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SIDE FORCE ANALYSIS OF SUSPENSION STRUT UNDER VARIOUS LOAD CASES

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Abstract

This paper presents the side forces extraction of a Macpherson suspension strut under extreme load cases using multibody dynamics model. Inevitable side forces subjected to the Macpherson suspension system during vehicle driving cause the damper rod and top mounting failure. Bending moment generated could increase the friction of damper piston and inner tube and simultaneously decrease the ride comfort of the vehicle, or in a more severe condition, failure occurs. In this study, the side forces magnitude subjected to the Macpherson strut under various severe load events were obtained through a quasi-static multibody dynamics half vehicle model simulation. Outcomes of the multibody simulation that showing the forces exerted on the suspension Macpherson strut were derived into three axes which were vertical, lateral and longitudinal. The lateral and longitudinal side forces on the strut were highest during the pothole striking event which achieved 11052 N. The extracted force provided useful information for suspension linkages design and damper friction analysis to prevent failure.

Keywords: Load cases; multibody dynamics, suspension strut, side forces

Abstrak

Kertas kerja ini membentangkan pemerolehan beban sisi pada topang ampaian Macpherson di bawah beban lampau dengan menggunakan model jasad berbilang dinamik. Beban sisi yang tidak dapat dielakkan dikenakan pada sistem ampaian semasa pergerakan kereta menyebabkan kegagalan pada rod peredam dan cagak atas. Momen lentur yang dijana akan meningkatkan geseran pada omboh peredam serta tiub dalaman dan seterusnya mengurangkan keselesaan penunggangan kereta. Pada keadaan yang lebih serius, kegagalan akan berlaku. Dalam kajian ini, magnitud beban sisi yang dikenakan pada topang Macpherson akibat daripada pelbagai keadaan beban yang serius telah diperolehi melalui simulasi kuasi-statik model jasad berbilang dinamik separuh kereta. Hasil simulasi jasad berbilang menunjukkan beban yang dikenakan pada topang ampaian Macpherson telah diterbitkan kepada tiga paksi, iaitu menegak, sisi dan membujur. Beban sisi dari paksi sisi dan membujur pada topang adalah paling tinggi semasa peristiwa pelanggaran lubang sehingga mencapai 11052 N. Beban sisi yang diperoleh memberikan maklumat yang penting untuk reka bentuk rangkaian sistem ampaian dan analisis geseran peredam bagi mengelakkan berlakunya kegagalan.

Kata kunci: Peristiwa beban; jasad berbilang dinamik, topang ampaian, beban sisi

1.0 INTRODUCTION

Suspension system of a vehicle plays an important role in absorption of road disturbances during vehicle operating conditions. The kinematics of a suspension system affects the path of the wheel relative to the vehicle body and thus influences the system performances [1]. In most of the passenger car, Macpherson type suspension strut is extensively used as front suspension system. The Macpherson strut is a type of automotive suspension system that consists of a coil spring placed together with a shock absorber through the spring seat [2]. This type of suspension is well-known of its simple structure and low manufacturing cost. However, Macpherson strut promote the generation of side forces which are detrimental to the damper pairs [3]. In side force analysis, multibody dynamics analysis is a popular method to identify the magnitude of side force.

Multibody dynamics (MBD) analysis consists of rigid bodies or links that connected to each other via joints that restrict their relative motion [4]. In automotive applications, multi-body relevance finite element analysis of automotive suspension components have been extensively investigated, such as leaf springs [5], coil springs [3, 6] and torsion beam suspension systems [7]. The nonlinear leaf spring finite element model beam elements developed by Omar et al. [8] were used in vehicle dynamic simulations. The reduced order model and component mode synthesis method can effectively simulate the leaf spring dynamics in shorter time. Liu et al. [3] utilized both finite element and multi-body dynamic to optimize a coil spring and remarkably reduced the side force without compromised the system performance. Ryu et al. [6] proposed S-type coil spring design to reduce vehicle side force through multi-body dynamic simulation. For coil spring analysis, it is crucial to indentify the side forces in order to prevent failure.

Recently, Zhu et al. [9] proposed a suspension strut side forces simulation on L Type Macpherson suspension design to improve the vehicle anti-lift phenomena. Nagarajan et al. [10] also proposed a double pig tail double conical spring to encounter the transverse load transmitted from the road. Kim et al. [11] modelled a coil spring using helix beam model to capture the spring lateral deformation. Traditional studies of coil spring focused on numerical modeling just for vertical experiment [11]. However, the existence of side forces causes failure in vehicle components, such as ball joint [12]. Therefore, it is crucial to analyze the side force during the suspension design stage to prevent components failure.

In this study, the Macpherson strut type suspension system was analyzed through MBD of half front half vehicle model to extract the side forces under extreme load cases for suspension components failure prevention. These forces were expressed in *X*, *Y* and *Z* directions which are similar to the subjected side loads. Owing to the side loads, the top mount could be damaged and cause failure in damper strut. Moreover, bending moment generated can increase the friction of piston and rod, thus decreasing the ride comfort of the vehicle as well as causing critical failure. However, experimental approach to determine the side loads is too complex and costly. Hence, MBD method has been proposed to reduce the sophisticated experimental method.

2.0 METHODOLOGY

The overall process flow of the methodology is depicted in Figure 1. First step of the analysis was to determine the hard points and component mass input to the MBD vehicle model. The vehicle suspension data such as hard points, components mass and centre of gravity (CG) were the primary input for this half vehicle model. The model was built and numerically converted into differential algebraic equation (DAE). The DAE was then solved using quasi static FIM algorithm. Vertical forces from the model were extracted from the MBD model as force output request in the simulation.

The vehicle model built comprised of front gross vehicle mass (GVM) of 850 kg, two front Macpherson strut suspension, a rack and pinion steering system, and tires. In this analysis, the left and right of the suspension modules were identical. Vehicle motions were described in fixed global frame system of *X*, *Y*, *Z* as shown in Figure 2. Local frame markers of X_i, Y_i, Z_i were attached to moving bodies *i*. Quasi static analysis was performed to find the equilibrium for models with one or more degree of freedom and to determine the force at output located points. Subsequently, the extreme load cases as shown in Table 1 were applied to the MBD model to simulate the output forces under those driving events.

The derivation of the load cases were based on the vehicle dynamics. Braking was similar to acceleration but in opposite direction. During braking, the load was transferred between rear and front. By considering the longitudinal forces and moments, the normal loads on the front axle, W_f and rear axles W_r, could be expressed as follows [13]:

$$W_f = FAW + \frac{W \times a_x \times h}{g \times l} \tag{1}$$

$$W_r = RAW - \frac{W \times a_x \times h}{g \times l} \tag{2}$$

where *FAW* was the front axle weight, *RAW* was the rear axle weight, *W* was the vehicle weight, a_x was the longitudinal acceleration, *h* was the height from the wheel to vehicle gravity centre and *l* was the vehicle wheel base length, *g* was the gravitation constant. The longitudinal load for front, F_{xf} and rear, F_{xr} were derived from following equations [13]:

$$F_{xf} = F_{BR} \times \frac{W}{g} \times a_x \tag{3}$$

$$F_{xr} = (1 - F_{BR}) \times \frac{W}{g} \times a_x \tag{4}$$

where F_{BR} was the front brake ratio. Hence, when the vehicle was braking at 0.4 g longitudinal acceleration, 1.77 g longitudinal load and 0.78 g normal load were obtained.

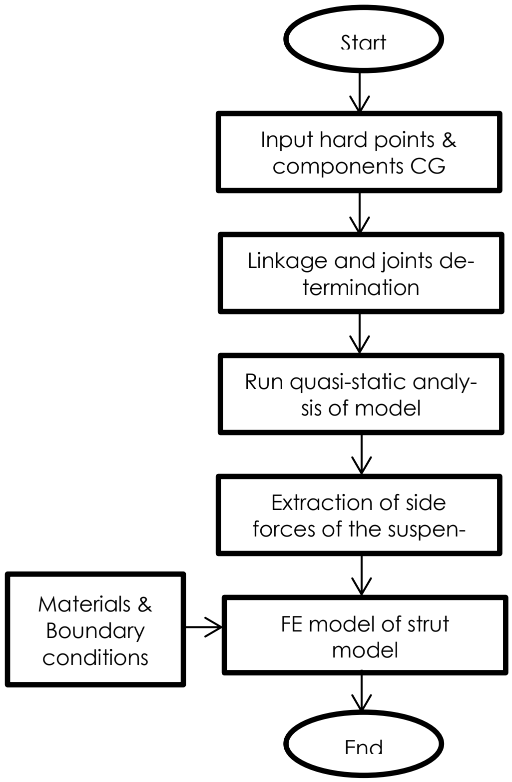


Figure 1 Methodology flow of multibody dynamics model simulation

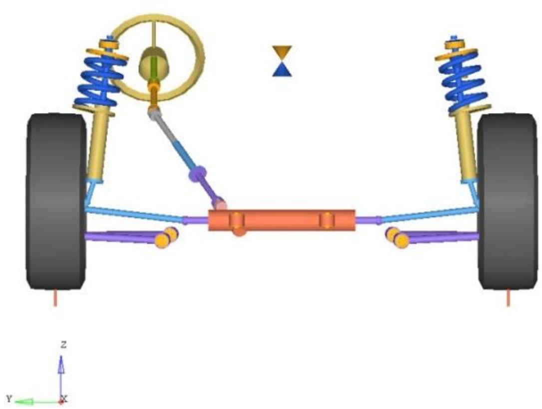


Figure 2 MBD model of suspension strut

In the cornering load case, the summation of moments about the inner and outer tire which indicates the left and right tire was assumed to be in equilibrium. The equilibrium equation was derived as follows:

$$W_o T - \frac{WT}{2} - Ma_y h = 0 \tag{5}$$

where a_y is the lateral acceleration. Hence, the normal load of the outer, W_o and inner, W_i wheel were derived as follows:

$$W_o = \frac{W}{2}\left(1 + \frac{2 \times a_y \times h}{g \times T}\right) \tag{6}$$

$$W_i = \frac{W}{2}\left(1 - \frac{2 \times a_y \times h}{g \times T}\right) \tag{7}$$

The lateral forces during cornering were taken in considerations through equations as follows:

$$F_{yo} = \frac{W}{2} \times \left(1 + \frac{a_y \times h}{g \times T}\right) \times \frac{a_y}{g} \tag{8}$$

$$F_{yi} = \frac{W}{2} \times \left(1 - \frac{a_y \times h}{g \times T}\right) \times \frac{a_y}{g} \tag{9}$$

The standard load cases were derived based on the generic load cases [4, 13, 14] where the load amplitudes were suitable for passenger car analysis.

For validation of the Macpherson strut, the coil spring and damper strut were modelled and meshed with finite element as shown in Figure 3. A total of 83901 nodes and 82025 elements were obtained from the model. The coil spring was meshed with three dimensional (3D) hexa element while two dimensional (2D) shell elements were added to obtain the surface stresses. The solid hexa element is a type of 8-noded solid element with co-rotational and one Gauss point for physical stability. The 2D shell elements were formulated with hourglass physical stability. Elastic material properties were assigned to the model with Young's modulus of 210 GPa, Poisson ratio 0.3 and density of 7900 kg/m^3. The bottom of the damper strut was fixed with rigid body and no movement was allowed while the load was applied from the top mount of the strut. An imposed displacement of 180 mm (3446 N load) was applied in vertical Z-direction.

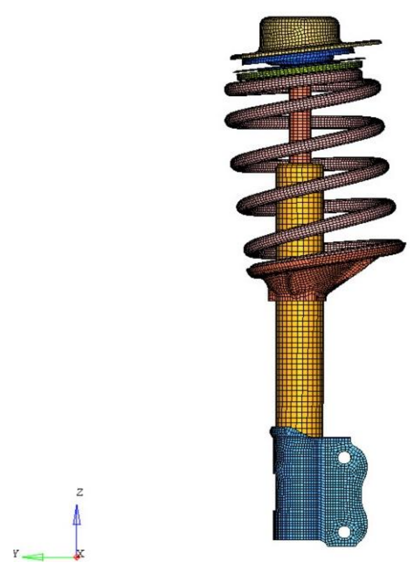


Figure 3 Finite element model of Macpherson strut

3.0 RESULT AND DISCUSSIONS

Based on current coil spring design, the material used is medium carbon steel SAE 5160H. For this carbon steel, the material ultimate tensile strength is 1450 MPa and material yield strength is 1280 MPa [15]. The design of coil is always according to the material yield strength to ensure the spring lie within the elastic limit. Therefore, the stress level of the spring after full compression shall be below 1280 MPa. Coil-damper strut was simulated with maximum compression force and the results are illustrated in Figure 4. As shown in Figure 4, the coil spring has been compressed until 180 mm. The maximum von Mises stress under fully compressed coil spring was 1250 MPa which was slightly below the yield strength of the material. Hence, the coil spring was in acceptable range because the stress levels fall within the elastic limit.

The forces induced in the spring damper strut were extracted from quasi static simulation of the front suspension system and depicted in Figure 5. This analysis serves to identify the side loads exerted on the spring damper model when the vehicle experiences severe driving events listed in Table 1. Each load cases were simulated in 2 s time interval, and therefore consist of 22 s for 11 load cases. As observed from Figure 5, the most critical vertical load came from vertical bump 3 g load. The highest force in longitudinal direction (*X*-axis) was during the pothole strike 1 which was 1900 N. Meanwhile, the highest force in lateral direction (*Y*-axis) was 3294 N, which also was during pothole strike 1. The vertical forces (*Z*-axis) are highest during bump 3 g and left hand bump 3 g load conditions which was 11,052 N.

Under these extreme load events, the force magnitudes obtained were considerably high when compared to vertical force simulation. The vertical force until the bump stopper activated was 3446 N where the 3 g bump strike can achieve 11052 N forces. The high magnitude forces under various directions will increase the inner friction between damper parts and result wearing in damper rod side. Meanwhile, when vehicle with Macpherson suspension is travelling on a smooth road, vertical vibration may be transferred to the vehicle body directly. The small road excitation could not be filtered by the damper due to inner friction between piston [3, 6]. Therefore, it is important to reduce the side forces. Based on the obtained force magnitude, the coil spring was estimated to survive for very short life. At the same time, the lateral forces exerted on the damper will damage the damper. These side forces can be useful guideline for design of Macpherson suspension strut to prevent failure. On the other hand, these forces can also be used to optimize the damper strut design, for example, through replacement with special side force designed coil springs [16].

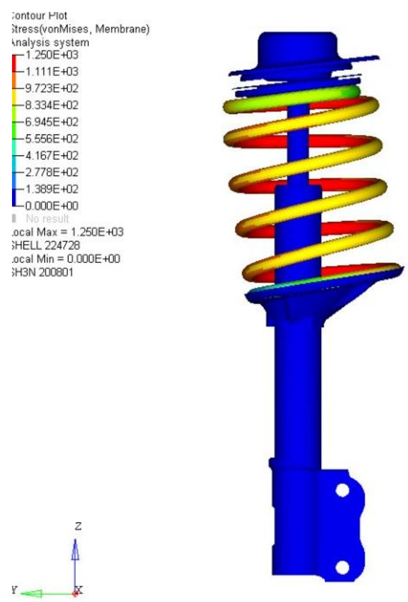
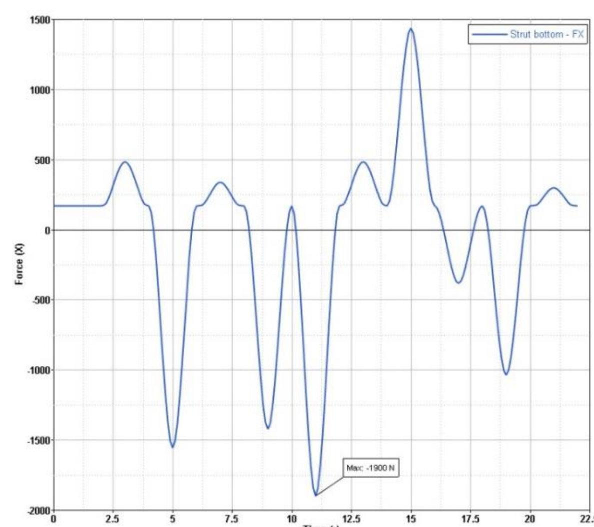
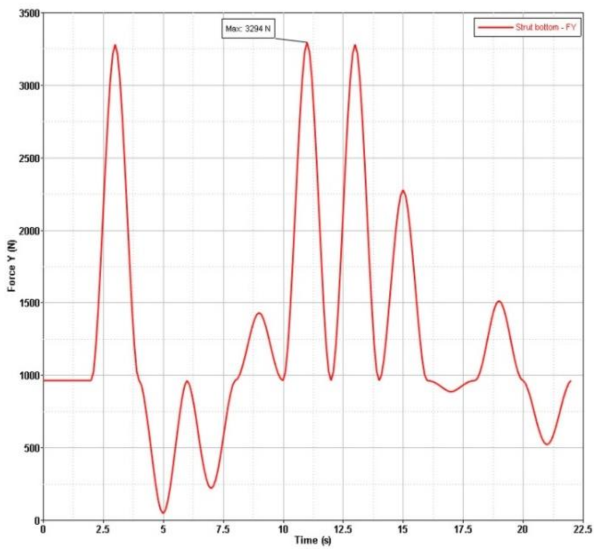


Figure 4 Stress contour of suspension strut



(a)



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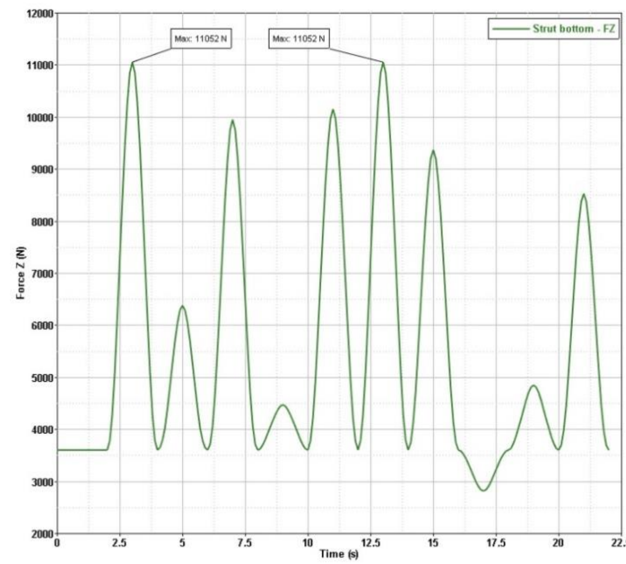

(c)

Figure 5 Side forces extracted from quasi static simulation in (a) *X*-axis (b) *Y*-axis (c) *Z*-axis

Table 1 Applied extreme load cases

	Load location											
Load cases	**LWC**			**RWC**			**LTP**			**RTP**		
	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z
Vertical load	-	-	4169	-	-	4169	-	-	-	-	-	-
Bump 3g	-	-	12507	-	-	12507	-	-	-	-	-	-
Curb strike 1	-	-	6254	-	-	4169	6254	-	-	6254	-	-
Curb Strike 2	-	-	8338	-	-	4169	-	8338	-	-	-	-
Pothole 1	-	-	5878	-	-	4169	-	-	5878	-	-	-
Pothole 2	-	-	12507	-	-	4169	8338	-	-	-	-	-
Left hand bump 3g	-	-	12507	-	-	4169	-	-	-	-	-	-
Left hand bump 3g 30D	6254	-	10839	-	-	4169	-	-	-	-	-	-
Acceleration	-3168	-	3168	-3168	-	3168	-	-	-	-	-	-
Braking	-	-	6170	-	-	6170	4503	-	-	4503	-	-
Cornering	-	-	7671	-	-	667	-	6128	-	-	542	-

4.0 CONCLUSION

This paper has discussed a detailed multi-body vehicle model for side forces extraction of the Macpherson suspension strut. The side forces extraction requires high accuracy suspension MBD model where the suspension strut model was built with sets data of hard points, linkage and components CG. According to the MBD results, the damper strut will experience most severe side forces during pothole striking event which is around 3294 N. Compared to the simulated vertical load, FE strut model stress level indicates that the force around 3294 N is extremely high and very damaging. Meanwhile, for most of the load cases, the coil spring has been compressed until the maximum displacement and the additional side forces will be transmitted to the damper strut and damage the damper. In conclusion, high magnitudes of side force which could severely damage the components have been observed in the simulation. Hence, the side forces extracted in this study could be used as a guideline for suspension strut design and failure prevention especially under extreme load cases.

This study has limitation on linkage and bushing details of the vehicle MBD model where the actual rubber components consist of stiffness and damping properties. In future, the bushing and linkage components could be studied and used as input to the MBD model to enhance accuracy of simulations.

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