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Fatigue Analysis of the Weldments of the Suspension-System-Support for an Off-Road Vehicle under the Dynamic Loads Due to the Road Profiles

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Abstract: In this study, a numerical method has been developed to estimate the fatigue life of the vehicle chassis-components. The application of this method will significantly reduce the expenses for the required practical or in-service tests. In this method, firstly, using MATLAB, the real but random profiles for the standard roads have been produced on the basis of the random vibration theories, which give the spatial profile data in terms of time. These profiles have been used to study the dynamic behavior and to design the vehicle chassis. Then, a complete off-road-vehicle model has been developed and first-stage results have been used as input data to simulate the dynamic behavior of the model and to calculate the forces and displacements of the chassis structure using MSC.ADAMS. In this way, the optimum stiffness of the suspension system has been obtained. Finally, using the FEM based software; ANSYS, the stress distribution in a selected part-weldment (support of the front suspension system) has been obtained and fatigue life of the structure-components welds; joining the suspension system to the off-road vehicle chassis, has been calculated and the effects of the dynamic loads due to the road profiles have been studied.

Key words: Road profile, road profile induced dynamic loads, fatigue life, dynamic model of an off-road vehicle, weld joints

INTRODUCTION

In principle fatigue occurs due to the alternative and cyclic loads. Designing based on the fatigue life is one of the main requirements in the automotive industry. Fatigue failure can be divided into three stages: crack initiation, its propagation and growth up to the critical length and ultimate fracture. For the structures of vehicle or aircraft bodies, because of full time high rate alternative loading, crack initiation time is usually regarded as the fatigue lifetime (Huang *et al.*, 1998). Generally, the acceptable vehicle life is defined about 200000 km (Rice, 1997). Up to now, the usual method for vehicle-structure-life-time estimation was experimental tests. By developing the numerical methods, nowadays, estimation and evaluation of this lifetime can be carried out using these methods (Benasciutti and Tovo, 2006).

The numerical lifetime assessment of the vehicle chassis can be carried out by the generation of a virtual road profile and studying the vehicle response on this road path. These chassis are mainly made of formed metal plates attached to each other using welded joints. Other parts, such as engine or suspension system are also connected to chassis using welded joints. In heavy trucks and some other special duty vehicles, different parts of chassis are connected to each other by welding. Therefore, the chassis and their welded joints carry the applied loads and should be designed to sustain these loads with acceptable safety factor and fatigue lifetime. In general, the applied loads can be divided into internal and external loads. The former are the loads due to the vibration and inertial forces of the moving parts (primarily the engine) whereas the latter one is due to the moving of the vehicle along the road with uneven profile.

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The main objective in this study is to develop a methodology to calculate the dynamic loads due to the road profiles for a Basic Utility Vehicle (BUV) and analyse the fatigue life of the Weldments subjected to these loads using numerical methods. For this purpose, the effect of the road profiles on the suspension system of a (BUV) in different speeds has been studied. The stiffness of the tires has also been taken into account. A model has been developed to simulate the practical road profiles and using these results, the optimum combination of the suspension system has been obtained (Uys *et al.*, 2007; Pesterev *et al.*, 2005). At the final stage, the dynamic model of the vehicle has been developed and used to study the behavior of the suspension system and its vibration under different loading and road profile conditions to calculate the absorbed forces. Also, using these data the fatigue life for the joining weldment of the front-suspension-support system has been calculated and the critical location in this suspension-support has been identified. These results provide a basis to evaluate the fatigue damage and to predict the remaining life of the suspension support.

MATERIALS AND METHODS

This study has been conducted at the University of Tabriz, Iran to provide a basic methodology to help the design procedure in domestic Automotive Industry. The basic goal was to reduce the expenses by eliminating or at least supporting the expensive and time consuming tests. The major steps of this research have been taken since 2006 and its results have been used to design a 150 horsepower 4WD -BUV.

Different Road Profiles

There are two types of road profiles: stationary and ergodic (Sayers and Karamihas, 1998). In general, the real road profiles have a completely random nature and therefore their shape does not follow a defined pattern, however, using different methods, these profiles can be measured and modeled. As it can be shown in Fig. 1, if the roughness of the path has been measured between A and B and also C and D and the condition in both paths would be the same, the profile is stationary. Also,

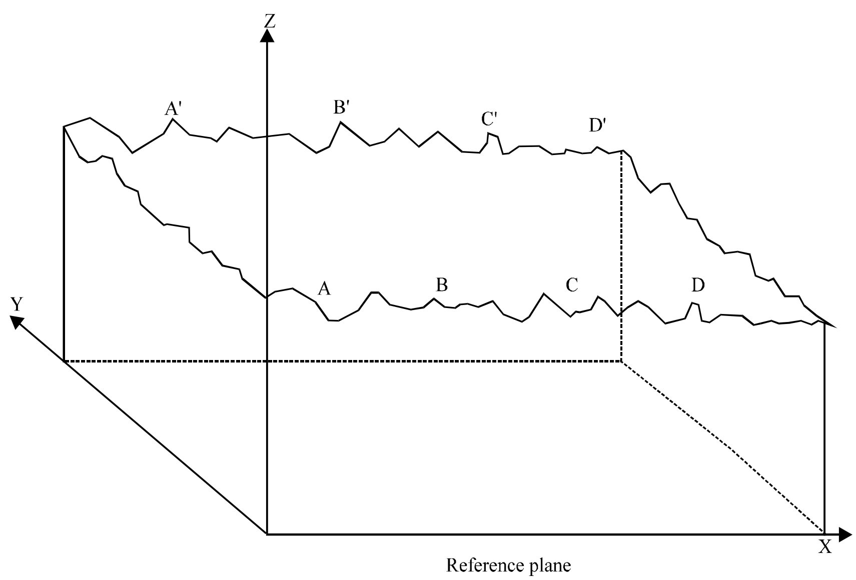

Fig. 1: Surface elevation as a random function

if the profile in A-D and A'-D' in parallel planes would be the same, the profile is ergodic. This would be the most ideal situation and in this case, by developing a method to produce a part of a profile, it can be extended to produce a 2D or 3D model of the whole path.

Materials of the Suspension Support

The rack is AISI 304-Si steel with $S_y = 235$ MPa, $S_u = 350$ MPa, $\mu = 0.292$, $\rho = 7800$ kg m^{-3} and E = 207000 MPa. To observe the strength and mechanical behaviour of the structure, At first, a 3D Finite Element model of the chassis and suspension-supports have been developed and static analysis has been carried out to identify the critical points and it has been observed that the node 4326 at the weldment joint between rack and chassis has the maximum stress level. In this way the critical point (location) to study the fatigue life of the structure under dynamic loads has been identified. Also, for the E6013 welding electrode, it has been shown that the stress levels are below the elastic limit.

Modeling and Generating the Road Roughness Functions

Basically two methods can be proposed for modeling the road roughness and producing a profile function.

Experimental Method

In this method special instruments such as profiler or rough-meter are used to measure the roughness of a certain path of the road and the collected data are used to produce a road profile. Figure 2 and 3 show some of the major methods (instruments) which are used to measure the roughness of the road. Also, as it can be seen in Fig. 4, G.M or I.C.C. can be used to measure the road roughness using the vertical acceleration of the device when it passes over the road profile (Ruotoistenmaki *et al.*, 2006).

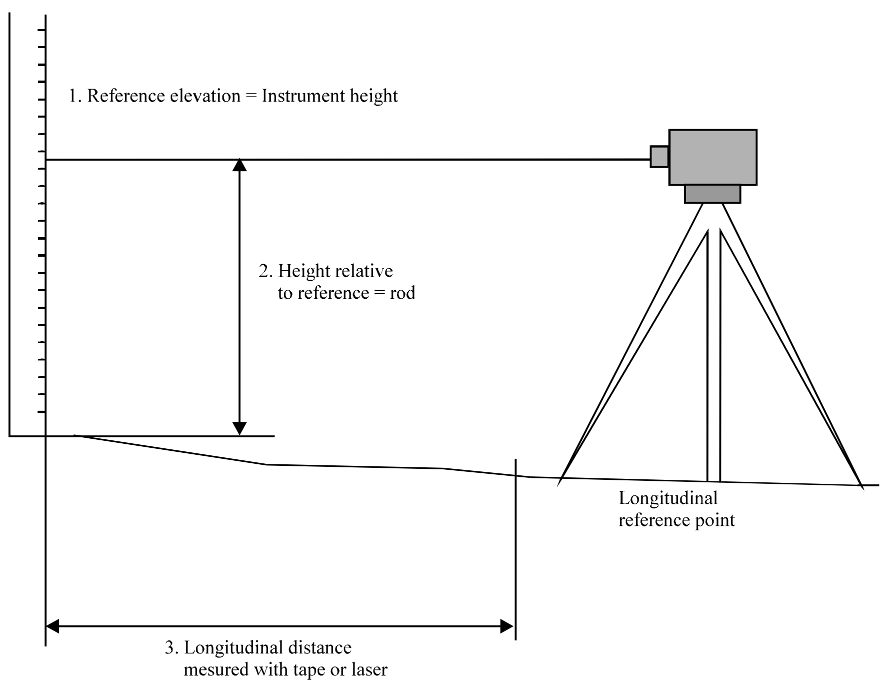

Fig. 2: Static measuring instrument-rod and level

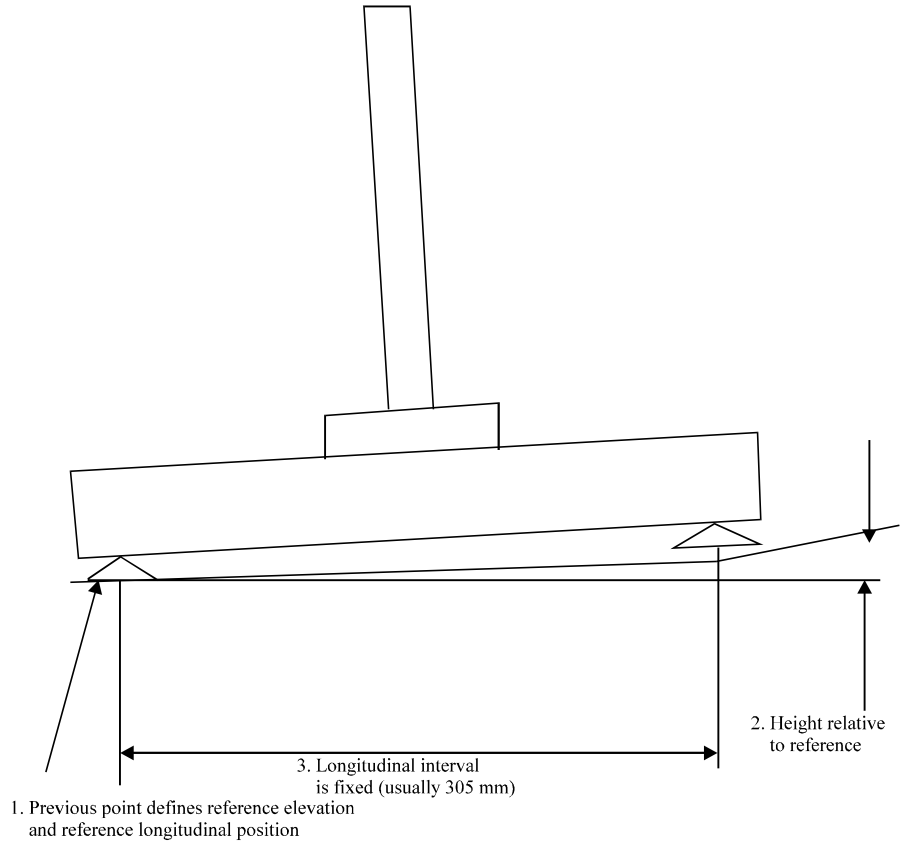

Fig. 3: Dipstick static measuring instrument

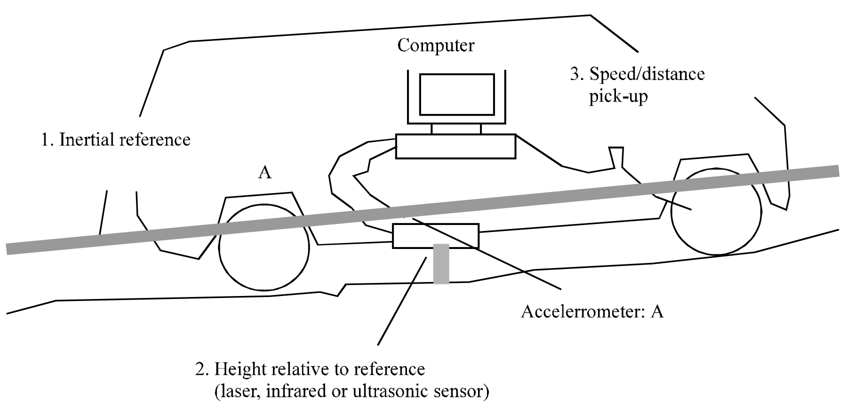

Fig. 4: Dynamic measuring instrument-GM

After measuring the road profile, these point-data should be filtered and also be processed. These data usually include local roughness and also overall shape of the road such as hills and valleys along the path. In filtration process, the latter types should be eliminated. This can be carried out using high or low pass filtration. Using the high pass filtration such as Eq. 1, the data larger than a certain level are accepted and others will be ignored (Sayers and Karamihas, 1998; Gillespie, 1993):

For a profile p that has been sampled at interval DX, a moving average smoothing filter is defined by the summation:

$$p_{fL} = p(i) - \frac{1}{N}\sum_{j=i-\frac{B}{2\Delta x}}^{i+\frac{B}{2\Delta x}} p(j) \tag{1}$$

where, P_{fL} is the smoothed profile (also called a low-pass filtered profile), p(i) is main profile, B is the base length of the moving path, x is the distance and N is the number of samples included in the summation.

Low pass filters such as Eq. 2 ignore the local roughness and produce the global or over all shape of the road which can be used as the defined set-level of the road.

$$p_{fL}(i) = \frac{1}{N}\sum_{j=i-\frac{B}{2\Delta x}}^{i+\frac{B}{2\Delta x}} p(j) \tag{2}$$

After the filtration of data, some important parameters such as Power Spectral Density (PSD) and International Roughness Index (IRI) are defined (Andren, 2005). The former can be obtained using Fourier series and give the density of different functions in different frequencies and the latter show the effect of the vehicle speed.

Direct Method of Generating the Function in Frequency Domain

The experimental method to measure and produce a road profile is a very time consuming and expensive procedure; therefore, mathematical methods are used for this purpose. This method is known as the Direct Method of Generating the Function, in which the PSD represents the weighted effect of the functions that define the road roughness. In this research, PSD of the road roughness have been used and different road types have been modeled based on ISO standard categories (Kropac and Mucka, 2005). For this purpose, the profile for one wheel; independent from other wheels, has been developed. Also, some empirical relationships have been used to predict road profiles for other wheels (Newland, 1984). With the assumption of ergodic profile and using one wheel input and Fast Fourier Transformation, the whole road profile has been obtained (Park *et al*., 2004):

$$Z_r = \sum_{K=0}^{N-1} \sqrt[3]{S_K}\, e^{j\{\phi_K\}} \tag{3}$$

where, S_k is the PSD, k is wave number in rad/sec, $\{\O_k\}$ is a combination of a set of independent and random phase angles which distribute evenly in 0-2π. In this research, using the above equations a computer code has been developed to produce road roughness profiles. For different categories of road profiles, S_o or $S_u(k)$ which rated roughness parameters based on ISO standards and are given in Table 1 and 2.

Since a Multi Purposes Vehicle (MPV) will pass over different road types based on ISO standards (from good urban paths up to very rough off-roads), PSD values and ISO coefficients for medium type road have been selected and are used in this research with 120 mm maximum amplitude (Fig. 5). These results or road profiles have been used as input data to study the dynamic behavior or response of the vehicle.

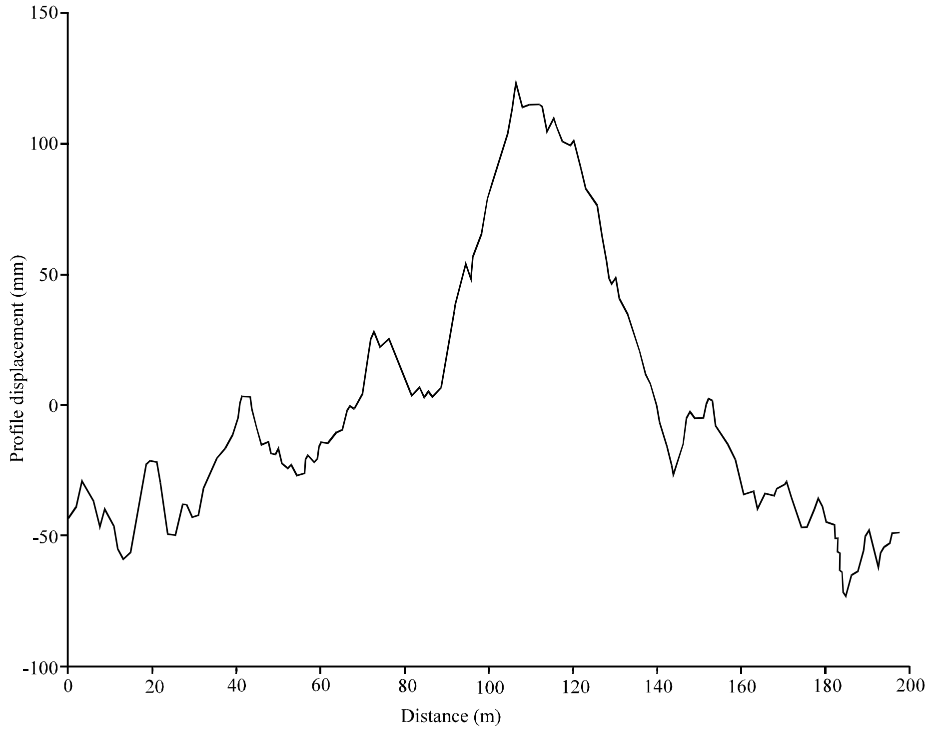

Fig. 5: Road roughness profile for average road type

Direct Function Generation Method in Time Domain

Road roughness can usually be considered as a random stationary phenomenon with a Gaussian probability distribution. Usually the ratio of roughness domain to the road length is expressed as the spectral density of vertical height of the road. Stationary or semi-stationary profiles, in other words those in which the average and co-variance of the samples are equal in different time domains, can be regarded as time independent. Other wise the profile should be regarded as time dependent and is known as ergodic.

While the vehicle moves along the road roughness, inputs for front wheels will or should be imposed to the rear wheels with only a time delay. This time delay is equal to the distance between front and rear wheel (wheel base) divided by the vehicle speed.

The random feature of the vibration due to the road roughness can be approached in different ways. Using the central limit theory, this random phenomenon can be described by the following normal distribution (Zhang *et al.*, 2002):

$$P(x) = \frac{1}{\sqrt{2\pi\sigma}} e^{\left[-\frac{(x-x_0)^2}{\sigma^2}\right]} \tag{4}$$

where, x is the mean value, x_0 is the base value, σ is the standard deviation and P(x) is the distribution function. The limitation to apply this theory, which is known as the Gaussian normal distribution, is that the random parameter would have the same contribution (Zhang *et al.*, 2002). The importance of this distribution is that by knowing the mean and standard deviation, the whole distribution can be defined. In this method, also, a new function called auto correlation function can be defined as:

$$R_u(\tau) = \lim \int_{-\tau/2}^{+\tau/2} [u(t).u(t+\tau)]dt \quad (5)$$

in which the unit step function of u(t) is a function of road roughness, t and σ are time and integration domain respectively. In stationary phenomena, when the mean value is zero, the standard deviation can be obtained using the auto correlation function:

$$R_u(\tau)_{\tau=0} = R_u(0) = \sigma^2{}_u = U^2 = \lim \frac{1}{\tau}\int_{-\tau/2}^{\tau/2} U^2(t)dt \ 11 \quad (6)$$

which is the standard deviation for road roughness. Therefore, the road roughness can be regarded as a Gaussian Distribution Profile, if the global road-shape such us uphill or valleys have been filtered and processed separately. In this way, the amplitude and wave length of the road roughness profiles can be given using spectral density of the profile-height (Zhang *et al*., 2002) given the Fourier transformation of $R_u(\tau)$:

$$S_u(\omega) = \int_{-\infty}^{+\infty} R_u(t)e^{-j\omega t}dt \quad (7)$$

and if t = 0 it will be:

$$R_u(0) = \frac{1}{2\pi}\int_{-\infty}^{+\infty} S_u(\omega)d\omega \quad (8)$$

In Eq. 7, $S_u(\omega)$ or the Fourier transformation function of $R_u(\tau)$ is the power spectral density of the road roughness. Since the random function of u(t) is real and even:

$$R_u(0) = \frac{1}{\pi}\int_0^{\infty} S_u(\omega)d\omega \quad (9)$$

The above equations show that by having the power spectral density of the road roughness, the mean value of the road profile can be obtained by Zhang *et al*. (2002). To carry out the time domain simulation of the road profiles, the PSD of them can be used directly to calculate the PSD of the vehicle response. For this purpose, it is necessary to develop some sample profiles. These profiles, which are single track, can be produced using a target PSD (Park *et al*., 2004; Zhang *et al*., 2002). In addition to this, if other PSDs were available, such as one for road inclination, pair of track or complete profiles can be obtained, which can be used to model the behavior of the whole vehicle (Gillespie, 1993; Wong, 2001).

Having obtained PSD of road roughness, using Eq. 4 or 5, the st andard deviation of road profiles can be generated using direct function generation method in time domain.

The speed of the vehicle is another important factor, which can be defined using International Roughness Index (IRI). IRI is the sum of the vertical displacement of a 1/4 model of a selected vehicle in a defined interval with a specified speed (Newland, 1984). Some local roughness such as pits can also be introduced using:

$$H = \begin{cases} -t/2 & t \le 0.1 \\ -y & 0.1 \le t \le 0.4 \\ [-0.1 + (t+0.4)]/2 & 0.4 \le t \le 0.5 \end{cases} \quad (10)$$

where, y is the depth of pit and t is time. If in this equation ($t-t_{delay}$) is used instead of t, the displacement of the rear wheels would be obtained. $t_{delay} = L/V$ is the delay-time for the rear wheels, where L is the distance between front and rear wheels and V is the vehicle speed.

SOME OF RECOMMENDATIONS FOR POWER SPECTRAL DENSITY

ISO International Standard Recommendation

In 1972 and on the basis of the empirical measurements in Europe for single track spectral density along the secondary roads the following relationships have been proposed by Kropac and Mucka (2005):

$$S_u(K) = \begin{cases} S_u(K_0)\cdot(\frac{K}{K_0})^{-n_1}, \; K \le K_0 \\ S_u(K_0)\cdot(\frac{K}{K_0})^{-n_2}, \; K \ge K_0 \end{cases} \tag{11}$$

In these equations, K is wave number in cycle/m, K_o is the base wave number, $S_u(K)$ is the PSD of the vertical displacement in m^3/cycle, $S_u(K_o)$ is the PSD in K_o. The ISO recommendation for $n_1 = 3$, $n_2=2.25$ and $K_o= 1/2\pi$ are shown in Table 1.

Robson Recommendation

Robson has recommended that the spectral density for vertical roughness and road profile can be defined as below (Gillespie, 1993).

$$S_u(K) = S_0 |K|^{-n} \tag{12}$$

which empirical power n is equal to 2.5 and S_o are given in Table 2.

Sayers Recommendation

Sayers also has proposed a general relationship to give the road roughness (Gillespie, 1993):

$$S_u(K) = \frac{G_a}{(2\pi K)^4} + \frac{G_s}{(2\pi K)^2} + G_e \tag{13}$$

where, G_a, G_s and G_e are constants and K is the wave number.

Table 1: ISO classification of road roughness types and their parameters (Kropac and Mucka, 2005)

Classification and road roughness	$S(K_0)/10^{-6}\,\frac{m^3}{cycle}$
Very good	2-4
Good	8-32
Average	32-128
Poor	128-512
Very poor	512-2048

Table 2: Classification of roughness by Robson and their parameters (Gillespie, 1993)

Classification and road roughness	$S_0/10^{-8}\, m^{0.5} cycle^{1.5}$
Highway	3-50
Main road	3-800
Rural	50-3000

DYNAMIC MODEL OF THE VEHICLE

To apply the road profile data-inputs and to carry out the necessary dynamic analysis, it is necessary to have a model of vehicle for calculating the values of the applied forces and displacements which are applied to the vehicle tires and carried through the suspension system in to the upper h and systems like chassis and its attaching links. Displacement input due to the road surface roughness has been applied to the vehicle wheel and tire and the response of the suspension system as spring and damper reaction forces will be produced because of their stiffness and resilience. These forces will be transferred to the upper hand systems. Ultimately, because of the alternative and repetitive nature of these forces, they cause fatigue in the vehicle parts and in their joints (Fu and Cebon, 2000; Donders ***et al.,*** 2006).

To develop a dynamic model and to find the reaction forces, different methods can be used such as 1/4 vehicle model, 1/2 vehicle model and the full model of vehicle which is the most sophisticated one and takes into account the effect of pitch, roll and bounce caused by passing the vehicle over a road with random roughness profile.

For this reason, in this research a full vehicle model has been used. To make a full vehicle model these steps has been taken:

- Producing a body model
- Introducing the axels, tires, wheels and suspension system
- Applying the longitudinal and cross suspension system links and defining the connections
- Defining the degrees of freedom in connections and applying major items like compliance in linkages and so on

To carry out the dynamic analysis using numerical simulations, because of the large volume of data required for the real and accurate body models, simplified models has been used. However, to obtain an accurate model, the major parameters, such as mass properties and inertia have been defined. Due to the large number of mechanical parts in the real vehicle body, their equivalent mass and inertia has been obtained using Solid Works software and has been introduced in the simple model. Figure 6 shows the 3D model of the off-road vehicle model. Figure 7 shows the body parts and Fig. 8 shows

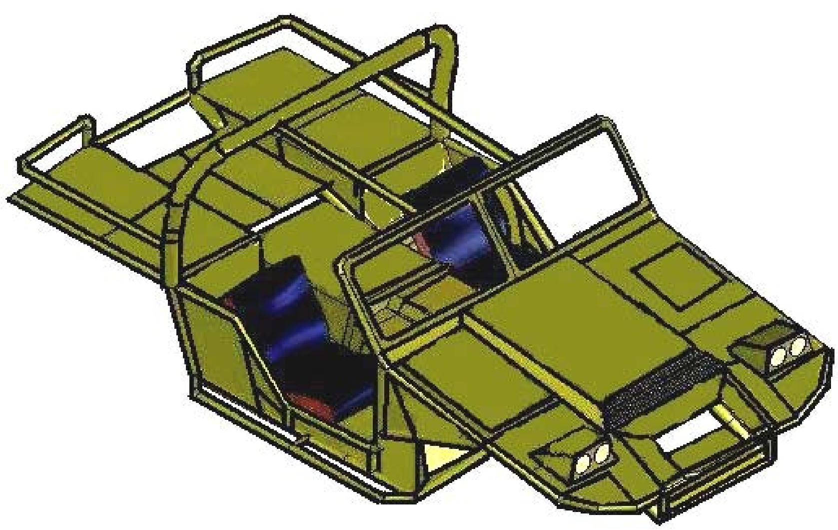

Fig. 6: 3 D model of the off- road vehicle body

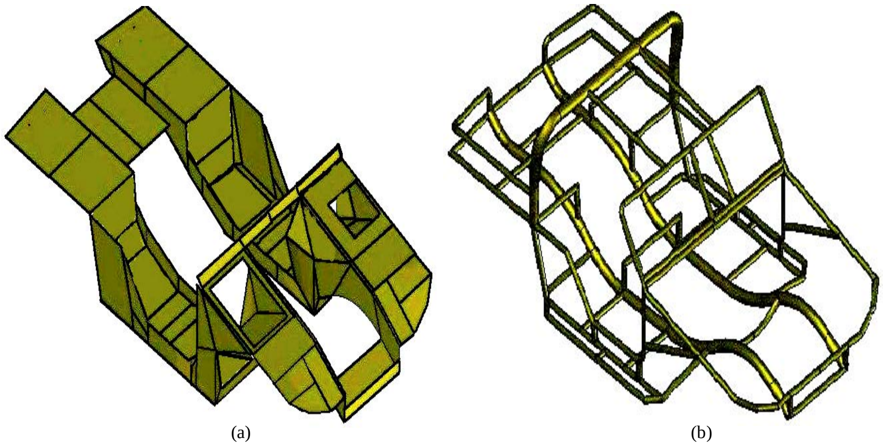

Fig. 7: Model of the body parts

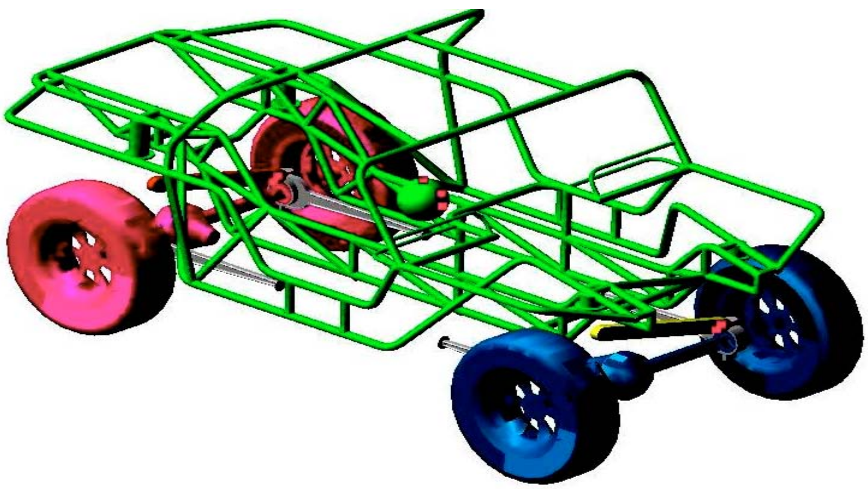

Fig. 8: Suspension system with joining parts

the suspension system together with their joining parts. By introducing the effect of these parts, an accurate but a simple model has been developed to study the dynamic behavior of the vehicle. In addition, the role of other parts, such as tires, suspension system itself and axels have also been taken into account. Also, the damping effect of the tiers has been introduced to the model. For this purpose the well known model of Fiolla has been used (Lee and Kim, 2008). The flexibility of the joining parts was also used to improve the accuracy of the model.

IMPLEMENTATION OF THE ROAD ROUGHNESS PROFILE TO THE DYNAMIC MODEL OF THE VEHICLE

For applying the road roughness conditions on dynamic model, two methods can be used (Benasciutti and Tovo, 2005). First, generating a 3D road surface profile and moving the vehicle on it. In this case the roughness applied to the front and rear wheels or right and left ones would be the same only with a time delay. Another method is using the vibrating table (Shaking table) in which the road roughness can be applied to each one of wheels through four plates of the tables. In shaking table, there is the possibility of defining independent roughness profiles for each one of the wheels. The latter method is more realistic in terms of the random nature of the road profile.

In this research, all the dynamic simulation steps and full vehicle dynamic analysis to obtain the mass accelerations, displacements and forces have been carried out using MSC. ADAMS dynamic simulation computer Code. Also a computer code has also been developed to transform the results from dynamic analysis into frequency domains.

In addition, the optimum values for spring stiffness and damping coefficients of the suspension system have been calculated using a trial and error method (Sun, 2002). For this purpose, five different combinations of the stiffness and damping coefficients have been used (Table 3) based on the manufacturing company's recommendations/limitations. For these combinations, the acceleration of the center of gravity of the suspension system has been calculated and the one which leads to a minimum value has been selected i.e., case study 5 in Table 3. Finally, the results of the analysis of the dynamic model have been obtained, which include the applied forces to the linkage joints of the suspension system.

Calculating the Forces Applied to the Vehicle-chassis and the Fatigue Life of Linkage Joints

At this step, the dynamic behavior of the vehicle has been analyzed and the loads in the weldment joining main suspension system members to the chassis have been obtained. These forces are shown in Fig. 9 and 10. These forces are obtained based on the vehicle velocity of 90 km h^{-1} moving over Average road based on ISO standards (Table 1). Figure 9 and 10 have been selected which show the maximum load applied to the support. These alternative loads will cause fatigue which restricts the life of the suspension system. To calculate the fatigue life of these parts, Finite Element Analysis of the parts which carry the alternative loads has been carried out using ANSYS. Figure 11 shows the Finite Element Mesh of the rack which supports suspension spring. Four parts joined together by welding

Table 3: Different combinations of the stiffness and damping coefficients for the suspension system and its CG acceleration

Case	1	2	3	4	5
Stiffness (K_0) (N m^{-1})	65000	84500	84500	45500	45500
Damping coef. (C_0) NsM	7500	9750	5250	9750	5250
CG acceleration (m sec^{-2})	0.1129	0.1216	0.1358	0.1983	0.09816

Fig. 9: Time-history of the suspension spring load in Newton for front wheel-left

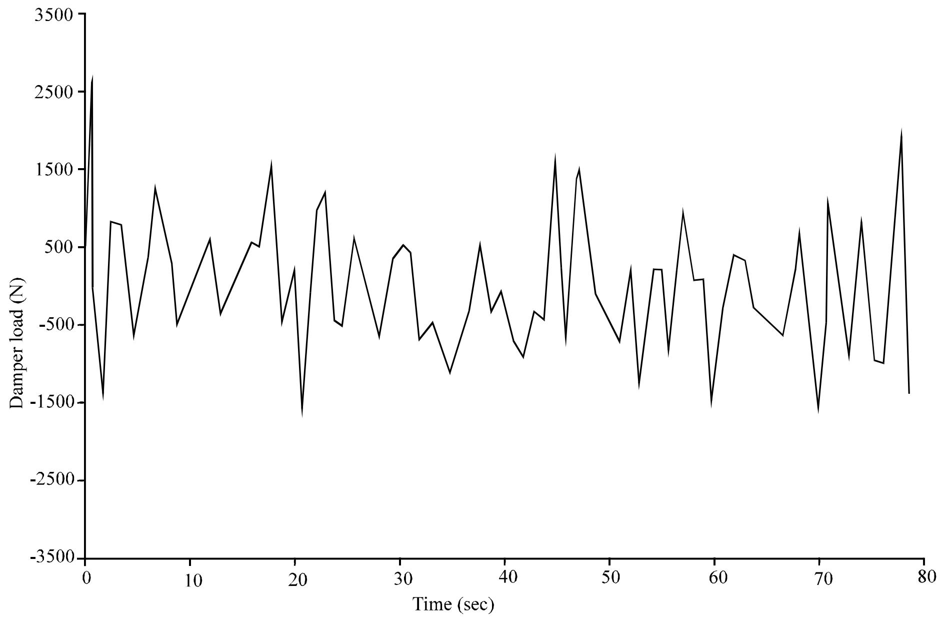

Fig. 10: Time-history of the suspension damper load in Newton for front wheel-left

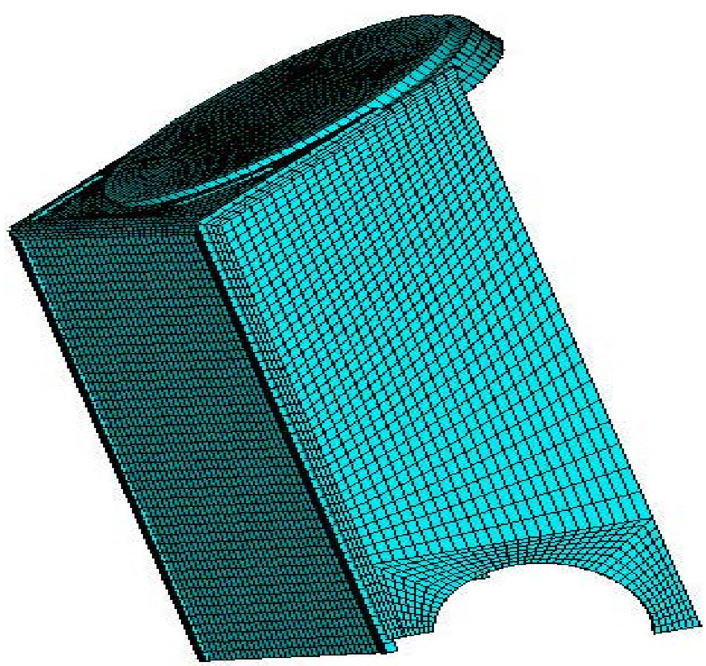

Fig. 11: Finite element mesh of the spring-rack

and make the rack which itself is attached to the chassis by welding (Fig. 12). The local effects of the weldments, have also been taken into account as it is shown in Fig. 13. This can be done by introducing stiffening layers on the weldments (Lee and Kim, 2008).

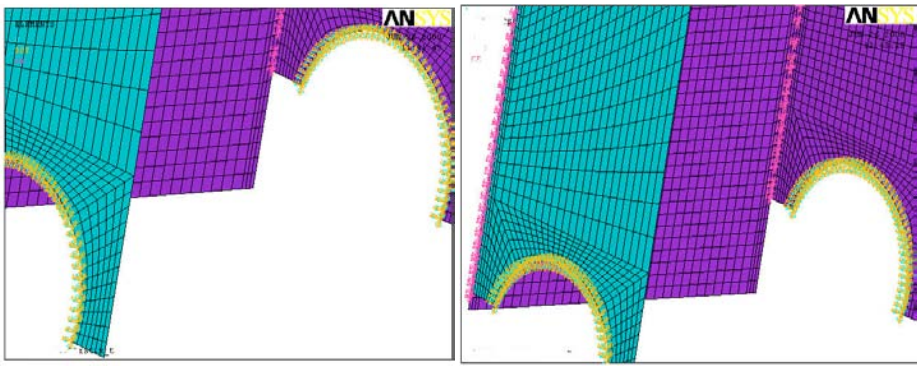

Fig. 12: Boundary conditions for suspension support and welding joints

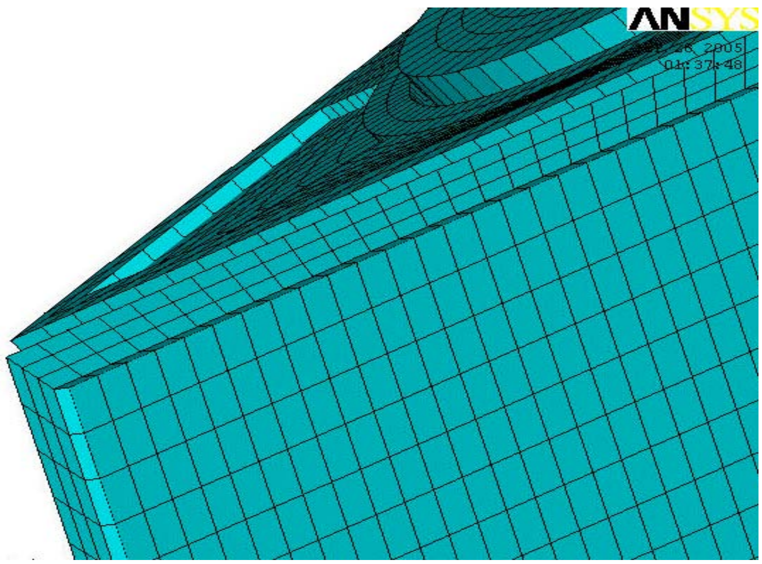

Fig. 13: Local effects of the weldment

The FEM analysis of welding joints can be accomplished in two ways: first, by using Rigid Link or Rigid Beam between joining nodes. In the second method, some equations, which restrict the degrees of freedom of joining nodes, can be used (Lee and Kim, 2008). In this research the latter method has been used, in which the restricting equations of:

$$U_{1x} + U_{2x} = 0.0 \Rightarrow U_{1x} = -U_{2x} \tag{14}$$

have been used. These equations relate the displacement of the joining nodes, where U_i is the displacement in i direction. Figure 14 shows the use of this method. Also, Fig. 15 illustrates the way that external loads have been applied along the co-centre rings on the top of rack. Figure 16 shows the equivalent loads that are applied by spring and damper.

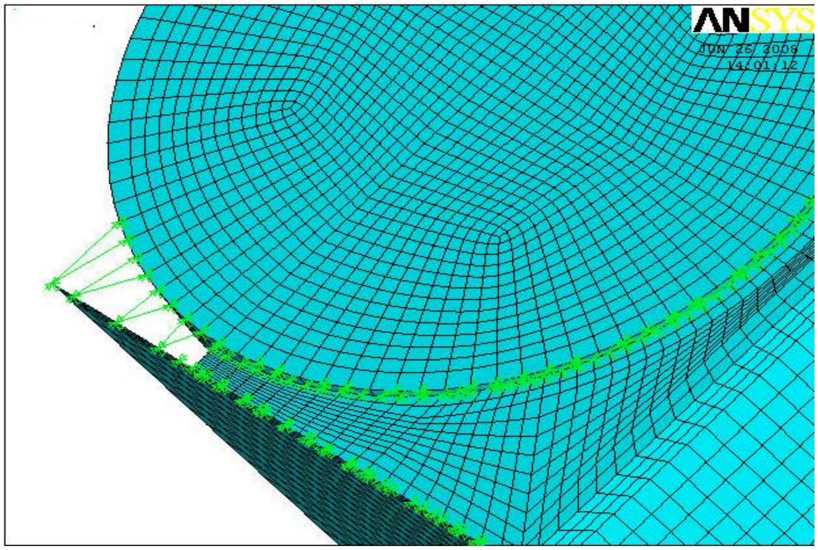

Fig. 14: Restricting the degrees of freedom for joining nodes

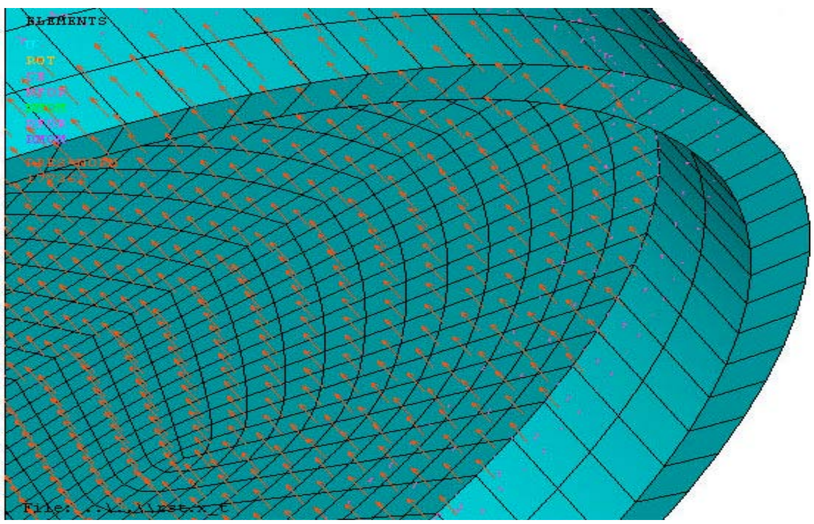

Fig. 15: Distribution of applied forces on the top of the suspension support

The alternating loads have been divided into different time steps such as 16, 40, 80, 100 and 120 steps. Using the FEM analysis, the alternative load has been applied and the time-history of stress level has been calculated and the results are shown in Fig. 17-19. It has been shown that with steps more than 80, the results converge. Therefore, the step value of 80 has been selected with time-step of 0.1 sec for 8 sec of time interval. The time-history of the node with maximum stress is shown in Fig. 19. Also, the stress distribution for the rack is shown in Fig. 20.

The Palm-Green and Miner equation has been used to evaluate the total fatigue life (Radaj and Sonsino, 1998):

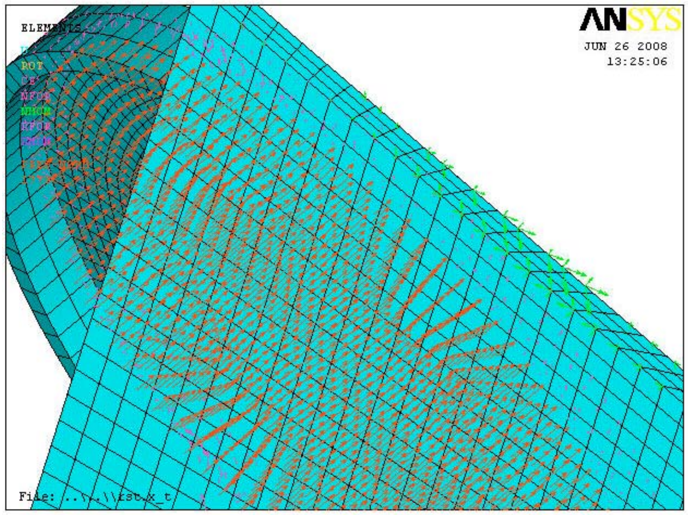

Fig. 16: Distributed equivalent load for spring and damper

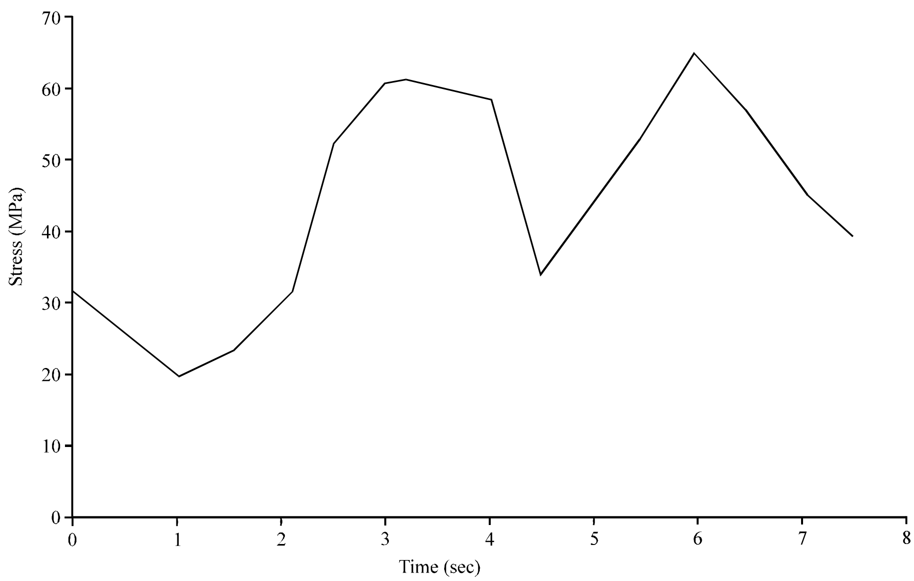

Fig. 17: Time-history of maximum stress for 16 load steps: time in seconds and stress in MPa

$$\frac{n_1}{N_{f1}}+\frac{n_2}{N_{f2}}+....+=\sum\frac{n_i}{N_{fi}}=C \tag{15}$$

$$B_f\sum\frac{n_i}{N_{fi}}=1 \tag{16}$$

where, n_i is the number of cycles of applied stress σ_i, C, which is usually between 0.7-2.2, is chosen to be 1 (Rychlik and Gupta, 2007). B_f is the number of repetition of applied alternative

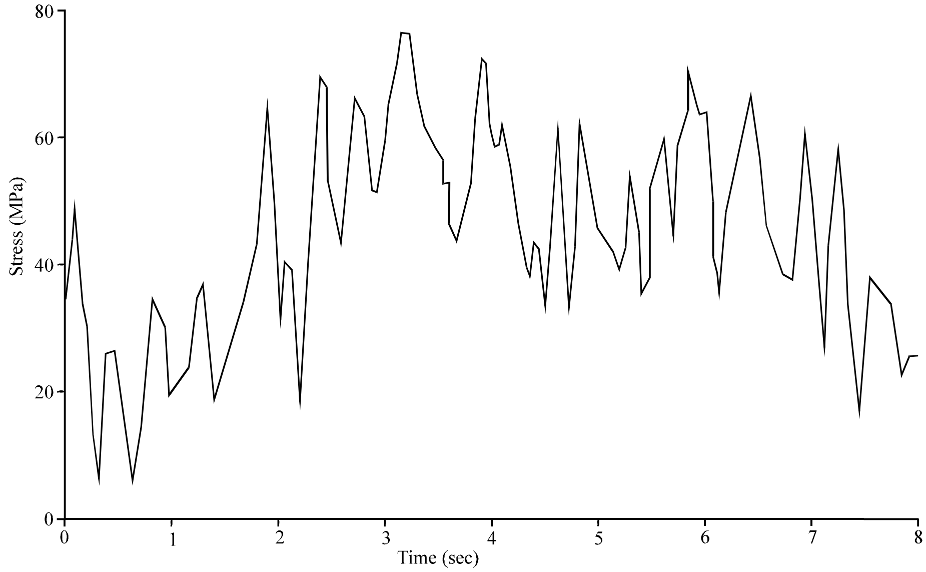

Fig. 18: Time-history of maximum stress for 80 load steps, time in seconds and stress in MPa

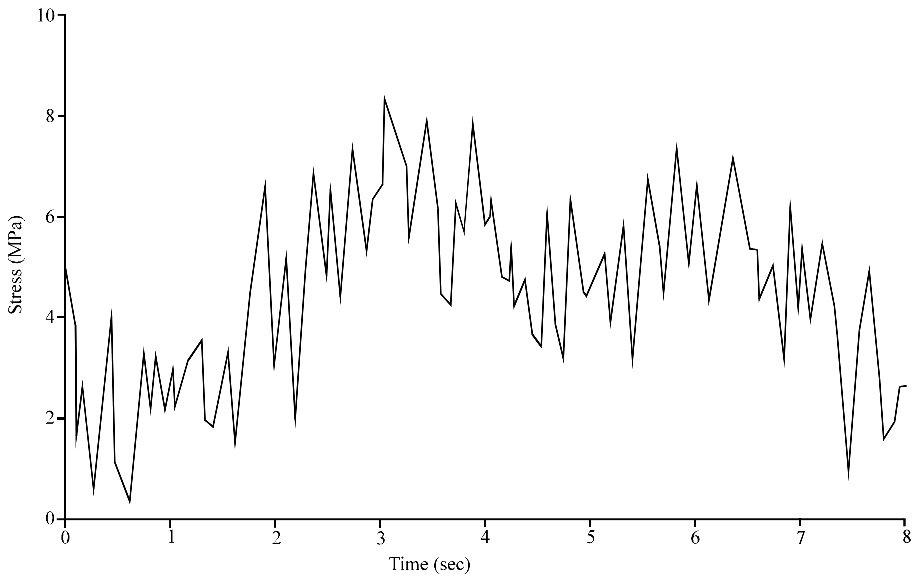

Fig. 19: Time-history of maximum stress for 200 load steps, time in seconds and stress in MPa

loads. To calculate the effect of the accumulated damage due to alternative loads with variable amplitude, the Rain-follow method has been used (Budynas and Nisbett, 2008). Table 4 presents a sample result which is shown in Fig. 21.

For weldments under alternative loads, the initiation of crack is usually known as the fatigue life in aircraft and vehicle body design (Huang *et al.*, 1998). Therefore, Notch Strain Approach can also be used to study the fatigue life. Total number of cycles N_t can be given as:

Table 4: Typical results of load cycle numbering (stresses are in MPa)

σ_{min}	σ_{max}	Äσ	σ_m
11.5	22.70	11.20	17.10
62.6	69.90	7.30	66.25
38.8	39.00	0.20	38.90
66.1	72.90	6.80	69.50
39.3	44.00	4.70	41.65
58.8	71.30	12.50	65.05
61.1	94.80	33.70	77.95
87.1	99.10	12.00	93.10
126	129.00	3.00	127.50
98.6	122.00	23.40	110.30
146	155.00	9.00	150.50
122	124.00	2.00	123.00
87.4	89.30	1.90	88.35
111	115.00	4.00	113.00
141	142.00	1.00	141.50

Table 5: Typical results for calculating fatigue life using *Bf* and rain-flow method (first 16 rows of 80-row table)

N_i Cycles	1/Ni	$\sum_{i=1}^{80} 1/N_i$
2.47079×10^{-22}	4.05404E-09	246667472.7
6.35599×10^{-24}		
3.12273×10^{-40}		
3.21958×10^{-24}		
4.59399×10^{-26}		
1.60495×10^{-21}		
5.50461×10^{-17}		
1.66404×10^{-21}		
1.83764×10^{-27}		
2.19088×10^{-18}		
2.33366×10^{-22}		
2.58606×10^{-29}		
$8.44403\times10E^{-30}$		
2.77836×10^{-26}		
2.81232×10^{-32}		
1.5971×10^{-27}		

$$B_f \sum \frac{n_i}{N_{fi}} = 1 \tag{17}$$

in which N_i is the crack initiation time with the pre assumed initial crack length of a_i=0.25 mm and N_p is the crack propagation period. The crack initiation period can be obtained using Basquin-Morrow equation (Starke *et al*., 2006; Radaj and Sonsino, 1998):

$$\sigma_a = \frac{\Delta\sigma}{2} = (\sigma'_f - \sigma_m)N^b \tag{18}$$

where, σ_a is the amplitude of local stress, $\Delta\sigma$ is the stress range, σ'_f is the coefficient of endurance limit, σ_m is the average of stress and b is the material coefficient. Also

$$N_{per} = \frac{N_i}{J_N} N \leq N_{per} \tag{19}$$

where, N is the number of applied load cycles, N_{per} is the number of allowable cycles, N_i is the number of cycles before crack initiation and J_N is the safety factor. For welding joints the safety factor is usually greater or equals 0.5.

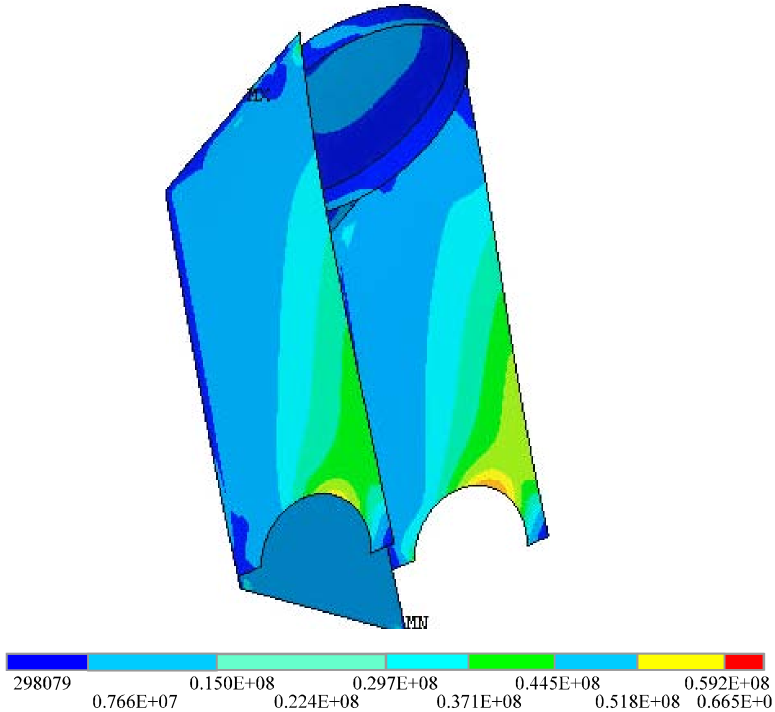

Fig. 20: Stress distribution (Pa) in suspension support showing the point with maximum stress

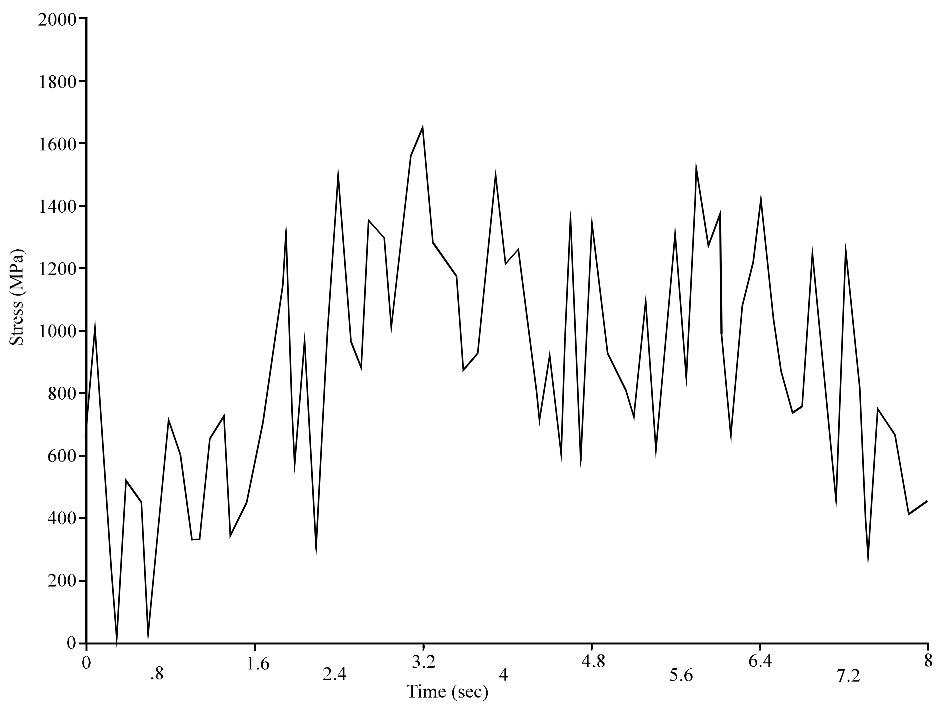

Fig. 21: Time-History of stress (MPa) for the FE analysis due to the dynamic loads

For each case, based on the calculated stress distribution, which is the output of the FEM analysis and according to the number of load cycles, the fatigue life of the welding joint has been calculated using Eq. 15 with C = 1 and the value of B_f has been obtained using:

$$B_f \sum \frac{n_i}{N_{fi}} = C \quad (20)$$

Table 5 shows the result of one case, in which B_f is the maximum allowable number of load cycles which can be applied to the joint. Since, each loading case is happening in 8 seconds, using B_f and the acceptable safety factor of J_N, the fatigue life for the welding joint would be 109630 h.

DISCUSSION

In this study, different methods have been reviewed which can be used to produce road profiles. Although the experimental methods are the most reliable ones, they are very time consuming and expensive, therefore, different mathematical models have been reviewed and Direct Method of Generating the Function due to its accuracy has been selected and used. PSD of the road roughness have been used and different road types have been modeled based on ISO standard categories (Kropac and Mucka, 2005). With the assumption of ergodic profile and using one wheel input and Fast Fourier Transformation, the whole road profile has been obtained by Park *et al*. (2004). A complete dynamic model of the off-road vehicle has been developed using SOLID WORKS and MSC.ADAMS and the applied forces due to the road roughness have been calculated taking into account the dynamic characteristics of the main parts of the chassis and the suspension system. An optimum combination of the stiffness and damping coefficients of the suspension system has been obtained to produce minimum forces applied to the system.

CONCLUSION

In this study, a comprehensive methodology has been developed to study the fatigue life of the welding joints in the vehicle body (chassis) using numerical analysis. This method has been used to design and predict the fatigue life of the weldments in the support of a suspension system for an off-road vehicle. Firstly, a set of road profiles have been produced using Direct Method of Generating the Functions in Frequency Domain using Matlab Code. The output results have been used as input data for the FEM analysis of the welding joints to calculate the alternative stress distribution. Using the Rain-flow method, the alternative stresses have been used to calculate the fatigue life of the welding joints. The maximum stress is on the upper support of the chassis, which is connected by seam welds. It has been shown that the welding joints have a fatigue life of 109630 h. The conclusions obtained can be summarized as follows:

- The results show that the developed method can be used to study the dynamic forces which are applied to the vehicle-body or parts due to the road roughness
- This method, also allows the designer to calculate an optimum combination of the suspension system characteristics in order to minimize the applied forces
- For a particular MPV vehicle an optimum combination of the stiffness and coefficients of the suspension system has been obtained to produce minimum forces to the suspension system
- A critical location with maximum stresses in the support of a suspension system has been identified. The results show that the maximum stress is on the upper support
- These results provide a basis for evaluating fatigue damage and predicting the remaining life of the suspension support

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