The needs to investigate the effect of road surface vibrations to the fatigue life of a coil spring

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Abstract. The study aims to investigate the effect of vibrations on the fatigue life of an automotive coil spring. Acceleration signals were measured at a coil spring while the passenger car was being driven on road surfaces. Using a developed multi-body dynamic simulation, the acceleration signals were transformed into strain signals, and then, they were utilized as the input for performing fatigue tests. From the results, it was obtained that the rough road surface provided a shorter fatigue life, which was 64.5%. It indicated that rough road surfaces significantly effect to component life.

1. Introduction

Many automotive components, which are often subjected to variable driving conditions, can be associated with failure caused by fatigue during their operations. The physical conditions of road surfaces and the different velocities of the car are the main factors contributing to failure [1]. The control and stability of a car entirely depend on the collision and the friction between road surfaces and tyres [2-4], which are uncertain and can change quickly and extremely. This dynamic interaction gives a certain amount of vibration causing problems concerning car components and the ride quality. This vibration acts as a catalyst to speed up the crack initiation interfacing the function of the components and give a great impact on the performance of the car; contributing to mechanical failure due to fatigue; as the components are exposed to cyclic loads.

In such a case, vibrations are unavoidable and their isolation advantageous [5]. The reduction of vibrations is not only to provide comfort to the passenger(s) but also as important, to help in reducing the probability of fatigue failure at car components; which results in less cost and reduces the possibility of a fatal accident from occurring. Thus, a system absorbing vibrations of the vertically accelerated wheel is needed. The best system to resolve this problem is the suspension system. The suspension system is a mechanical device in which its main function is to minimize vertical displacements and accelerations [6] and maximize tractions between road surfaces and tyres to provide steering stability [7]. It permits the control arm and wheel to move up and down. It operates under multi-axial service loadings, identified as the existence of more than one principal stress.

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large. Vibrations are absorbed by the tyres, the springs in a seat and the coil spring in a suspension system. The tyres and seat springs absorb a little vibration, while the coil spring does the rest to provide convenience to the passenger(s) and ensure driving safety. Passenger(s), however, will still feel a little vibration at a certain level. In this case, the coil spring has a forced vibration, which occurs when energy is continuously added to the component by applying the oscillatory force at some forcing frequency. A significant amount of shock affects the coil spring, and at higher rates than other car components. Thus, coil springs play a vital role in the failure of car structures since they withstand the majority of resulting vibrations from road surfaces; contributing to fatigue failure. Problems that may occur to a coil spring are because of stiffness and low fatigue life [9].

According to Das et al. [10], the existence of residual stresses on a component affects its durability. If the resulted stresses are repeated in a certain period, they increase the probability of fatigue failure for the component. Thus, the objective of the current study is to investigate the effect of vibrations for an automotive coil spring, driven on different road surfaces. It was hypothesized that a smooth road surface gives small vibrations to the coil spring, while rough road surfaces are the largest contributor to the occurrence of vibrations and are considered to contribute substantially to fatigue failure.

2. Material and method

2.1. Acceleration signal acquisition

The frontal coil spring of an automobile with 1,300 cc capacity was used as a case study for data measurement. It uses a passive McPherson strut suspension system with the spring stiffness, and the damping ratio are 18,639 N/m and 0.95, respectively. The specifications were from the car manufacturer. The selected material for the simulation was the SAE5160 carbon steel since it is commonly employed in automotive industries for fabricating a coil spring [11]. Its monotonic mechanical properties of this material are tabulated in Table 1.

| Properties                        | Values |
|-----------------------------------|--------|
| Ultimate tensile strength (MPa)   | 1,584  |
| Material modulus of elasticity (GPa) | 207    |
| Fatigue strength coefficient (MPa) | 2,063  |
| Fatigue strength exponent         | -0.08  |
| Fatigue ductility exponent        | -1.05  |
| Fatigue ductility coefficient     | 9.56   |
| Cyclic strain hardening exponent  | 0.05   |
| Cyclic strength coefficient (MPa)  | 1,940  |

A linear static analysis was performed for determining the stress distributions at the coil spring. A uni-axial force of 3,600 N was applied on the bottom of the component model and the upper was fixed, considering the car weight of 10,600 N as well as the four passengers and the load carried of 3,800 N. The total force was divided by four because the car weight and the passenger(s) were assumed to distribute to four springs uniformly. An acceleration signal acquisition was performed placing an accelerometer at the component. The position of the accelerometer installation was selected based on the possibility of higher stress area [13] based on the finite element analysis. The frequency of the acceleration signals was assumed to be sampled at 500 Hz. Collecting load histories at a frequency of 500 Hz is sufficient to detect and capture all damaging load cycles [14]. After installing the equipment, the car was then driven on the highway, urban and rural road surfaces at velocities of
70 km/h to 80 km/h, 30 km/h to 40 km/h and 20 km/h to 40 km/h, respectively. The road surfaces were chosen because they are the most commonly used in Malaysia. The velocities above were matched with an average car speed when driven on specified road conditions [4, 15-16].

2.2. Strain signal development
The Hooke law, developed by R. Hooke in 1660 states that the displacement $x$ of an elastic object, such as coil spring, is directly proportional to the spring force $P_s$ applied to it. That is:

$$P_s = kx$$  \(1\)

Where $k$ is the spring stiffness. Assuming the elastic limit is not exceeded, a graph of force against extension produces a straight line that passes through the origin. The gradient of the line is the spring stiffness. While carrying out an experiment, the spring stiffness is found to be different depending on the objects and materials — the greater the spring stiffness, the stiffer the spring. The softer the spring and the greater the mass, the longer the period of vibration [6, 17].

The spring deflects an amount proportional to the force required to accelerate the spring system. The spring stiffness provides a restoring force to move the mass back to equilibrium, and dampers oppose any displacement away from equilibrium. Damping is usually the result of viscous or frictional effects. If such force is also proportional to velocity, the damping force $P_d$ may be related to the velocity $\dot{x}$ by:

$$P_d = d\dot{x}$$  \(2\)

and the damping coefficient $d$ is expressed by:

$$d = 2\zeta \sqrt{km}$$  \(3\)

where $m$ is the mass and $\zeta$ is the damping ratio.

Vibrations are caused by imperfections, especially involving operation from the car tyres through rough road surfaces [18]. Since the vibration is defined as oscillatory motion resulted from varying force to a structure, it involves a change of position or displacement. Velocity is the rate of change in displacement with respect to time that can be described as the slope of the displacement curve. Similarly, acceleration is the rate of change of velocity with respect to time or the slope of the velocity curve [19]. Displacement, velocity, and acceleration are also referred to as a shock or vibration, depending on the waveform of the forcing function that causes the acceleration. A forcing function that is periodic generally results in an acceleration that is analyzed as a vibration. On the other hand, a force input having a short duration and a large amplitude would be classified as a shock load [20].

A method that has been used to classify a vibration is based on its degree of freedom, which is the number of independent kinematic variables to describe the movement of a system. Fig can characterize a mass-spring-damper system with a single degree of freedom can be characterised by Fig. 2. Single degree of freedom system was considered in the study assuming the vibration act in each suspension without affecting another suspension. This shows the relationship among four basic components of the dynamic system, which are mass, resistance (spring), energy dissipation (damper) and applied force [19, 21].

According to the Newton second law of motion [22], all forces have both magnitude and direction. As the mass-spring-damper system moves in response to an applied force, forces induced are a function of both the applied force and the motion in the individual components. The governing equation of this system can be derived from the law as:
where $P$ is the applied force and $P_i$ is the inertial force, which is a force opposite in direction to an accelerating force acting on a component, and is equal to:

$$P_i = m\ddot{x}$$

(5)

Where $\ddot{x}$ is the acceleration. Submitting Eqs (1), (2) and (5) into (4), the equation of motion may then be expressed as:

$$m\ddot{x} + d\dot{x} + kx = P$$

(6)

Each overdot denotes a time derivative. The equation describes the behavior of a physical system of its motion as a function of time. The spring and the damping forces are proportional to the displacement and velocity, respectively, whereas the inertial force is dependent on the acceleration. Considering the free-body diagram in Fig. 1, the spring and the damping forces must balance the inertial force for the mass [20].

Motivated by the success of the researches related to automotive suspension system simulations [4-6, 23-25], the study used the equation of motion as the basic formula to develop a mathematical expression for generating strain signals. Eq. (6) Represents the equation of motion for a damped forced vibration. Typically, it is separated into internal and external forces. Internal forces are found on the left side of the equation, and external forces are specified on the right side. External forces induced from engine operations, tyres and the like influence the simulation results, however, they were not considered due to limitations observed in the equipment and the multi-body dynamic software package. A tyre force, for example, is measured during the prototype phase and is mostly estimated using tyre models and auxiliary sensors. The measurement of a tyre force involves many sensors to be placed in many locations since the force does not always flow in the same way for different maneuvers.

Thus, the equation of motion became a damped free vibration, generated as follows:

$$m\ddot{x} + d\dot{x} + kx = 0$$

(7)

Thus, the displacement $x$ is derived as:

$$x = \frac{-m\ddot{x} - d\dot{x}}{k}$$

(8)
It is known that the displacement has a strong relationship to the strain axially loaded. The strain $\varepsilon$ is expressed as the change in length $\Delta L$ of the original unloaded length $L_0$ of a component:

$$\varepsilon = \frac{\Delta L}{L_0} \quad (9)$$

For obtaining the change in length, the original unloaded length of the component is measured. Different numbers of slotted masses are added to the component, and its new length is measured each time. When the component is stretched, the increased length is called its extension. The extension or the change in length $\Delta L$ is the final length $\ell$ minus the original unloaded length $L_0$, and then:

$$\varepsilon = \frac{\ell - L_0}{L_0} \quad (10)$$

In this case, the change in length $\Delta L$ is equal to the displacement $x$, so:

$$\varepsilon = \frac{x}{L_0} \quad (11)$$

Then, the displacement $x$ is:

$$x = \varepsilon L_0 \quad (12)$$

Substituting Eq. (12) into (8), the strain $\varepsilon$ is resulted as:

$$\varepsilon L_0 = -\frac{m \ddot{x} - d \dot{x}}{k} \quad (13)$$

Acceleration can be determined from differentiation of velocity, which the velocity is directly measured by mechanical means only over very short periods or small displacement, due to limitations in transducers. Alternatively, if the acceleration of a rigid body is measured at identifiable time intervals, velocity can be determined through integration of the time-dependent acceleration [20]. Thus, the equation solves for the strain $\varepsilon$ by:

$$\varepsilon = -\frac{m \ddot{x} - d \dot{x}}{kL_0} \quad (14)$$

2.3. Fatigue test
The reason to perform variable amplitude loading fatigue tests in the study was to investigate the coil spring fatigue life caused by road surface vibrations. The major reason for carrying out a variable amplitude loading fatigue test [26-27] was the fact that a prediction of fatigue life under this complex loading is impossible by any hypothesis [28-29]. The fatigue testing specimens were designed according to ASTM E606-92 [30], as shown in Fig. 2. The geometry and the dimensions of the specimens were specified along with appropriate grips for the fatigue testing machine used. The specimens were fabricated using computer numerical controls and grinding machines and were mechanically polished [31] to ensure complete removal of machining marks in the testing section. Sandpaper with a grit size of 240 was used to give the rounded shape, and the finer 1,500 grades gave a suitable surface finish. Since fatigue cracks begin to occur at irregular surfaces, the condition of a
specimen surface is important in fatigue tests. It reduced stress concentration to prevent crack initiation at those sites. The tests were performed at the room temperature, which was 20 °C [30].

![Image](image1)

**Figure 2.** A specimen for the fatigue test: (a) specimen geometry and dimensions in millimeter, (b) photograph of the specimen

Strain signals were converted into the stresses using the Ramberg-Osgood equation [32]. In its original form, the equation is described as:

\[
\varepsilon = \frac{\sigma}{E} + K \left( \frac{\sigma}{E} \right)^n
\]  

(15)

where \(\sigma\) is the stress, \(E\) is the material modulus of elasticity, \(K'\) is the cyclic strength coefficient and \(n'\) is the cyclic strain hardening exponent. The stresses could then be converted into the loads as the input for the fatigue testing machine using the following equation:

\[
\sigma = \frac{P}{A_0}
\]  

(16)

where \(A_0\) is the original unloaded cross-sectional area. The cross-sectional area for the equation was 18 mm coming from the area of gauge length of the specimens.

Variable amplitude loading fatigue tests were performed, where frequency at 100 Hz [33-34] was selected. Increasing the frequency was done to accelerate the tests. There is no effect of higher frequency on crack initiation [33, 35]. The influence of frequency on a fatigue testing result is often considered minimal for frequencies reaching 1,000 Hz [34].

3. Results and discussion

The measurement of an acceleration signal is required for a variety of purposes. It is an interesting fact that a lot of random vibrations that occur should have a Gaussian distribution. Accelerations occur around a fixed point and have a zero-mean over time [36], as shown in Fig. 4. The acceleration signals produced were a variable amplitude loading because the component experienced various amplitudes. The total recorded length was each 60 seconds for 30,000 data points, which were sufficient for evaluating fatigue life. The highway acceleration signal provided the lowest amplitude range. Higher amplitude range could be seen in the urban and rural acceleration signals because their surfaces were uneven. Differences of the acceleration amplitude range could be correlated to the road surface profiles.

Almost all fatigue loading histories collected in engineering practices contain a lot of noise [37-38]. If the signal trends are not removed from the input signal, useful information is corrupted such that the
working state cannot be recognized. Thus, it is an important task that the signal trends are extracted and separated from the noise during signal processing [39-43]. However, in the strain signals shown in Fig. 3, noise at higher frequencies contained in the acceleration signals had been removed in the strain signals. It gave an additional advantage for the simulation since the strain signals did not need to be filtered before analyzing them. Referring to Eq. (14), the strain is parallel to the acceleration. The acceleration with higher amplitudes produced the strain with higher amplitudes as well. The urban and rural strain signals gave higher amplitudes than the highway strain signal did because they were produced from the acceleration signals measured on uneven surfaces. These indicated that the urban and rural road surfaces gave higher vibration energy since they also had higher strain amplitude.

Based on the fatigue testing results, the highway strain signal required at least 1,222 reversals of blocks until failure and the time required was more than 101.8 hours. The time needed to perform the fatigue test for the rural strain signal was 81.5 hours, involving 978 reversals of blocks. The time was reduced by 20 % comparing to the highway strain signal. Furthermore, only 36.2 hours was needed to perform the tests using the urban strain signal, involving 434 reversals of blocks, or 55.6 % and 64.5 % faster than the time needed for the rural and highway strain signals, respectively. The urban strain signal provided the shortest testing time since it had the highest strain amplitudes.

These findings were similar to previous studies. After analyzing more 3,000 repaired cars, [44] concluded that automotive suspension components must be replaced quite early, after about five years or with a traveling distance of 73,500 km. According to data of the Ministry of Transport of United Kingdom, suspension components recorded a high fault, which was 13.18 % of 24.2 million vehicle tests [45]. Holes, bumps, turns, brakes, accelerations, speed changes and the effects of an uneven patch of tars on a road surface are examples of a non-stationary vibration and are factors that cause higher accelerations. These features support the potential for shorter fatigue life. It indicated that the urban

![Image](image_url)
and rural road surfaces provided a shorter useful life compared to the highway road surface since the roads had a rough surface.

4. Conclusion
The effect of vibrations to a coil spring was investigated. For the highway road surface, the acceleration received by the component contained a lot of lower acceleration amplitudes, with uniform characteristics because the road had a similar surface. In the acceleration trend from the urban and rural road surfaces, the acceleration signals revealed parts with higher amplitudes because the car was driven on a less flat surface. It indicated that the road surfaces caused stronger vibrations to the vehicle. It concluded that rough road surfaces contribute to shorter fatigue life.

References
[1] N S Ottosen, R Stenström, M Ristinmaa 2008 Continuum approach to high-cycle fatigue modeling International Journal of Fatigue 30 996-1006
[2] C Ferreira, P Ventura, R Morais, A L G Valente, C Neves, M C Reis 2009 Sensing methodologies to determine automotive damper condition under normal vehicle operation Sensors and Actuators A: Physical 156 237-244
[3] X Yang, Z Wang, W Peng 2009 Coordinated control of AFS and DYC for vehicle handling and stability based on optimal guaranteed cost theory Vehicle System Dynamics 47 57-79
[4] S Lajqi, S Pehan 2012 Designs and optimizations of active and semi-active non-linear suspension systems for a terrain vehicle Journal of Mechanical Engineering 58 732-743
[5] G Priyandoko, M Mailah, H Jamaluddin 2009 Vehicle active suspension system using skyhook adaptive neuro active force control Mechanical Systems and Signal Processing 23 855-868
[6] A Tandel, A R Deshpande, S P Deshmukh, K R Jagtap 2014 Modeling, analysis and PID controller implementation on double wishbone suspension using SimMechanics and Simulink Procedia Engineering 97 1274-1281
[7] N Singh 2013 General review of mechanical springs used in automobiles suspension system International Journal of Advanced Engineering Research and Studies
[8] B G Scuracchio, N B de Lima, C G Schön 2013 Role of residual stresses induced by double peening on fatigue durability of automotive leaf springs Materials and Design 47 672-676
[9] M I Z Abidin, J Mahmud, M J A Latif, A Jumahat 2013 Experimental and numerical investigation of SUP12 steel coil spring Procedia Engineering 68 251-257
[10] S K Das, N K Mukhopadhyay, B R Kumar, D K Bhattacharya 2007 Failure analysis of a passenger car coil spring Engineering Failure Analysis 14 158-163
[11] Y Prawoto, M Ikeda, S K Manville, A Nishikawa 2008 Design and failure modes of automotive suspension springs Engineering Failure Analysis 15 1155-1174
[12] nCode, ICE-flow 4.1: GlyphWorks Worked Examples, nCode International, Ltd., Sheffield, 2007
[13] B. -Y. He, S. -X. Wang, F. Gao 2010 Failure analysis of an automobile damper spring tower Engineering Failure Analysis 17 498-505
[14] M Haiba, D C Barton, P C Brooks, M C Levesley 2003 The development of an optimization algorithm based on fatigue life International Journal of Fatigue 25 299-310
[15] S Ilic 2006 Methodology of Evaluation of In-Service Load Applied to the Output Shafts of Automatic Transmissions, Ph.D. Thesis, The University of New South Wales
[16] W Tong, K H Guo 2012 Simulation testing research on ride comfort of vehicle with global-coupling torsion-elimination suspension Physics Procedia 33 1741-1748
[17] A N Thite 2012 Development of a refined quarter car model for the analysis of discomfort due to vibration Advances in Acoustics and Vibration
[18] İ Eski, Ş Yildirim 2009 Vibration control of vehicle active suspension system using a new robust neural network control system Simulation Modelling Practice, and Theory 17 778-793.
[19] MSC NASTRAN, Dynamic Analysis User's Guide, MSC Software Corporation, Santa Ana, 2012.
[20] R S Figliola, D E Beasley 2011 Theory and Design for Mechanical Measurements, 5th Ed., John Wiley & Sons, Inc., New Jersey.
[21] S G Kelly 2012 Mechanical Vibrations: Theory and Applications, Cengage Learning, Stamford.
[22] S I Newton 1846 The Mathematical Principles of Natural Philosophy, Daniel Adee, New York.
[23] V Goga, M Kľúčik 2012 Optimization of vehicle suspension parameters with use of evolutionary computation Procedia Engineering 48 174-179.
[24] E Alvarez-Sánchez 2013 A Quarter-car suspension system: car body mass estimator and sliding mode control Procedia Technology, 7 208-214.
[25] H Zhang, X Zhang, J Wang 2014 Robust gain-scheduling energy-to-peak control of vehicle lateral dynamics stabilization Vehicle System Dynamics 52 309-340.
[26] M Aykan, M Çelik 2009 Vibration fatigue analysis and multi-axial effect in testing of aerospace structures Mechanical Systems, and Signal Processing 23 897-907.
[27] A Nieslony, M Böhmi 2013 Mean stress effect correction using constant stress ratio S-N curves International Journal of Fatigue 52 49-56.
[28] C M Sonsino 2007 Fatigue testing under variable amplitude loading International Journal of Fatigue 29 1080-1089.
[29] A L M Carvalho, J P Martin, H J C Voorlwad 2010 Fatigue damage accumulation in aluminum 7050-T7451 alloy subjected to block programs loading under step-down sequence Procedia Engineering 2 2037-2043.
[30] ASTM E606-92 Standard Practice for Strain-Controlled Fatigue Testing, ASTM International, West Conshohocken, 1998.
[31] ASTM E112-96 Standard Test Methods for Determining Average Grain Size, ASTM International, West Conshohocken 2004.
[32] W Ramberg, W R Osgood 1943 Description of stress-strain curves by three parameters National Advisory Committee for Aeronautics.
[33] A Hosoi, K Takamura, N Sato, H Kawada 2011 Quantitative evaluation of fatigue damage growth in CFRP laminates that changes due to applied stress level International Journal of Fatigue 33 781-787.
[34] P Schaumann, S Steppeler 2013 Fatigue tests of axially loaded butt welds up to very high cycles Procedia Engineering 66 88-97.
[35] S Kovacs, T Beck, L Singheiser 2013 Influence of mean stresses on fatigue life and damage of a turbine blade steel in the VHCF-regime International Journal of Fatigue 49 90-99.
[36] J Wijker 2009 Random Vibrations in Spacecraft Structures Design: Theory and Applications, Springer Science+Business Media B.V., New York.
[37] J Lin, L Qu 2000 Feature extraction based on Morlet wavelet and its application for mechanical fault diagnosis Journal of Sound and Vibration 234 135–148.
[38] P Ren, Z Zhou 2014 Strain response estimation for the fatigue monitoring of an offshore truss structure Pacific Science Review 16 29-35.
[39] Y.-T. Sheen 2009 On the study of applying Morlet wavelet to the Hilbert transform for the envelope detection of bearing vibrations Mechanical Systems and Signal Processing 23 1518-1527.
[40] H Chen, M J Zuo, X Wang, M R Hoseini 2010 An adaptive Morlet wavelet filter for time-offlight estimation in ultrasonic damage assessment Measurement 43 570-585.
[41] W Su, F Wang, H Zhu, Z Zhang, Z Guo 2010 Rolling element bearing faults diagnosis based on optimal Morlet wavelet filter and autocorrelation enhancement Mechanical Systems and Signal Processing 24 1458-1472.
[42] B Tang, W Liu, T Song 2010 Wind turbine fault diagnosis based on Morlet wavelet transformation and Wigner-Ville distribution Renewable Energy 35 2862-2866.
[43] H Li, P Li, S.-L J Hu 2012 Modal parameter estimation for jacket-type platforms using noisy
[44] L Roman, A Florea, I I Cofaru 2014 Software application for assessment the reliability of suspension system at OPEL cars and road profiles Fascicle of Management and Technological Engineering 1 289-294

[45] M Hamed, B Tesfa, F Gu, A D Ball 2014 Vehicle suspension performance analysis based on full vehicle model for condition monitoring development Proceedings of VETOMAC X 23 495-505