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Fatigue life prediction of package of suspension automotive under random vibration based on road roughness

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ABSTRACT

The impact of a suitable suspension system for passenger comfort and vehicle steering is an obvious order and direct impact on the safety of passengers must be considered, to do so different kinds of tests must be exerted, one of these is fatigue testing which is one of the most significant ones. Another issue is the high cost in practical ways, and to cope with this issue various ways must be assessed and analyzed, one of the best and the most efficient ways is modelling and testing in virtual software environments. In the present paper, predict fatigue life of suspension component and package of automotive suspension are the main purposes. First, using MATLAB software, road roughness according to the intercity roads for constant vehicle velocity (100Km/h) has been studied. After that frequency response of components has been analysed, its critical points determined to calculate the fatigue life of the part, and the amount of critical stress obtained based on Von Misses, Tresca and Max Principle criterion for a quarter car model (passive suspension System in 206 Peugeot). Fatigue life of the vehicle components are calculated in terms of kilo-Meters in specialized fatigue software such as 116944, 92638.9, 46388.9 and 191388.9 Km respectively wheel hub, pitman arm, suspension arm and package of suspension. Finally, to prove the given results of the finite element method compared with reported results by other researchers and the results match very well with those.

1. Introduction

Investigating Fatigue is one the most important factors in designing most mechanical structure. The reason is that, in many cases, the specimens of the structure break down without any warning or alarm. However, Fatigue is the main reason for the failure of most mechanical parts during operation [1]. Such as :

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– Comet airliner crashed in the Mediterranean Sea.
– Catastrophic crash of jet F-111 that has an important role to development of B-1 bomber base on fracture mechanics [1].
– Plazant Bridge in West Virginia failed without any warning [2].

Vibration Responses have a great importance in all fields of science and engineering, so study of the material behavior under random vibrations is necessary in automotive industry.

Suspension systems are part of chassis that have an important role in the behavior and performance of vehicle. Any suspension system consists of 4 parts such as springs, shock absorbers, suspension mechanism and interface link connection that are responsible for the following key tasks [3] :

Create a separately conditions of chassis from road roughness to comfort and safe travel.

Create a permanent contact between tire and road conditions in order to provide Proper tire performance on vehicle longitudinal and lateral movement.

Create proper conditions for the stability of the vehicle with good behavior during the rolling and heading movement (degree of freedom decrease).

ElMadany in 1990 has been analyzed the efficiency of passive suspension system of transport vehicles like trucks that obtained the power spectral density function of road roughness on the vehicle axles (front and rear axles) by linear and nonlinear springs [4].

Bishop et al. has been studied the finite element analysis of fatigue failures that occurred in several of the industry specimens (railroads, pitman arm of suspension system of vehicle, etc.) [5]. Badih in 2002 was presented a complete design and construction of a wheel of the race car including geometric model of the suspension and its components (hub, pitman arm, etc) [6].

Haiba et al. are discussed in pitman arm's suspension to develop optimization algorithm based on fatigue life. In this research used 1/2 car model movement with constant velocity ($34Km/h$) on the paved asphalt road and under 3 different loads such as static, transient dynamic and harmonic with constant amplitude. So, predict fatigue life by using superposition of 3 directions of concentrated load [7]. After that Nadot study of the fatigue failure of pitman arm's suspension with multi axial loading that is discussed on the defects in the heat treatment and its effects on low and high cycle fatigue [8].

Svensson et al. predict fatigue life of suspension arm [9] and Buciumeanu developed the design of suspension components in corresponding wear and fatigue. Reports that there is more damage in the screw, so, published several solutions to decrease its effect [10].

Soleimani eyvari elicited transferred shocks to the vehicle on a paved road by experimental data [11]. Martinez measured and analyzed vibration behavior in terms of speed for truck transportation in Spain [12]. Colombo studied the unusual fracture of McPherson suspension and found that critical stress on the top of connection linking between pitman arm and suspension arm is about $358Mpa$ [13].

Borg reported An Approach to Using Finite Element Models to Predict Suspension Member Loads in a Formula SAE Vehicle in his Master thesis of Science in Mechanical Engineering at University of Technological Virginia [14].

Rahman et al. investigated to predict Fatigue Life of Lower Suspension Arm Using Strain-Life Approach [15]. Breytenbach optimized vehicle suspension characteristics for increased structural fatigue life [16] and Tung investigate Optimization of the exponential stabilization problem in active suspension system using PSO Method [17].

Schwab published one chapter about how can predict fatigue life base on random vibration loading. So, used 3D modeling and assumes 3 function for each direction to obtain random vibration response versus stress history and RMS function [18].

In most published studies, investigate fatigue life of suspension component separately but in the present research, investigate package of suspension system in addition suspension components such as wheel hub, pitman arm, and triangular suspension arm. Another innovation of this research is applying random vibration response to predict fatigue life directly, however in the past apply uniaxial or biaxial load, so use superposition law to predict it. This advantage should be noted that in order to achieve safety data, we are considered vehicle velocity as an influence parameter on entering loads to it from roads.

2. Road Roughness Simulation

Among the factors that are important in simulating a suspension system and achieving the desired response are the inputs of that system which are the natural inputs of road roughness. Generally, to study the characteristics of passenger cars, the excitations of the ground are used in different forms of sine waves, step functions or triangular waves. Later, it was found out that road surface profile would be more practical and realistic as a random function which it is closer to reality. So, International Organization for Standardization (ISO) has proposed a road roughness classification (classes A-H) based on the power spectral density functions such as PSD that is presented parametrically in Table 1.

Table 1. Parametrical classification of road roughness suggested by international standard organization ISO [19-20].

Road Class	**Degree of Roughness Range**	$\mathbf{S}_g(\Omega_0)$, $\mathbf{10^{-6}m^2/cycles/m}$ **Geometric Mean**
A (Very Good)	<8	4
B (Good)	8 – 32	16
C (Average)	32 – 128	64
D (Poor)	128 – 512	256
E (Very Poor)	512 – 2048	1024
F	2048 – 8192	2048
G	8192 – 32768	4096
H	More than 32768	16384

In the classification of the road surface roughness suggested by the international organization for standardization (ISO), relationships between the power spectral density function $Sg(\Omega)$ and spatial frequency Ω for different roads can be expressed with two straight lines and different slopes in the logarithmic scale that for two different ranges of frequencies are as follows [20-21] :

For $\Omega \leq \Omega_0 = \frac{1}{2\pi}$Cycles/m

$$Sg(\Omega) = Sg(\Omega_0) \times \left(\frac{\Omega}{\Omega_0}\right)^{-N_1} \tag{1}$$

And for $\Omega > \Omega_0 = \frac{1}{2\pi}$Cycles/m

$$Sg(\Omega) = Sg(\Omega_0) \times \left(\frac{\Omega}{\Omega_0}\right)^{-N_2} \tag{2}$$

Values of $Sg(\Omega_0)$ in the spatial frequency $\Omega_0 = 1/2\pi\ Cycle/m$ for different roads are presented in Table 1 and constants of N_1 and N_2 are respectively considered as 2.0 and 1.5.

In the present research, Firstly By using a quarter Car Model of passive suspension of 206 Peugeot in according road classification ISO 2631-1 is achieved road roughness as PSD function in constant speed of car ($V = 100Km/h$) by using MATLAB Code [22-23] and compared with Road Classification suggested by ISO as shown in Figure 1.

In the next step, vertical displacement of road in classification A until E in term of travel length as shown in Figure 2.

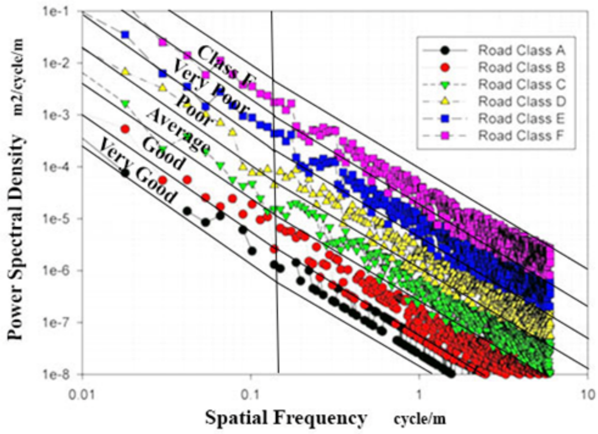

Fig. 1. Compared classification of surface roughness simulated in MATLAB Software and suggested by ISO [24-25].

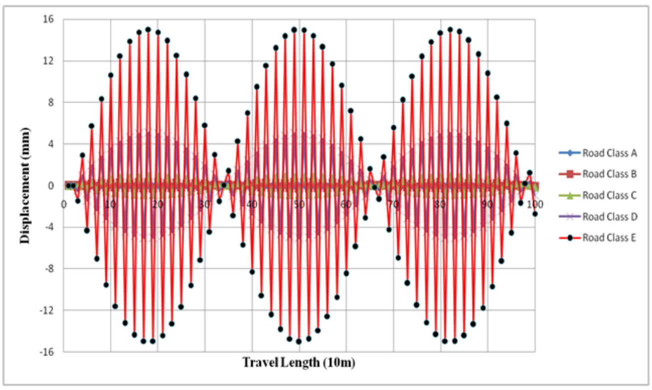

Fig. 2. Road Roughness function in terms of Travel Length in different Road Classification by ISO with Constant Velocity 100 Km/h [22].

For vehicle vibration Analysis, it is more convenient to express the power spectral density of surface profiles in terms of the temporal frequency in Hz rather than in terms of the spatial frequency, since vehicle vibration is a function of time. The transformation of the spatial frequency Ω in Cycles/m to the temporal frequency f in Hz is that of the speed of the vehicles [20-21] :

$$f\,[Hz] = \Omega\,[Cysle/m] \times V\,[m/s] \tag{3}$$

The transformation of the power spectral density of the surface profile expressed in terms of the spatial frequency $Sg(\Omega)$ to that in terms of the temporal frequency $Sg(f)$ is through the speed of the vehicle [20-21] :

$$sg\,(f) = \frac{sg\,(\Omega)}{v} \tag{4}$$

By using MATLAB Coding of Road Roughness function in terms of Travel length in a quarter car Model express compared ISO classification of road roughness as PSD function in terms of frequency Hz [11, 25] with constant speed of car as shown in Figure.3.

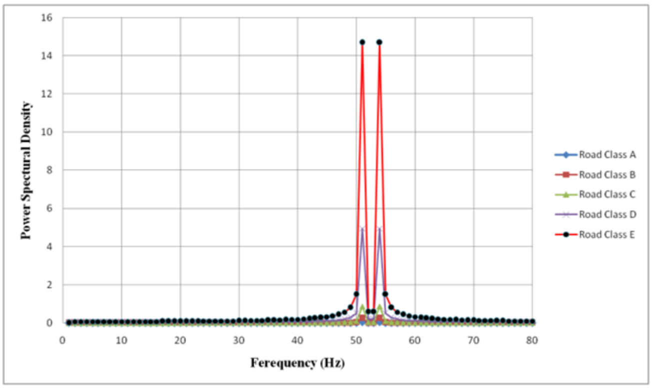

Fig. 3. Compared Road Roughness as PSD function in terms of ferequency Hz in Different Class Road by ISO 2631-1 with Constant Velocity 100 Km/h [22].

3. Finite Element Analysis

In this research, predict fatigue life of suspension components such as wheel hub, suspension arm, pitman arm and package of suspension system, means calculating the number of failure cycles. So for the finite element analysis in software, stress-life method (S-N method) has been used based on the Rain flow cycle counting. A flowchart of the working process in software for the analysis is presented in Fig 4.

3.1. Geometric Modelling

Firstly, the suspension components has been modelled in CATIA modelling software and in the next step, assemble all parts to create package of suspension system as shown in Figure.5.

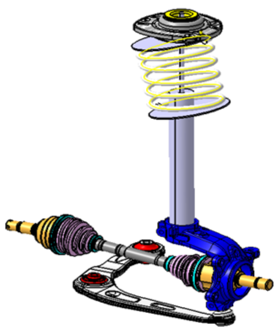

Fig. 4. An image of assemble modelling of package of suspension system in CATIA.

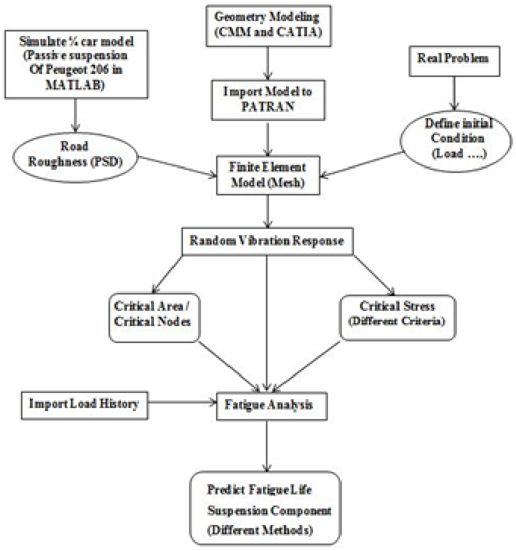

Fig. 5. A flowchart of the working process in software for the finite element analysis of suspension components.

3.2. Material Information

To completely simulate finite element model, information and standards related to the part type provided and considering the constituent materials of the type and determining the part construction operation (casting, good, average and bad machining, polished surface, thermal treatment, etc.) it was applied in the section of materials management in software so that mechanical properties of the part type can be seen in table 2.

Table 2. Mechanical Properties of suspension component [26].

Piece	DIN Standard	Ultimate Stress (MPa)	Yield Stress (MPa)
Wheel hub	20MnCr5	780-1080	540
Pitman Arm	41Cr4	800-950	560
Suspension Arm	31NiCr14	780-930	635
Spring	60SiCr7	1320-1570	1130
Pin	38MnSi4	980-1180	785

3.3. Loading Information

Considering a car moving at the speed of $100Km/h$ and given that the car is moving on a straight road without any changes in the steering column, loading based on the road signals obtained in accordance with the class D road roughness has been applied vertically on the wheel hub located on the pitman arm where can see the fixture and load location on Figure 6.

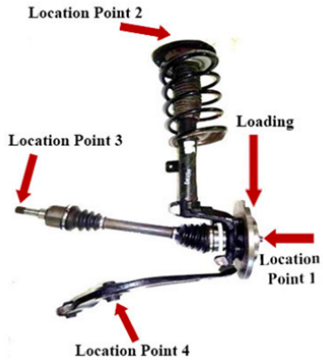

Fig. 6. Fixture and load location on the assembly suspension model in FEM.

To specify the location of the suspension system is used three-dimensional Cartesian coordinate. Where X-axis is along the length of the vehicle, the y-axis and z-axis along the width and height direction of the vehicle. So, Zero point of coordinate is considered at the contact surface of the wheel and ground. location of other 4-point on the system is as follows :

Location Point 1 : the center point of wheel hub is $(0, 0, +28cm)$.

Location Point 2 : spring is in the $Y - Z$ plane and makes an angle of 15 degrees with Z axis.

Location Point 3 : end point of shaft compared to the initial point has no component along the x-axis. In the Z direction has component value of $27mm$ placed above the wheel hub center.

Location Point 4 : coordinates x and y of middle Bush of triangular suspension arm is exactly located under shaft. On the other hand it is parallel to the axis of the wheel hub center, only in the Z direction ; there is component value of $189mm$ in height. In the other world, is located $91mm$ below of the wheel hub center axis.

In addition, the loading location is shown in the figure 6.

4. Results and Discussion

4.1. Random Vibration Response

The analysis is used to calculate the steady state response of a structure under sinusoidal excitation or generalizable to a sinusoidal excitation with variable amplitude and phase. The most important results obtained from this analysis are : Critical areas and stress, velocities and accelerations in the nodes, forces and stresses on elements.

Since, the analysis is very time-consuming and subsequent fatigue analyses are more time-consuming, it is attempted to choose a proper mesh size by obtaining the extent the mesh size of triangular element Tet10 effects the critical stresses in the suspension system constituents, and overviewing the results. So it needs less workspace for calculations, and results in a shorter time. Therefore, given different sizes for mesh grading, the information related to the critical stresses in the different parts is reported in table 3.

Table 3. The effect of mesh size on the critical stress in the suspension system according to von Misses criterion.

Model	Size Mesh	Number of Nodes	Number of Elements	Critical Stress (MPa)
Wheel hub	0.1	8017	4394	282
	0.3	7311	4040	283
	1	5352	2902	237
Pitman arm	0.1	17723	35478	369
	0.3	14860	29752	362
	1	9632	19296	348
Suspension arm	0.1	16789	32932	393
	0.3	14524	28466	385
	1	11842	23219	360.5
Package of suspension	0.1	18412	35897	402
	0.3	15621	30437	396
	1	10487	20250	370

In the following, critical stresses of the suspension components caused by random vibrations analysis according to different criteria are presented in Table 4.

Table 4. Critical stress in the suspension's pitman arm according to different criteria.

Model	Criterion	Von Misses	Tresca	Max principle
Wheel hub	Max Stress (MPa)	237	241	247
	Min Stress (MPa)	160	167	168
Pitman arm	Max Stress (MPa)	348	352	367
	Min Stress (MPa)	198.7	211	217
Suspension arm	Max Stress (MPa)	360.5	372	376
	Min Stress (MPa)	240	247	251.3
Package of suspension	Max Stress (MPa)	370.2	385.46	394.9
	Min Stress (MPa)	206.7	218.01	221

4.2. Fatigue Analysis

To predict the fatigue life under random vibration loading in MSC.FATIGUE software, in the first phase of the part random vibration response, it is necessary to determine the critical area in the part, identify the most critical nodes in the part under random vibration loading, and consider maximum critical stress of the part for using in the next step, so that complicated and time consuming fatigue calculations be done on the critical points of the parts' critical areas. Thus, using the analysis in the previous section, critical areas of components are shown in Figures 7-10 for using in fatigue analysis and its fatigue life prediction.

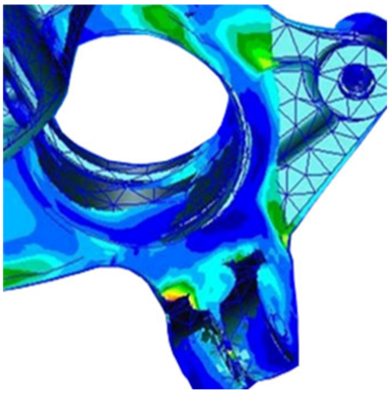

Fig. 7. A stress contour of the critical area of the pitman arm under random vibration loading.

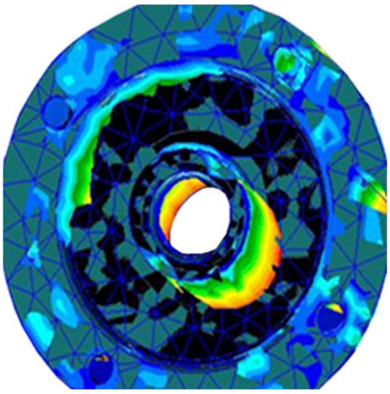

Fig. 8. A stress contour of the critical area of the wheel hub under random vibration loading [23].

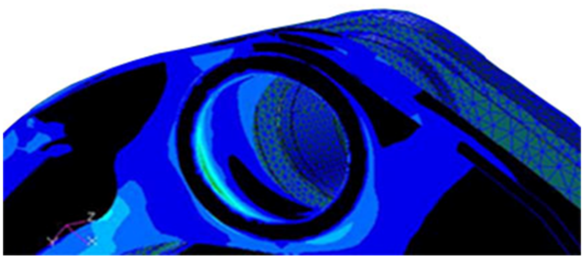

Fig. 9. A stress contour of the critical area of the suspension arm under random vibration loading.

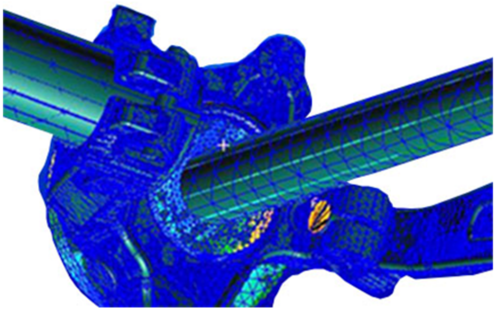

Fig. 10. A stress contour of the critical area of the package of suspension system under random vibration loading.

Then, using two different computational methods for predicting the fatigue life in software (Dirlik and narrow-band), the parts' fatigue life are obtained under loading conditions (car moving at a constant speed in class D road). Finally, given the elapsed time in a loading cycle, determined number of cycles' repetition before failure, and the car's speed, number of loading cycle before failure to the distance travelled by the car in terms of kilometer can be calculated. Therefore, fatigue life of the model has been reported in terms of its function in Table 5 [27-29].

4.2.1. Dirlik Theory

This is one of the best methods that has been subject to modifications. The numerical simulations of the time histories are used for two different spectra. By using a concoction of one exponential and two Rayleigh probability densities, the cycle-amplitude distribution has been approximates (In 1985). The Rain-flow cycle amplitude probability density estimate is given by [30]

$$P_a\left(s\right) = \frac{1}{\sqrt{m_0}}\left[\frac{G_1}{Q}e^{\frac{-z}{Q}} + \frac{G_2 Z}{R^2}e^{\frac{-z^2}{2R^2}} + G_3 Z e^{\frac{-z^2}{2}}\right] \tag{5}$$

Where Z is the normalized amplitude and x_m is the mean frequency, as defined by the author of the method [30] :

$$Z = \frac{s}{\sqrt{m_0}} \tag{6}$$

$$x_m = \frac{m_1}{m_0}(\frac{m_2}{m_4})^{0.5} \tag{7}$$

And the parameters G_1 to G_3, R and Q are defined as [30] :

$$G_1 = \frac{2\left(x_m - \infty_2^2\right)}{1+\infty_2^2} \tag{8}$$

$$G_2 = \frac{1-\infty_2 - G_1 + G_1^2}{1-R} \tag{9}$$

$$G_3 = 1 - G_1 - G_2 \tag{10}$$

$$R = \frac{\infty_2 - x_m G_1^2}{1 - \alpha_2 - G_1 + G_1^2} \tag{11}$$

$$Q = \frac{1.25\left(\alpha_2 - G_3 - G_2 R\right)}{G_1} \tag{12}$$

While α_2 has already been defined with [30] :

$$\alpha_i = \frac{m_i}{\sqrt{m_0 m_{2i}}} \tag{13}$$

The closed form expression for the fatigue life intensity has been derived in the form [30] :

$$\bar{D}^{DK} = C^{-1}\vartheta_p m_0^{\frac{k}{2}} \left[G_1 Q^k \Gamma\left(1+k\right) + \left(\sqrt{2}\right)^k \Gamma\left(1+\frac{k}{2}\right)\left(G_2\left|R\right|^k + G_3\right)\right] \tag{14}$$

4.2.2. Narrow Band Theory

Miles presented Narrow Band Theory in 1956. There is feasible consider that every peak is congruous with a cycle .as a result, the cycle amplitudes are Rayleigh distributed. Here, there is presented stress amplitudes [30] :

$$\bar{D}^{NB} = \vartheta_0 C^{-1}\left(\sqrt{2m_0}\right)^k \Gamma\left(1+\frac{k}{2}\right) \tag{15}$$

Where ϑ_0 is the expected positive zero crossings intensity, which is very close to the peak intensity ϑ_p.

For a narrow band process, C and k are material fatigue parameters. m_0 is the 0-th spectral moment and $\Gamma(.)$ is the Euler gamma function, which is defined as [30] :

$$\Gamma\left(z\right) = \int_0^\infty t^{z-1} e^{-t} dt \tag{16}$$

Table 5. Fatigue life in terms of kilometres travelled using different computational methods in finite element software.

Component	Dirlik Method	Narrow Band Method	Damaged Part
Wheel hub	116944	111541.7	The outer surface of the inner shaft
Pitman Arm	92638.9	86736.1	The Top right corner of joint in relate to Mounting link
Suspension Arm	46388.9	41388.9	The Internal of Middle rubber Bush
Package of Suspension System	191388.9	157222.2	The Top of Mounting link between pitman arm and suspension arm

5. Conclusion

Nowadays, the suspension systems designers used new technology in advanced vehicles. Studies in different fields of fatigue loading are very significant. Most of the time, it's directly related to the passenger comfort and safety without any warning about fracture time. For actual analysis and results, random vibration is applied for the simulation of the suspension components and assembly of them.

For avoiding of conducting of expensive tests, as well as, easy development of parameters for sensitivity analysis, engineers used finite element method. They can optimize the presented model parametrically.

We can. The research achievements are as follows :

1. The maximum and minimum stress is related to triangular suspension arm and Wheel hub respectively. So, if have a same cyclic loading on all parts of suspension components, the triangular suspension arm is failed. For instance, critical stress of pitman arm in Von Misses criteria was reported to be 348MPa that is less than its yield stress. So, this part will not fail under static loading and should be analyses fatigue phenomena.

2. According to the present study, when the vehicle was derived with a speed constant, 100 km/h, given the loading conditions burdened on the system. if investigate all parts in separate case, after 46388.9 travelled car, can see fatigue phenomena on the triangular suspension arm and the middle rubber bush will be failed and break dawn.

3. Finally, with respect to results, after 191388.9 km driving, failing occur on Mounting link between pitman arm and triangular suspension arm and with analysing of separated from each other, we can have exact inspection.

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Biographical notes

Kazem Reza Kashyzadeh, born in 1986, is currently a PhD candidate at Mechanical Engineering of Sharif University and Technology, International Campus, IRAN. His research interests include Fatigue and Fracture.