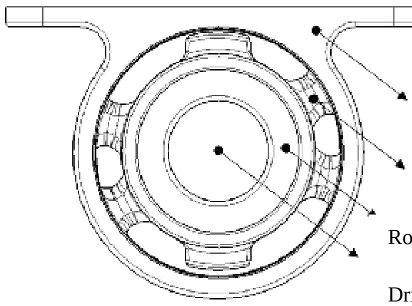

Figure 2. Driveshaft center bearing

In the book "Universal joint and driveshaft design manual "[1], noise and vibration from the propeller shaft are examined and general recommendations are given for driveshaft suspension characteristics. In the publication "NVH phenomena in light truck drivelines" [2], the causes and effects of cardan shaft noise and vibration known for the drivetrain of 4x4 vehicles are examined. In the publication "Aspects of Driveline Integration for Optimized Vehicle NVH" [3], the issues that cause noise and vibration in the drivetrain are combined with case studies and analyzes and the integration of the systems is examined. In the publication "Design and Analysis of Composite Drive Shaft" [5], the authors conducted bending mode analyzes for the propeller shaft with composite material using the finite element method. In the publication "Evaluation Method for Vibratory Forces Caused by Propeller Shaft," [8], the vibration forces that cause noise from the propeller shaft are examined. In multi flexible multibody simulation of automotive systems with non-modal model reduction techniques"[9], vehicle body simulation was performed and the effect of rigidity on vibration was investigated. In this study, a test systematic has been developed to determine the change in noise characteristics of center bearing sub-assembly during the real time operation of the driveshaft. The noise characteristics of center bearing sub-assembly before and after the tests were compared and investigated.

Theory

Sound generation or transmission in vehicles are produced in two ways, which are the result of sound waves propagating through the air (airborne sound) and generated as a result of progressive waves within the structure. The main purpose of the NVH tests on vehicles is to detect these sounds and bring them to optimal levels in terms of driver comfort. The sound pressure level is an expression of the measured sound pressure in dB, expressed in equation (1). The reference sound pressure in the equation is,

$$\mathrm{SPL} = \mathrm{L_p} = 10\log\left(\frac{p}{p_{ref}}\right)^2 = 20\log\left(\frac{p}{p_{ref}}\right)\ (\mathrm{dB})$$

Octave-Band Frequency Analysis

In order to gain an idea of the frequency distribution of the noise, the frequency scale is subdivided and the sound power per frequency range of each share is analyzed. Each portion of the frequency scale is called a band. For filtering covering the entire frequency spectrum, frequency ranges called octave bands are used. In an octave band, the upper value of the band is twice the lower value, and the upper value of each band is the lower value of the next band. The central frequency of each band is the geometric mean of the upper and lower boundaries. The band used for detailed frequency analysis is 1/3 octave band. The ratio between the maximum and minimum values of the 1/3 octave band = 1.26. The 1/3 octave band widths are obtained by dividing the 1/1 octave band widths into three. Mathematically, 1 (one) octave band is expressed by equation (2).

$$f_2 = 2f_1$$
$$f_0 = \sqrt{2}f_1 = f_2/\sqrt{2} \qquad (2)$$

In this study, for vibro-acoustic analysis, 1/1 or 1/3 octave bands are used. The center frequencies of the 1/1 and 1/3 octave bands are given in Table 1 [11].

Table 1. Octave band scale

Lower Band Limit (Hz)	1/3 Ocave Band Center Frequency (Hz)	Upper Band Limit (Hz)
(Hz)	(Hz)	(Hz)
14,1	16	17,8
17,8	20	22,4
22,4	25	28,2
28,2	31,5	35,5
35,5	40	44,7
44,7	50	56,2
56,2	63	70,8
70,8	80	89,1
89,1	100	112
112	125	141
141	160	178
178	200	224
224	250	282
282	315	355
355	400	447
447	500	562
562	630	708
708	800	891
891	1000	1122
1122	1250	1413
1413	1600	1778
1778	2000	2239
2239	2500	2818
2818	3150	3548
3548	4000	4467
4467	5000	5623
5623	6300	7079
7079	8000	8913
8913	10000	11220
11220	12500	14130
14130	16000	17780
17780	20000	22390

Structure Borne Noise

It occurs when mechanical vibration energy is generated by a product and circulates through the structure of the product and the materials it comes into contact with. One solution to reduce structure borne noise is to prevent vibration from friction. This can be done by placing a low friction surface between mutually moving parts of a product. Another solution is to change the vibration mode of the structure by using damping material to ensure that the vibration is distributed. In this case, the structure will not vibrate as easily as before, and the vibration energy from the structure will be limited and converted into heat energy. A third solution is to isolate the vibration to prevent it from moving from one part of the product to another or to adjacent structures. Structure-borne noise is transmitted through solid structures such as steel, wood, concrete, stone.

Airborne Noise

Airborne noise arises from the interaction of a vibrating surface with the surrounding air. To reduce airborne noise; absorbers, barriers or foam can be used. Airborne sound absorbers are typically foam or fiber materials that absorb and convert sound energy into heat energy. Sound waves expand outward from the source and propagate in all directions. The difference between the structure and the noise in the air lies in the transmission medium. The sound coming from the air occurs in the progressive motion (vibrations) of the mass particles and the speed of sound is transmitted in the form of sound waves.

Test Method

Test bench at Figure 3 can excite the center bearing assembly in radial and axial directions and it can also rotate the rolling ball bearing at the same time. The interior walls of the test bench chamber are acoustically conditioned. The tests were carried out at two different temperature values. For each temperature, at the same axial and radial excitation, noise levels of the center bearing for three different speed levels were calculated. First, the ambient was cooled down to 20̊ C, afterwards the noise level of the center bearing assembly was calculated at 5 mm axial and radial excitation and the noise level was calculated. Then, the noise level was calculated at the same axial and radial excitation at 20̊ C chamber temperature and 3.000 rpm rotational speed. The same process was repeated under the same conditions but at 6500 rpm rotational speed.

The medium was allowed to warm up to 80 degrees and the operations performed at 20 ºC were repeated and the first block was completed. The block cycle was continued until the center bearing elastomer failed. As a result, the noise data calculated in the first block and the noise data calculated in the block where the failure was detected were compared.

Figure 3. Test Bench

Results and Discussion

Noise data were obtained as a result of the above described test sequences. The octave band averages of the obtained results were taken. The noise levels in the first block and the block in which the failure (last block) was observed were compared under the specified test conditions. In Figure 4, the noise levels were given when the center bearing elastomer was axially and radially moved at 20̊ C. In Figure 5 the center bearing elastomer was axially and radially moved at 20̊ C and roller bearing was rotated at 3.000 rpm. In Figure 6, the center bearing elastomer was axially and radially moved at 20̊ C and roller bearing was rotated at 6.500 rpm. In Figure 7, the noise levels were given when the center bearing elastomer was axially and radially moved at 80̊ C. In Figure 8, the center bearing elastomer was axially and radially moved at 80̊ C and roller bearing was rotated at 3.000 rpm. In Figure 9, the center bearing elastomer was axially and radially moved at 80̊ C and roller bearing was rotated at 6.500 rpm.

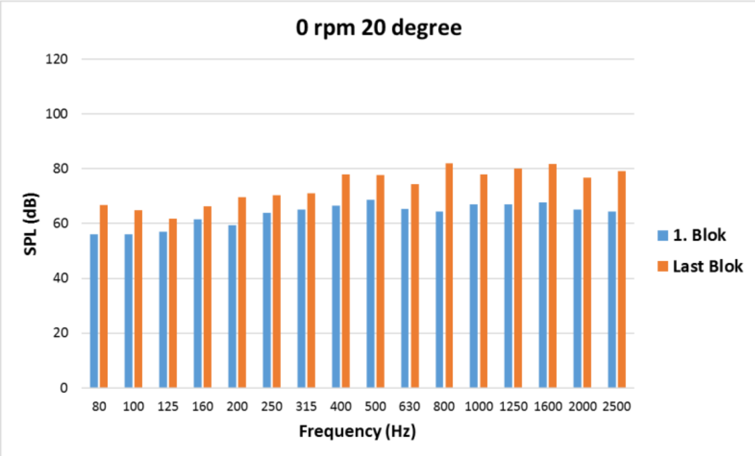

Figure 4. Sound Pressure Levels at 20ºC and 0 rpm

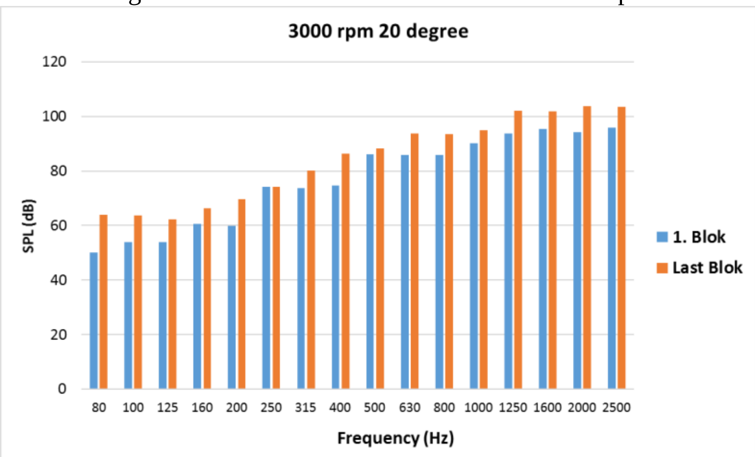

Figure 5. Sound Pressure Levels at 20ºC and 3.000 rpm

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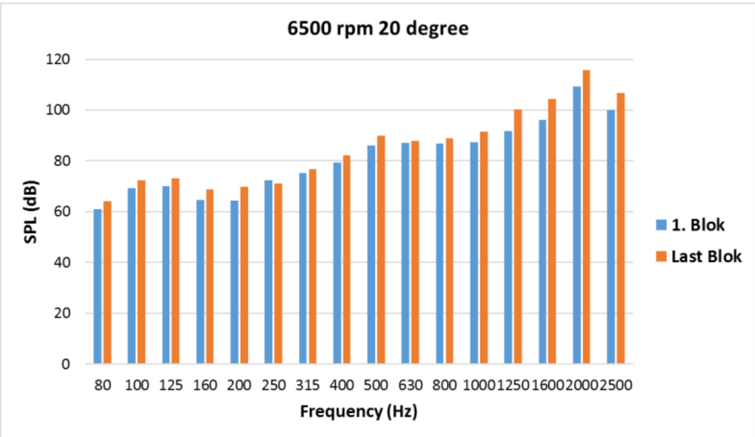

Figure 6. Sound Pressure Levels at 20ºC and 6.500 rpm

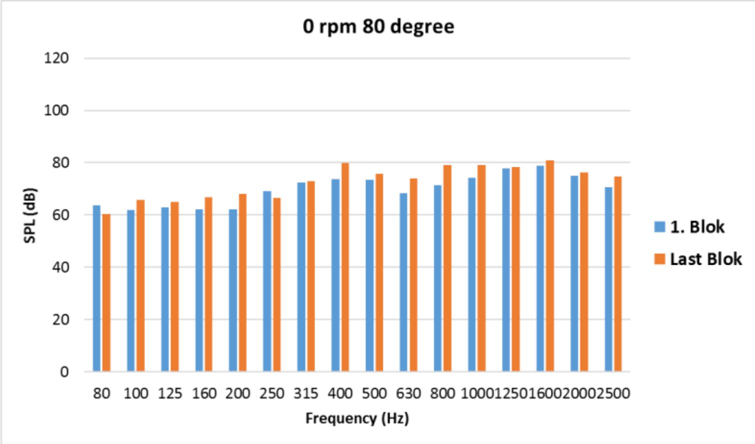

Figure 7. Sound Pressure Levels at 80 ºC and 0 rpm

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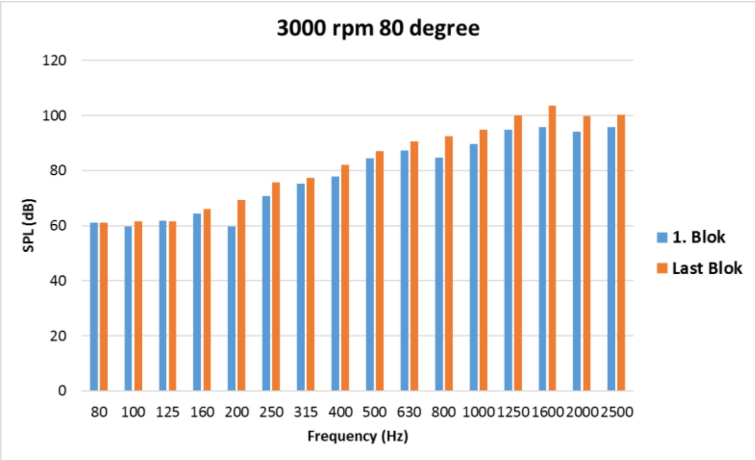

Figure 8. Sound Pressure Levels at 80ºC and 3.000 rpm

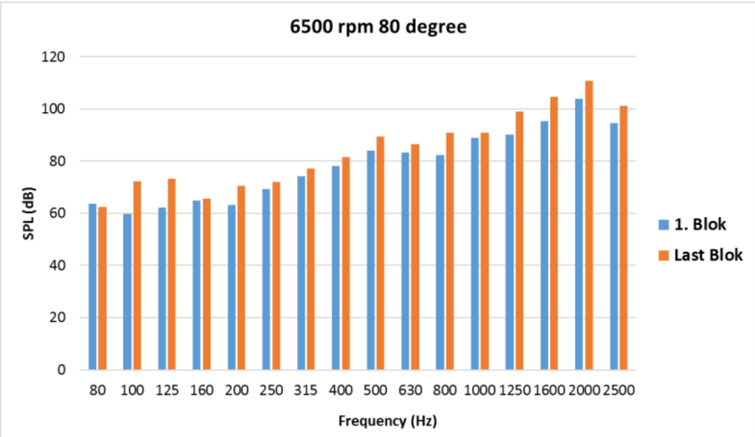

Figure 9. Sound Pressure Levels at 80ºC and 6.500 rpm

Conclusion

Table 2 shows that the equivalent noise levels for all test conditions. Results are as follows:

1. For both temperature levels, the noise level increases as the rotational speed of the roller bearings increase.
2. The comparison of noise levels in the 1st block and last block of the test at the stationary position of the roller bearing, shows that sound pressure level increases with cycles.
3. The noise level in the first block at 3000 rpm 80 degrees Celsius is 4 % lower than the noise level in the last block.

4. The noise level in the first block at 6500 rpm 80 degrees Celsius is 5 % lower than the noise level in the last block.

Table 2. Sound Pressure Levels in Test Conditions

Temp (C)	Velocity (rpm)	1. Blok SPL (dB)	Last Blok SPL (dB)
	0	85	88
20	3000	100	104
	6500	103	106
	0	87	89
80	3000	101	106
	6500	102	111

Recommendations

This study is for light duty driveshafts only, the work can be extended to all driveshaft product families in future work. In addition destructive inspection of roller bearings will be carried out to identify the root cause of the noise level increase.

Acknowledgements or Notes

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