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Development of analytical process to reduce side load in strut-type suspension †

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Abstract

Methods have been developed to reduce the side load and friction force acting on the shock absorber inherent in the MacPherson strut system, popularly used in vehicle suspension systems. Reducing this friction force is one of the most important issues in improving the ride comfort of a car. The side load of the shock absorber can be reduced by controlling the force line of the coil spring. To reduce the side load, we designed an S-shaped coil spring. For the design of the side load spring, we also developed an analytical process, which utilizes finite element analysis and mechanical system analysis. All analysis results for the stiffness, stress, fatigue life, and spring force line were validated through experiments.

Keywords: MacPherson strut suspension; Ride comfort; Optimal design; Side load; Spring force line

1. Introduction

Since the introduction of the MacPherson strut suspension into automobiles in the late 1940s, it has become one of the most widely used automotive suspension systems in modern automotive design. In general, MacPherson strut suspensions include a spring, which has an inclination or offset against the shock absorber axis, in order to minimize the suspension friction, which significantly influences ride comfort. The amount of offset or degree of inclination is determined based on the assumption that the reaction force of the spring acts along the geometric spring axis. However, the helical coil springs usually used in MacPherson strut suspensions have a reaction force eccentricity inherent to the basic design [1-2].

Furthermore, recent trends in compact packaging require springs to be aligned along the proper force action line, without an additional spring inclination or offset. Several types of springs with special design have been proposed by other researchers as solutions for these design constraints [3-5].

A coil spring to reduce the side load and friction applied between a rod and tube of a shock absorber was designed in this study by using the following steps.

We first constructed a customized front suspension kinematic model, which was used to calculate an ideal spring force line to minimize the inherent side load. Based on this line, we subsequently optimized the design. The spring force lines of the spring variants for the optimal design were calculated through finite element analysis (FEA) using the strut assembly model, and the reduction of the side load for an optimized spring model was evaluated by FEA using the suspension system model. Finally, we validated the analytical results for the spring force line, stiffness, stress, and fatigue life through corresponding experiments.

2. Analysis and experiment for spring force line

In a strut-type suspension system, the center axis of the coil spring has been initially coincident with the axis of the shock

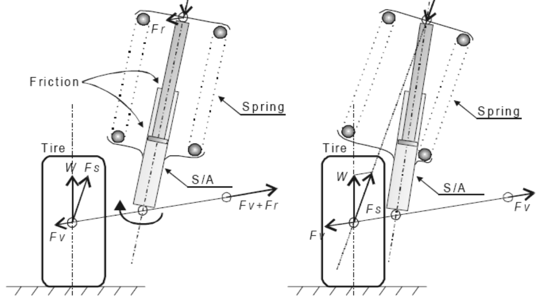

Fig. 1. Schematic diagram for loads acting on the MacPherson strut suspension.

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absorber, as shown on the left side of Fig. 1. The load W acting from the ground causes a bending moment in the shock absorber. During the piston movement of the shock absorber, the friction is increased due to this bending moment, which results in increased riding discomfort. One common approach to reducing this friction is by using an offset strut, shown on the right side of Fig. 1. With this type of strut, the friction and side load can theoretically be eliminated; however, the actual side load, in practice, does not become zero because (1) an eccentric load may be created by the deviation between the load axis and geometric center axis of the coil spring, (2) it is difficult to keep the required amount of offset because of a lack of space around the suspension, and (3) the load axis position is greatly influenced by the boundary condition of the coil spring

2.1 Calculation of the spring force line

To calculate the spring force line from FE analyses or experiments, the reaction forces (F) and moments (M) have to be measured at the location (point A) where the spring lower seat is fixed on the strut tube. This location can be the same with the intersecting point (point B) of the spring lower seat plane and the strut axis. The spring force line means the line connecting two points (point L and point U) on the spring upper and lower plane at which no reaction moments are exerted. The two points can be calculated as follows: (Fig. 2)

Measured forces: $F = [f_x, f_y, f_z]$

Measured moments: $M = [m_x, m_y, m_z]$

Point O where the moments are zero:

$$[a,b,c] = \left[-\frac{m_y}{f_z}, \frac{m_x}{f_z}, z_o \right]$$

The line equation passing point O:

$$\frac{x-a}{f_x} = \frac{y-b}{f_y} = \frac{z-c}{f_z}$$

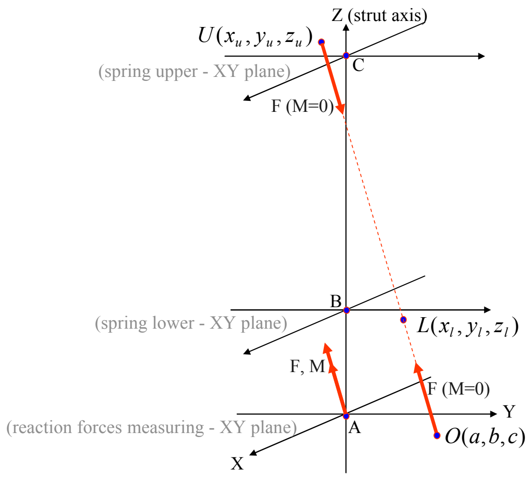


Fig. 2. Schematic diagram to calculate the spring force line.

Point L and point U:

$$[x_l, y_l, z_l] = \left[\frac{f_x(z_l - c)}{f_z} + a, \frac{f_y(z_l - c)}{f_z} + b, z_l \right]$$

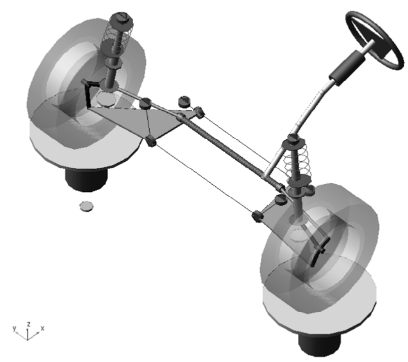
$$[x_u, y_u, z_u] = \left[\frac{f_x(z_u - c)}{f_z} + a, \frac{f_y(z_u - c)}{f_z} + b, z_u \right]$$

2.2 Determination of the ideal spring force line

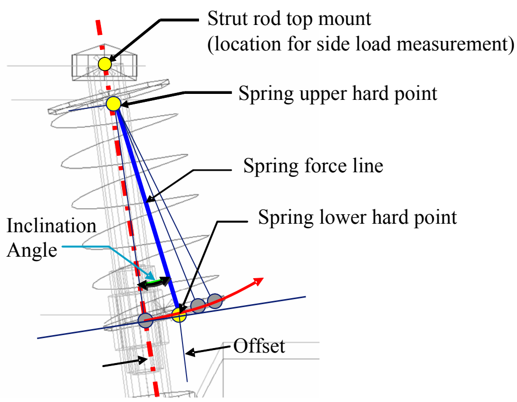
We used the MSC.ADAMS software, which can solve multi-body dynamics problems, to calculate the side load applied to a strut-type suspension and find the spring force line that would minimize the side load.

Fig. 3 shows the front suspension kinematic model used to find the ideal spring force line for the target vehicle, and describes the terms used to define a spring force line. The spring force line is defined as the line that connects the spring upper hard point to the lower hard point. The ideal spring force line can be found by moving the spring lower hard point to the location which minimizes the side load generated at the strut rod top mount during vertical movement of the wheel.

Fig. 4 shows the change in side load at the strut rod top mount for various inclination angles. A vertical wheel travel, in the general driving conditions, between -20 and 20 mm is assumed. The side load can be minimized in this range most



(a) Front suspension kinematic model



(b) Spring force line

Fig. 3. MacPherson strut suspension kinematic model to find the ideal spring force line.

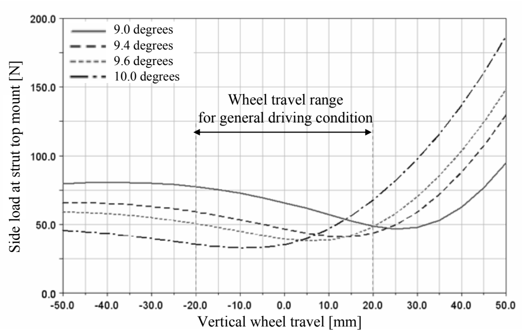


Fig. 4. Side load at strut rod top mount.

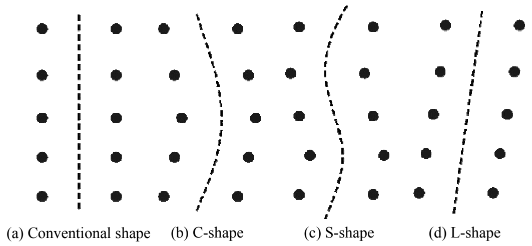


Fig. 5. Shapes of coil spring.

significantly when the inclination angle is 9.6 degrees, which means that the spring lower hard point offset must be 37.54 mm as the target spring force line.

2.3 Optimal design

There are many design restrictions, such as material properties, packaging, stiffness, strength, loading, force line, and fatigue life. Coil diameter, wire diameter, total number of coil turns, and maximum design stress are calculated according to these constraints. In this study, the basic design requirements were selected as follows:

(1) Stiffness ≈ 23.0 N/mm
(2) Maximum Shear Stress < 1100 MPa
(3) Fatigue Life > 300000
(4) Spring Lower Hard Point Offset ≈ 37.54 mm

The coil springs, for reducing the side load, are classified into C-, L-, and S-shaped springs according to the shape of the spring center lines, as shown in Fig. 5.

In this study, we selected the S-shaped spring, which has advantage of spring force line adjustment, and performed the optimal design.

2.4 Analytical evaluation at the strut assembly level

The force lines of spring variants for the optimal design were evaluated through FEA, using strut models composed of a shock absorber and spring. We used a nonlinear finite element solver (ABAQUS) for the FEA, which can solve geometric nonlinear problems, such as contact conditions between spring and spring seats, as well as evaluate any non-linear geometric behavior.

Fig. 6 and Table 1 show the analysis results for the spring force line in the design weight condition of a vehicle. The spring force line of the optimal spring design closely tracks the target line, while the line of the conventional shape spring shows a difference from the minimum side load target line.

The shear stress contours at each loading condition are shown in Fig. 7. The maximum shear stress is 1051 MPa, which is lower than the allowable stress of 1100 MPa. Thus, the stress level is satisfied with stresses lower than the design limit.

Table 1. Force line position on upper and lower seat.

	Upper seat		Lower seat	
	x [mm]	y [mm]	x [mm]	y [mm]
Target	0.00	0.00	0.00	37.54
Conventional	-2.95	7.51	-1.19	18.27
Optimum design	1.04	0.59	-1.68	37.91

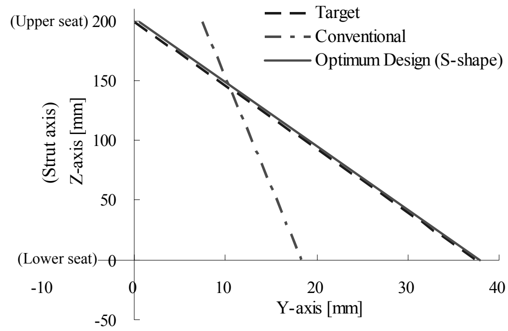


Fig. 6. Comparison of spring force lines in the design weight condition.

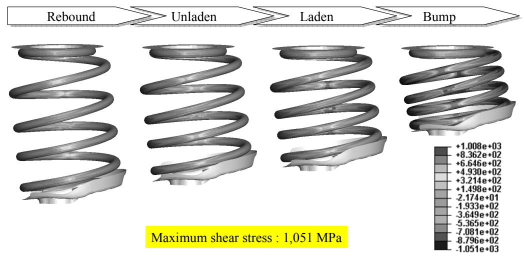


Fig. 7. Stress analysis results.

2.5 Analytical evaluation at the suspension system level

The reduction of the side load for the optimized spring model was evaluated through FEA, using a suspension system model which consists of strut assemblies, lower control arms, knuckles, and tierods.

Fig. 8 shows the finite element model for the front suspension system. Except for springs, all suspension components were modeled as rigid bodies connected by joint elements, with degrees of freedom representative of the actual connections in the vehicle.

Fig. 9 shows the changes of side loads at the strut top mounts for the conventional and optimized S-shaped springs during vertical wheel travel from design weight (or unladen) to gross vehicle weight (GVW or laden) conditions. At the same loading conditions, the side load generated by the con-

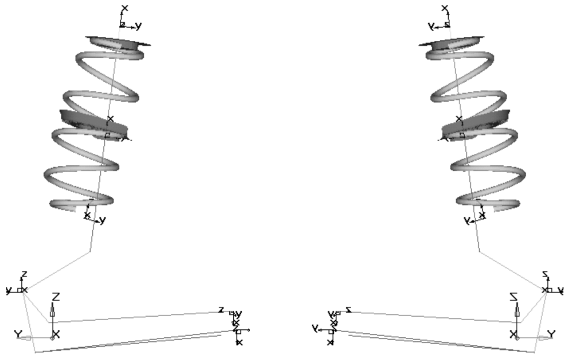


Fig. 8. Finite element model for front suspension system.

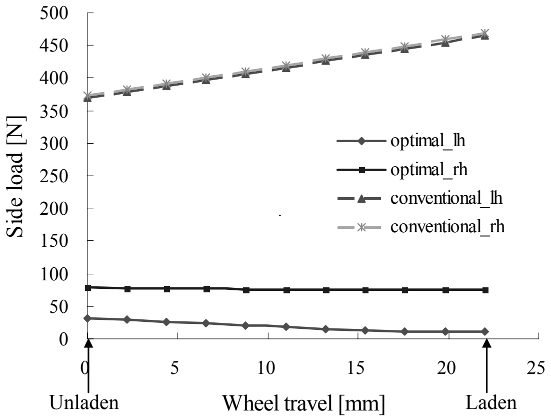


Fig. 9. Comparison of side loads at strut rod top mount.

ventional spring ranged from 375 N to 460 N, while the optimized spring generated maximum side loads of 80 N. The results show that side load was reduced by 80% through design optimization.

2.6 Experimental validation for spring force line and stiffness

To validate the design and analysis process of the spring for reducing the side load, we manufactured the optimized spring and performed an experiment to validate the calculated spring force line. We used a load cell, which can measure 3-axes forces and moments, for the experiment. Fig. 10 shows the experimental setup.

As shown in Fig. 11, the spring force lines, calculated using the forces and moments extracted from the experiment and FEA, respectively, are very close to the theoretical target spring force line.

The spring stiffness can also be calculated using the relation between force and displacement, obtained from the experiment and analysis for spring force line. Fig. 12 and Table 2 represent the comparison results for stiffness. In Fig. 12, the strut displacement means the compressed length of a spring. Table 2 shows that the stiffness values are very similar to the design requirement of 23 N/mm, and the difference between the experiment and FEA is 2.1%.

Table 2. Comparison of stiffness between experiment and FEA.

Calculation method	Stiffness [N/mm]		Error [%]
	Experiment	FEA	
Linear regression	22.86	22.78	0.3
2 Point at both ends	23.39	22.91	2.1

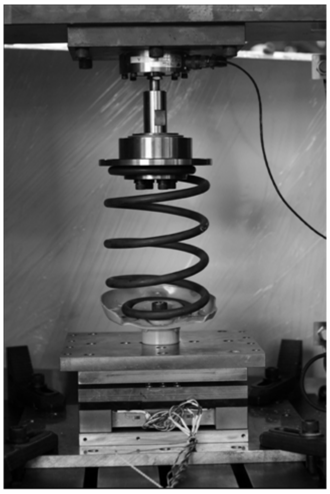


Fig. 10. Experimental setup for measuring spring force line.

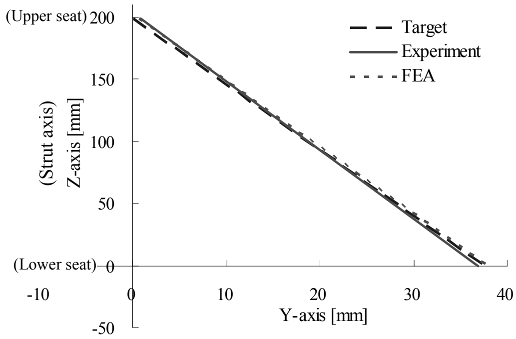


Fig. 11. Comparison of spring force line between experiment and FEA in the design weight condition.

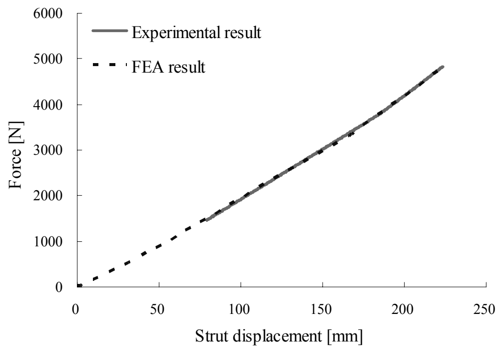


Fig. 12. Stiffness curves for experiment and FEA.

3. Analysis and experiment for stress and fatigue life

The spring force line and stiffness were validated experimentally in the previous section. In this section, we validate the reliability of an analytical approach to the stress and fatigue life through a series of experiments.

3.1 Finite element model and constraints

Fig. 13 shows the finite element model and its constraints for a coil spring. The upper seat can only rotate and the lower seat can only translate along the strut axis. The degrees of freedom of the other directions are fixed. Therefore, the spring is deformed only in the direction of the strut axis when a load is applied at the lower seat. The upper and lower seats are modeled as rigid bodies, with the contact conditions between the spring and seats. Fig. 14 shows the input load time history applied at the lower seat.

3.2 Strain gage instrumentation and experiment

We measured the strain on the spring surface for a comparison of stress between FEA and experiment. The strain measurement locations were selected based on FEA results, where in-plane maximum principal stress occurs in the GVW condition. The rectangular rosette gage was attached at this corresponding location, as indicated by the FE model (Fig. 15). The strain signals acquired from the experiment were used to calculate the actual stress (later compared to the predicted stress).

Fig. 16 shows the experimental setup. We fixed the strut tube and applied loads in the same condition as Fig. 14 at the strut top mount, and acquired the displacement signal from an actuator and strain signals from the rosette gage.

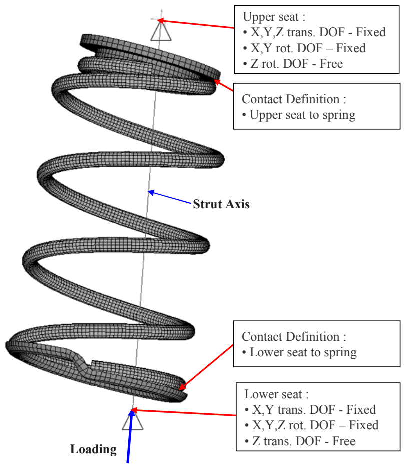


Fig. 13. Finite element model and constraints.

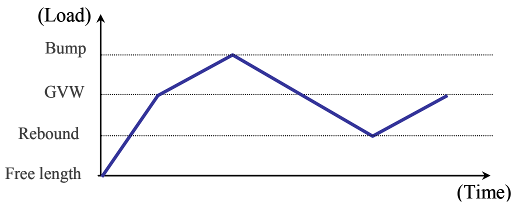


Fig. 14. Input load time history at spring lower seat.

Table 3. Strain and stress results for test and analysis.

	Experiment	FEA	Error [%]
Max. Principal Strain [/]	4.76E-03	5.12E-03	7.0
Max. Shear Strain [/]	8.42E-03	9.15E-03	8.0
Max. Principal Stress [MPa]	809	864	6.3
Max. Shear Stress [MPa]	653	709	8.0

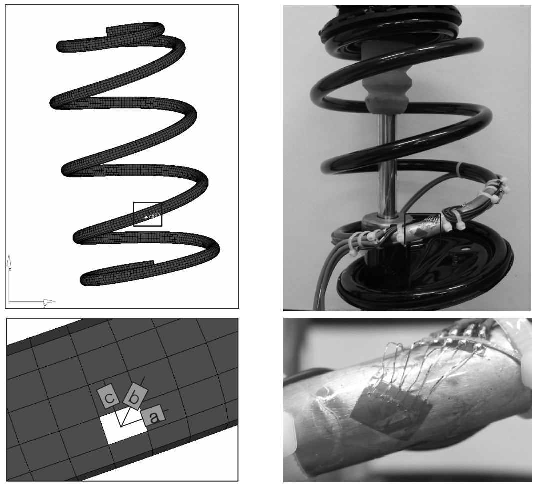
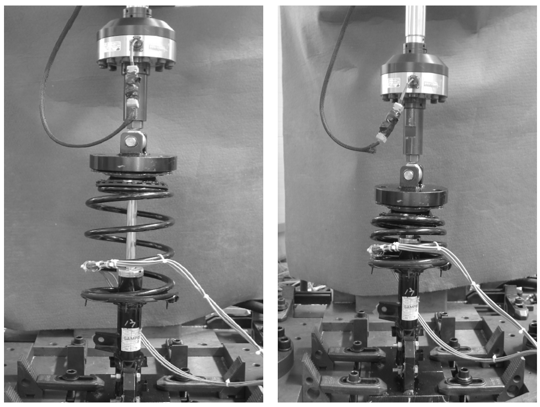


Fig. 15. Strain gage attached at max. principal stress location.



(a) Rebound condition (b) Bump condition

Fig. 16. Experimental setup for measuring strain.

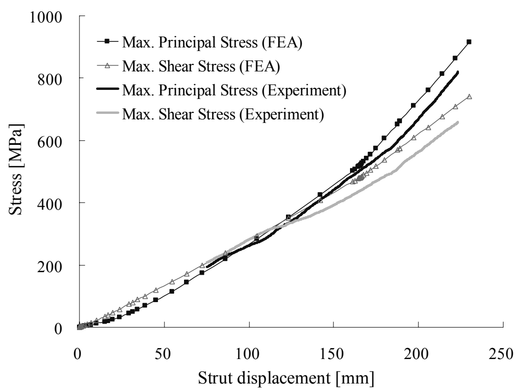
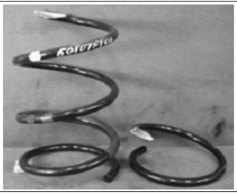


Fig. 17. Comparison of stress at maximum principal stress location.

Table 4. Comparison of fatigue life for test and analysis.

	Analysis Result	Experimental Result
Fatigue Life		
	459,968 cycles	479,875 cycles

3.3 Evaluation of stress

Fig. 17 shows that stress changes as a function of strut displacement at the maximum principal stress location. The stress values from the FEA are higher than the experimental values. The strain and stress values in the bump load condition are summarized in Table 3, which shows a maximum difference of 8.0 % between the FEA and experiment. It is presumed that the difference of stress occurred because the three strain gages, which make up the rosette gage, are separated by angle of 45 degrees, and thus there is a difference between where the data was gathered and the theoretical FE predicted stress peak.

3.4 Evaluation of fatigue life

The following shows the setup for the fatigue analysis of a coil spring

(1) Analysis Method: Strain-Life approach
(2) Mean Stress Correction: Smith-Watson-Topper
(3) Surface Treatment: Shot Peening
(4) Load Condition: GVW→Bump→Rebound→GVW

As shown in Table 4, the difference in the fatigue life between the analysis and experimental result is 4.1 %, which is a reliable analysis result with regard to the deviation of product quality.

4. Conclusion

We developed a series of analytical processes for the design of a coil spring to reduce the side load, and performed the following.

(1) We constructed a customized front suspension kinematic model, which can calculate an ideal spring force line to minimize the side load.
(2) The spring force line design was optimized.
(3) Experiments were carried out to validate the analysis results for spring force line, stiffness, stress, and fatigue life.

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