SSRG International Journal of Mechanical Engineering
ISSN: 2348-8360/ https://doi.org/10.14445/23488360/IJME-V12I6P102

Volume 12 Issue 6, 15-20, June 2025

Original Article

Some Observations on Premature Damage to Automotive Wheel Hub Bearings

S D Rasika Perera

Department of Mechanical Engineering, The Open University of Sri Lanka, Nawala, Nugegoda, Sri Lanka.

Corresponding Author : sdper@ou.ac.lk

Received: 03 April 2025 Revised: 04 May 2025 Accepted: 05 June 2025 Published: 30 June 2025

Abstract *- A wheel hub bearing could suffer damage from impacts during operation. The speed at which the vehicle strikes the pothole plays a critical role in determining the extent of the damage; higher speeds typically result in greater force and, consequently, more severe impacts on the axle and other components. The angle of attack is another parameter that determines the impact force on the axle bearing. Optical imaging revealed that at some operating point, the entire load of the axle was absorbed by a single ball. Driving over a hump or a pothole of 160 mm or above with an angle of attack above 70 degrees could be detrimental to the axle bearing of an automobile. Conventional bearing fatigue life calculations cannot accurately predict the time or number of load cycles needed to initiate a defect under rolling contact fatigue. The current research paper aims to analyze the premature failure of an automotive rear hub bearing due to false brinelling.*

Keywords *- Brinelling, Bearings, Fretting, Rolling element bearings, Rolling contact fatigue, Spall propagation.*

1. Introduction

Automobile wheels are mounted on wheel hub bearings, allowing the wheel to rotate freely about an axle while carrying the vehicle's weight [1]. Bearings are selected based on the loads acting on the bearing to last an expected lifetime for a particular application [2]. The standard bearing design specifications are categorized as angular, axial, or radial [3]. Bearings can fail for many reasons: "overloading, excessive load due to improper handling, poor shaft or housing accuracy, installation error, ingress of foreign objects, poor lubrication, excessive impacts or rusting [4]".

Further, Upadhyay et al (2013) [3] have noted that premature damage to the bearing may cause conditions such as driving at high speed, excessive loading, and extreme temperature conditions. The situation gets worse and more complex as the automotive hub bearings are subjected to random multi-axial loading conditions from the inputs from the road [1].

With new materials and technology development, automotive components should last longer. However, some observations on the premature failure of automotive hub bearings led to this investigation. Unveiling the underlying phenomena that promote premature damage of automotive hub bearings and their approximate lifetime is of interest to vehicle manufacturers and owners. The in situ gradual deterioration of hub bearings against road conditions is analyzed in this paper.

2. Literature review

Niu et al. (2013) [5] have studied the application of lateral loads to wheel hub bearings in real-world conditions and concluded that a large moment load will be produced with the increment of lateral acceleration, leading to a rapid decrease in bearing unit life. Studying ball bearings is more complex than it first seems. Much theory has to be involved to fully understand or simulate the behavior of bearings [6]. Having observed the damage to raceways in new automobiles after transporting to their destination via ship or train, Almen determined that such damage could occur from the ship engine vibration or when railway wagons vibrate each time they pass rail track joints [2]. Almen observed that raceways are damaged with periodic marks on the raceway.

Almen identified this type of damage as false brinelling for the first time [2]. According to bearing manufacturers [4], bearings should not get damaged if the load is below the static load rating of the bearing.

Hertzian elastic stress calculations can effectively interpret the stresses within a bearing against applicable loads [7]. Accelerated rolling contact fatigue tests were carried out at high Hertzian contact pressures, such as 6 GPa, to assess the effect of loading on the lifetime of bearings [8]. Under these high-stress conditions, silicon nitride specimens fail due to flaking or subsurface cracking. Dezzani and Pearson (1999) [9] have tested angular contact ball bearings at 2.62 GPa. They have reported that M50 bearings started to fail under contaminated conditions due to surface scratches originating at the stress level of 2.62 GPa [9]. A life of over 12 years at 12,000 rpm with a mean Hertzian stress of 0.52 GPa has been predicted by Jones (1997) [10].

Rosado et al. (2010) [11] have carried out Spall propagation experiments at 2.10 and 2.41 GPa for AISI 52100 and AISI M50 material. They have an initial low-rate spall growth followed by a critical spall growth rate [9]. Rosado et al. (2010) have induced initial fatigue spalls with four Rockwell hardness indents with a 150 kg load in these experiments. Arakere et al. (2009) [12] observed a significant spall propagation at 1.724 GPa peak Hertz stress.

In a separate test, Fouvry and Kapsa (2001) [13] observed surface damage at Hertzian pressure between 1 and 2 GPa based on experimental investigations on various hard metals. The number of fretting cycles imposed was between 1000 and 20,000 [13].

Table 1. A comparison of Hertzian pressure and type of damage to a bearing

Ref	Hertz pressure	Description of Life
8	6 GPa	Silicon nitride specimens fail due to flaking or subsurface cracking
9	at 2.62 GPa	Under contaminated conditions, M50 bearings started to fail
10	0.52 GPa	12 years running at 12,000 rpm for hybrid ball bearings
11	at 2.10 and 2.41 GPa.	Four Rockwell hardness indents were created with a 150 kg load. An initial low rate of spall growth, followed by a critical spall growth rate
12	1.724 GPa	have observed a significant spall propagation
13	1 GPa to 2 GPa	Observed true brinelling

A summary of observations made in the literature review concerning Hertz pressure and type of damage is summarized in Table 1. From Table 1, it is clear that a Hertz pressure above 1 GPa could be critical for a bearing, whereas it has been reported by Jones et al. (1997) [10] that under a Hertzian pressure of 0.5 GPa, a bearing could survive for a reasonable period of lifetime. Grebe et al. (2018) [14] have categorized the brinelling phenomena taking place within bearings due to different scenarios and loading conditions as "true" brinelling: standstill marks and false brinelling. The significance of false branding is that no plastic deformation occurs, as it is in a Brinell hardness test, and the indentation results from fretting-induced abrasion phenomena (Vincent 1994) [15]. The symptoms of true brinelling are Brinell marks similar to those that appear as indentations. It can cause premature fatigue failure. Once a bearing is damaged, it acts as a stress riser, leading to premature damage. The current research paper aims to analyze the premature failure of an automotive rear hub bearing due to false brinelling.

3. Material and Method

Damaged bearing assemblies from vehicles with known maintenance history and odometer readings were selected for observation. Identical bearing assemblies from vehicles with 32,000 km, 46,000 km and 64,000 km were selected for observations.

Figure 1 depicts the design of the hub-bearing assembly.

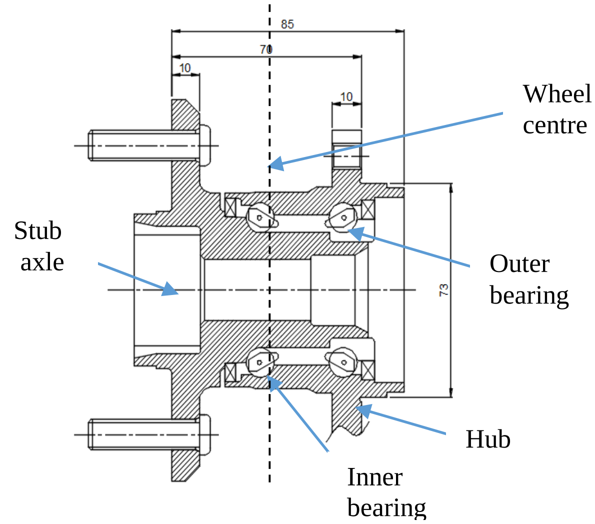

Fig. 1 Wheel–hub unit construction

The hub unit consists of two single-row angular baller bearings. Each bearing consists of 13 balls of 9.5 mm diameter. A preloading force is applied to the bearing during assembly manufacture.

The hub is subjected to the driving torque from the engine, a reaction from the ground opposing the driving torque, a component from rolling resistance torque, axial load and a reaction from the road. The wheel center was measured for the assembly by fixing the tire and taking measurements. It is assumed that the reaction from road inputs is directly transferred to the assembly at the road wheel center. The location of the wheel center is shown in Figure 1.

The loads acting on the hub bearing assembly are shown in Figure 2.

In Figure 2, Fa is the axial load, whereas Fr is the radial load transferred from the road surface, Fr1 is the reaction from the inner bearing, and Fr2 is the outer bearing. Fs1 and Fs2 are the derived forces in the axial direction generated from Fr1 and Fr2. An additional bending moment is created on the inner bearing as the load distribution is not symmetrical, as shown in Figure 1. The load distribution analysis reveals that the inner bearing supports more loads transferred from the road surface.

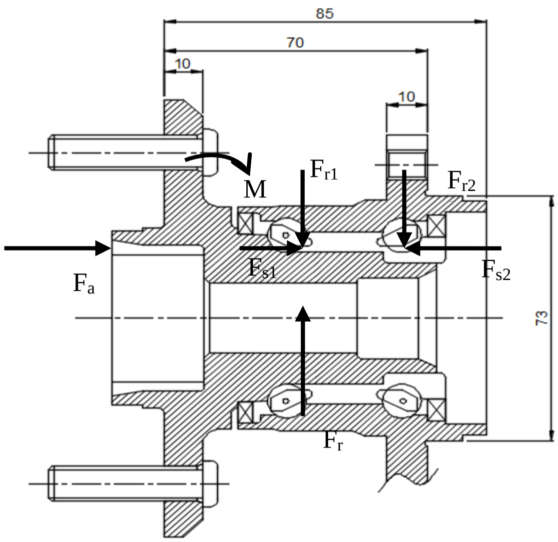

Fig. 2 Design of the Hub with bearings

The vehicle manufacturer originally fitted all three bearing assemblies. Bearing assemblies were replaced due to driver complaints of severe noise and the observations of service personnel during regular maintenance. All three bearing assemblies have been dismantled for observation. The visual examination of each bearing assembly is presented from Figure 3 to Figure 5.

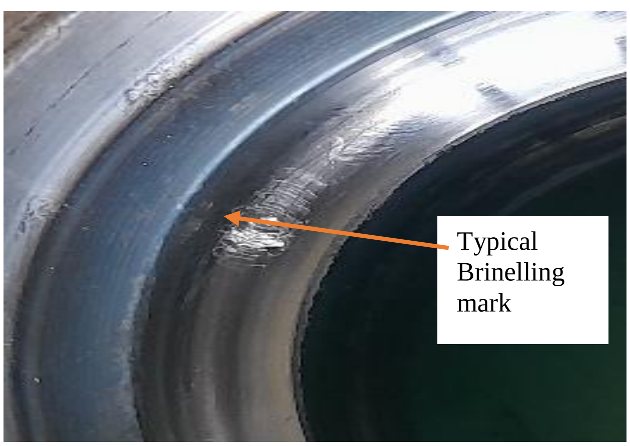

Fig. 3 Damage observed on the 32,000 km bearing assembly

The damage observed on the 32,000 km bearing assembly is shown in Figure 3.

A noticeable damage typical of that of brinelling appeared only in one location. The location was at the uppermost position of the bearing when assembled on the vehicle.

The damage observed on the 46,000 km bearing assembly is shown in Figure 4. A noticeable damage typical of that of brinelling appeared in multiple locations. The location was spread on either side of the uppermost position of the bearing when assembled on the vehicle.

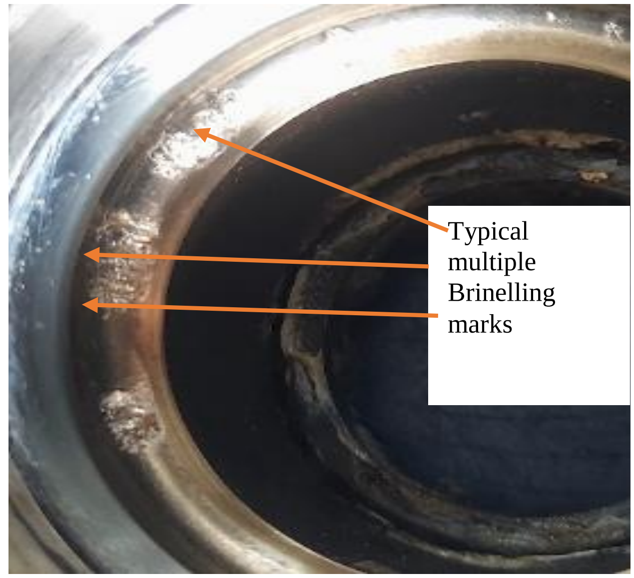

Fig. 4 Damage observed on the 46,000 km bearing assembly

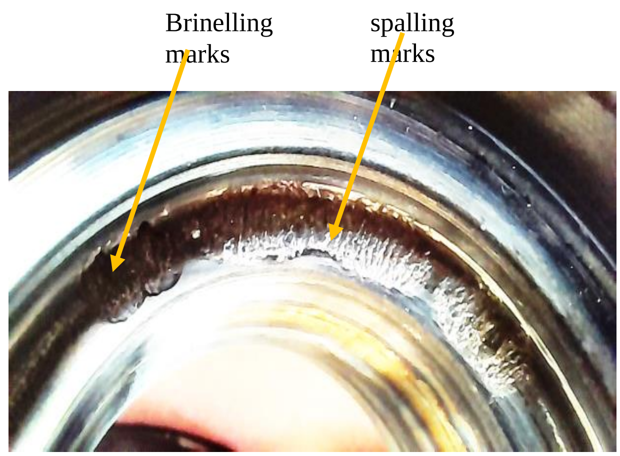

Fig. 5 The damage observed on the 64,000 km bearing assembly

The damage observed on the 64,000 km bearing assembly is shown in Figure 6. Severe damage typical of that of excessive loading was appearing in multiple locations. The location was spread on either side of the uppermost position of the bearing when assembled on the vehicle.

No damage was observed on the outer bearing. However, severe damage was visible on the inner bearing. The damage was predominantly on the outer race. The damage shows that typical damage corresponds to true brinelling [4].

It is clear from the optical image analysis that the profile corresponds to dents created by individual balls of the bearing. This can mainly occur due to true brinelling, vibration-induced fretting damage. According to the literature review, the damage could be due to the quality of material, misalignment during assembly or excessive loading. The misalignment issue was dismissed as the bearing was manufactured as an assembly and was fitted by the original manufacturer in all three cases. The material composition was tested using Arc Optical Emission Spectroscopy, and the results were compared with the standard specification for Bearing Steel. The composition comparison revealed that the material for manufacturing the bearing complies with a typical medium carbon steel used for Carburizing (case-hardened) bearing steel. The comparison of the chemical composition of the tested material and a typical medium carbon-bearing steel that matches the tested sample is given in Table 2.

Table 2. Material composition

	C	Cr	Si	Mn	P	S
test	0.4	0.14	0.27	0.8	0.040	0.004
1040	0.37 to 0.44		0.15 to 0.35	0.60 to 0.90		

The microstructure of the material was tested in the core and the case.

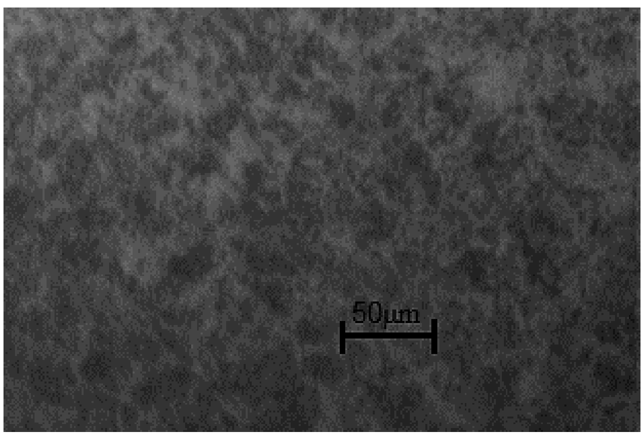

Fig. 6 Microstructure of the case hardened section

The microstructure of the case-hardened section is shown in Figure 6, whereas Figure 7 depicts the microstructure of the core of the material.

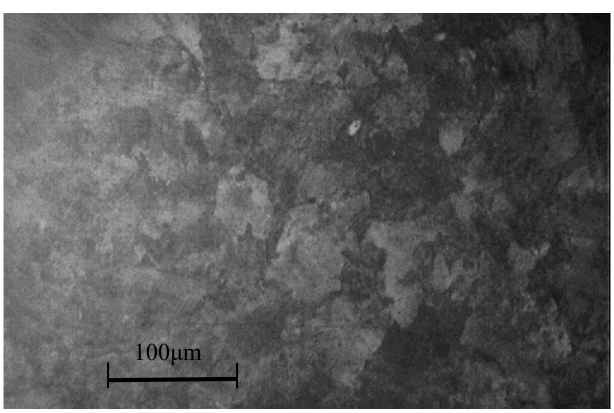

Fig. 7 Microstructure representing the core of the material

4. Analysis

Even though a road surface should be as flat as possible, defects such as potholes created over time and different surfaces such as speed breakers created on purpose can be found on roads.

The speed breakers can be categorized into speed humps, speed bumps and speed tablets [11]. It is expected that the vehicle speed will decrease with the height of the speed breakers [11]. However, drivers may drive over these speed breakers at high speed if the driver is not familiar with the area and the nature of the speed breaker. This kind of movement can give rise to an abrupt, jerky movement of the sprung mass.

This kind of movement results in severe stresses on suspension components. Georgiev and Kunchev (2019) [16] have experimentally determined the impact of speed breaker dimensions on the dynamic forces in the wheel axle. They have carried out experiments on a suspension with a 205/55 R16 radial tire on a steel wheel inflated to 2.1 bar at 20, 30 and 40 km/h for different profiles of speed barriers.

The height of the speed barrier was kept at 60 mm while the angle is measured from the flat level to the topmost position of the speed breaker. The angle is referred to as the angle of attack here. In the experimental investigation, Georgiev and Kunchev (2019) [16] have found that the acceleration increases with the linear velocity and the speed breaker angle of attack.

The maximum acceleration reported by Georgiev and Kunchev (2019) [16] was 5.39G at 40 km/h velocity for a 22.6-degree angle of attack barrier. Further, they have estimated by FEM analysis that, one of the members of the suspension system reached a stress level of 350 MPa at 65 km/h. The yield strength of the material used to fabricate the suspension system was 350 MPa.

The data given by Georgiev and Kunchev (2019) [16] was extrapolated to obtain further results for a vehicle moving at 60km/h and 70 km/h over a 160 mm speed breaker with a 70 and 80-degree angle of attack. The extrapolated results are given in Table 3.

Table 3. Results of G force for different speeds and angles of attack

Angle of attack	**Velocity 60km/h**	**Velocity 70km/h**
70 degrees	24.0 G	28.4 G
80 degrees	27.1 G	32.1 G

The unsprung mass of the suspension system was found to be 37 kg. The equivalent force is calculated by multiplying the G force by the unsprung mass. The analysis of stress state for the conditions extrapolated in Table 3 is given in Table 4.

Table 4. The stress state for the conditions extrapolated in Table 3

G force	Force equi.	Max. contact pressure (GPa)	Max. shear stress (MPa)	Depth of max shear stress mm	Circular contact area mm
24	8700	4.598	1373	0.473	1.90
27	9800	4.775	1426	0.492	1.98
28	10,000	4.807	1435	0.496	1.993
32	11,600	5.061	1511	0.52	2.09

The observed damage profile consists of spalling on the raceway surface with spacing equal to the distance between balls on the outer ring of the angular contact ball bearing. This is identical to that shown in [4] for damage due to: Excessive load, everyday fatigue life, improper handling improper installation, insufficient accuracy of shaft or housing, insufficient clearance, contamination, rust, insufficient lubrication, and reduction in hardness due to abnormal temperature rise [4].

5. Discussion

Some observations on the premature failure of automotive hub bearings led to this investigation to unveil the underlying phenomena that promote premature damage and their approximate lifetime. Three wheel hub bearings failed at 32,000 km, 46,000 km and 64,000 km were used for the analysis. Hertzian elastic stress calculations have been used to interpret the stresses within a bearing against applicable loads. From the literature review, it was clear that a Hertz pressure above 1 GPa could be critical for a bearing, whereas a bearing subjected to a Hertzian pressure of 0.5 GPa could survive for a reasonable lifetime period.

Analysis of the forces applied to the hub bearing assembly revealed that the inner bearing supports a greater portion of the loads transferred from the road surface. The visual examination revealed a gradual increase in damage with the mileage of the bearing. The damage location was spread on either side of the uppermost position of the bearing when assembled on the vehicle.

It was clear from the optical image analysis that the damage profile corresponds to dents created by individual balls of the bearing. This can occur mainly due to true brinelling due to vibration-induced fretting damage. Testing of material properties revealed no degradation of material composition or physical properties. In analyzing several road profiles against the linear velocity of vehicles travelling over them, conditions that favor damage were evident.

The numerical analysis revealed that under normal driving conditions, the local stress could reach 4 – 5 GPa within the hub bearing if the road wheel of a vehicle that is driven above 60 km/h encounters a road irregularity with an angle of attack greater than 70 degrees.

6. Conclusion

A wheel hub bearing could suffer damage from impacts during high-speed operation. Higher speeds typically result in greater force and, consequently, more severe impacts on the axle and other components. The angle of attack is another parameter that determines the impact force on the axle bearing.

- Optical imaging revealed that at some operating point, the entire load of the axle was absorbed by a single ball.
- At above 1 GPa, Hertzian pressure, bearings could initiate premature damage.
- At Hertzian pressure levels of 2.10 and 2.62 GPa, Bearings are subjected to spall propagation.
- Driving over a hump or a pothole of 160 mm or above with an angle of attack above 70 degrees could be detrimental to the axle bearing of an automobile.
- Conventional bearing fatigue life calculations cannot accurately predict the time or number of load cycles needed to initiate a defect under rolling contact fatigue.

6.1. Future Work

It is suggested that simulation tests be carried out under laboratory conditions, similar to real-world conditions, and further validate the outcome of the results presented in this paper.

References

[1] Li-Hui Zhao et al., “Service Life Prediction Method for Wheel-hub-bearing Under Random Multi-axial Wheel Loading,” *Engineering Failure Analysis*, vol. 122, 2021. [CrossRef] [Google Scholar] [Publisher Link]

[2] J.O. Almen, “Lubricants and False Brinelling of Ball and Roller Bearings,” *Mechanical Engineering*, vol. 59, no. 6, pp. 415-422, 1937. [Google Scholar]

[3] R.K. Upadhyay, L.A. Kumaraswamidhas, and Md. Sikandar Azam, “Rolling Element Bearing Failure Analysis: A Case Study,” *Case Studies in Engineering Failure Analysis*, vol. 1, no. 1, pp. 15-17, 2013. [CrossRef] [Google Scholar] [Publisher Link]

[4] NTN Make the World Nameraka. [Online]. Available: https://www.ntnglobal.com

[5] Rong Jun Niu, Chen Niu, and Xiao Juan Dai, "Life Estimation of Hub Bearing Unit Based on Test Load Spectrum," *Applied Mechanics and Materials*, vol. 457-458, pp. 443-446, 2013. [CrossRef] [Google Scholar] [Publisher Link]

[6] Simon Rommelaere, "*Ball Bearing Design Tool Evaluation*," Thesis Report, pp. 1-17, 2017. [Google Scholar] [Publisher Link]

[7] P.R. Anoopnath, V. Suresh Babu, and A.K. Vishwanath, "Hertz Contact Stress of Deep Groove Ball Bearing," *Materials Today: Proceedings*, vol. 5, no. 2, pp. 3283-3288, 2018. [CrossRef] [Google Scholar] [Publisher Link]

[8] J.F. Dill, "Hybrid Bearing Technology for Advanced Turbomachinery: Rolling Contact Fatigue Testing," *Journal of Engineering for Gas Turbines and Power*, vol. 118, no. 1, pp. 173-178, 1996. [CrossRef] [Google Scholar] [Publisher Link]

[9] M.M. Dezzani, and P.K. Pearson, "Hybrid Ceramic Bearings for Difficult Applications," *Journal of Engineering for Gas Turbines and Power*, vol. 118, no. 2, pp. 449-452, 1996. [CrossRef] [Google Scholar] [Publisher Link]

[10] William R. Jones, Long Term Performance of a Retainerless Bearing Cartridge with and Oozing Flow Lubricator for Spacecraft Applications, 1997. [Google Scholar] [Publisher Link]

[11] Lewis Rosado et al., "Rolling Contact Fatigue Life and Spall Propagation of AISI M50, M50 NiL, and AISI 52100, Part I: Experimental Results," *Tribology Transactions*, vol. 53, no. 1, 2010, pp. 29-41, 2009. [CrossRef] [Google Scholar] [Publisher Link]

[12] Nagaraj K. Arakere et al., "Rolling Contact Fatigue Life and Spall Propagation of AISI M50, M50NiL, and AISI 52100, Part II: Stress Modeling," *Tribology Transactions*, vol. 53, no. 1, pp. 42–51, 2009. [CrossRef] [Google Scholar] [Publisher Link]

[13] S. Fouvry, and Ph. Kapsa, "An Energy Description of Hard Coating Wear Mechanisms," *Surface and Coatings Technology*, vol. 138, no. 2-3, pp. 141–148, 2001. [CrossRef] [Google Scholar] [Publisher Link]

[14] Markus Grebe et al., "Damage Mechanisms in Pivoting Rolling Bearings and Their Differentiation and Simulation," *Bearing World Journal*, vol. 3, pp. 71-86, 2018. [Google Scholar] [Publisher Link]

[15] R.B. Waterhouse, and T.C. Lindley, *Fretting Fatigue (ESIS 18)*, Wiley, pp. 1-542, 1994. [Google Scholar] [Publisher Link]

[16] Z.A. Georgiev, and L.P. Kunchev, "Study of the Stresses in the Front Suspension Components of a Car Passing Over Speed Breakers," *IOP Conference Series: Materials Science and Engineering*, vol. 664, no. 1, pp. 1-6, 2019. [CrossRef] [Google Scholar] [Publisher Link]