Overall Strength Analysis of Trailing Arm Torsion Beam Suspension System

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Abstract. Previous studies focusing on analyzing components of automotive suspension system separately usually ignore the fact that each single component makes movement during operation resulting in simulation bias. This paper carries out a thorough analysis on the overall strength of a trailing arm torsion beam suspension system and compares the results with previous ones from analyzing single pieces. Consequently, significant improvements will be made in the design of suspension system.

1. Introduction
Trailing arm torsion beam suspension is a relatively new rear suspension system, to which two longitudinal arms with each attached to a separate wheel is welded. It is a commonly-seen rear suspension system in automobiles. It functions to pass through various forces and movements between the body and wheels, alleviate impact load passed by uneven roads, and control the movement of wheels as to ensure the ride comfort and handling stability. Therefore, the strength of the suspension system has a direct impact on the reliability and life span of vehicles [1].

The main components of the railing arm torsion beam suspension system include coil spring, shock absorber, trailing arms, shock absorber bracket, hub carrier bracket, trailing arm reinforcing plate and rear reinforcing plate. Previous studies on the strength of the suspension system only concern the strength check of components neglecting the fact that components make movements during operation, which leads to the result of miscalculation and inaccuracy [2-4].

This paper proposes to analyze the overall strength of the suspension system by use of the trailing arm torsion beam rear suspension system of a real car as a sample, aiming to establish constraint relationships between various components and model elastic units, such as coil spring, shock absorber, and bushing. Subsequent results will be compared with previous results on single components, based on which reasons explaining the differences will be concluded [5].
2. Finite Element Modeling and Load Cases Determined

(1) Finite Element Modeling. The trailing arm torsion beam rear suspension system of the sample mainly consists of coil spring, shock absorbers, rear beam, trailing arm, shock absorber bracket, hub carrier bracket, trailing arms reinforcing plate, and rear spar reinforcing plate. A geometric model is established in 3D software CATIA based on the design drawings, as shown in Figure 1. Then, analyzing material properties, defining connection properties between components and boundary conditions are completed in HYPERMESH software to generate finite element model, followed by calculation in ABAQUS software. Real beam, trailing arm, shock absorber bracket, trailing arm reinforcing plate, rear reinforcing plate are divided by SHELL elements; welding spots are simulated by RIGID element; welding line is simulated by SHELL element with the same thickness; hub seat holder is divided by the second-order tetrahedral elements; spring members including spring and shock absorber are simulated by bush unit, while bush connection parts are given corresponding nonlinear elastic properties. Overall, 39352 elements and 42242 nodes, form the finite element model, as shown in Figure 2.

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1. rear beam; 2. trailing arm; 3. trailing arm reinforcing plate; 4. shock absorber bracket; 5. hub carrier bracket; 6. coil spring; 7. shock absorber; 8. rear spar reinforcing plate

**Figure 1.** CAD model of a trailing arm torsion beam suspension

**Figure 2.** Finite element model of the railing arm torsion beam suspension system
(2) **Load Case.** Load born by wheels in motion is most likely random, but sometimes varies according to the load of vehicle, driver’s operation, engine output torque, road roughness, surface adhesion coefficient and speed. It is, therefore, of significant importance to determine the typical operating conditions as the strength check based on mechanical characteristics of the suspension [1].

The forces of suspension system can be summarized under four typical working conditions: (1) The maximum vertical force condition, namely full-load vehicle on uneven roads withstands impact loads; (2) The maximum braking power condition, namely full-load vehicle hits emergency braking; (3) The maximum longitudinal force condition, namely full-load vehicle applies Linear Acceleration; (4) The maximum lateral force condition, namely full-load vehicle make turns. Calculated as follows:

(1) The maximum vertical force conditions, the maximum vertical force $F_{Vm}$ of unilateral rear wheel at the contact with ground as follows:

$$F_{Vm} = k_2 F_{Vor}$$

(1)

Where: $k_2$ - dynamic load factor for the static strength, shown in Figure 3; $F_{Vor}$ - full of static load on the front wheels.

(2) The maximum braking power condition, the rear wheel brake load is:

$$F_{Br} = 0.8 F_{Vor}$$

(2)

$$F_{VBr} = F_{Vor}$$

(3)

Where: $F_{Vor}$ - full static load for rear wheel; $F_{Br}$ - one side braking force for the rear wheels; $F_{VBr}$ - vertical force of rear wheels during braking condition.

(3) The maximum longitudinal force conditions, the rear wheel vertical load increases, the rear wheel load cases as follows:

$$F_{A2r} = 1.1 F_{Vor}$$

(4)

$$F_{VA2r} = k_1 \cdot F_{Vor}$$

(5)

Where: $F_{A2r}$ - a longitudinal force on the rear wheel; $k_1$ - vertical dynamic load factor for the durability, shown in Figure 3; $F_{VA2r}$ - vertical load on the rear wheels during acceleration.

**Figure 3.** Characteristic curve of vertical dynamic load (k1 - vertical dynamic load factor for the durability, k2 - dynamic load factor for the static strength)
(4) The maximum lateral force condition, the load on the rear wheels shown in Figure 4.  
\[ F_{Zrl} = \frac{m_2 g}{2} + \Delta F_z \]  
\[ F_{Zrr} = \frac{m_2 g}{2} - \Delta F_z \]  
\[ \Delta F_z = \frac{m_z a_y h}{t} \]  

Where: \( m_z \)-the front axle load; \( a_y \)-the turning acceleration; \( h \)-centroid height; \( t \)-wheelbase;  
Therefore, lateral load \( F_{yrl} \), \( F_{yrf} \) on the left and right side wheels are:  
\[ F_{yrl} = \mu F_{Zrl} \]  
\[ F_{yrr} = \mu F_{Zrr} \]  

![Figure 4. Force analysis diagram of automobile during cornering](image)

3. Analysis of the overall strength of the Suspension System  
The basic parameters of a car: wheelbase 2530mm; full centroid height 550mm; tread 1550mm; tire radius 289mm; rear-sided full load 381kg. Torsion trailing arm suspension system for the material parameters: rear beam material QSTE420TM; vertical arm material STEEL10; hub carrier scaffold is ST52-3; the shock absorber bracket, trailing arm and rear reinforcing plate reinforcing plate materials are SPHE. Where, the ultimate strength for material of QSTE420TM, STEEL10, ST52-3 and SPHE respectively is 471MPa, 424MPa, 355MPa and 300MPa.  
Force exerted on wheels is rather complicated to analyze. In addition, material nonlinearity, geometric nonlinearity and contact nonlinear, together accounting for vast resources to be calculated, are all involved in finite element analysis, which can hardly be completed in a short term. Consequently, the role of the tire is not reckoned in this model. Instead, equivalent force taking place on a connection point between the wheel hub and the wheel is replaced. Constraints are: six degrees of freedom between vertical arm and body connection point are constrained; X, Y and Z directions of translational degrees of freedom between damper and body fixed point are constrained. Point load is applied in the
hub of the wheel assembly point. According to the conditions considered the hub assembly center in the equivalent load conditions, as shown in Table 1:

Table1. Load at the center of the wheel rim assembly in typical operating conditions

| Load case | F_x (N) | F_y (N) | F_z (N) | M_x(N·mm) | M_y(N·mm) | M_z(N·mm) |
|-----------|---------|---------|---------|-----------|-----------|-----------|
| Load case 1 | 0       | 0       | 9335    | 0         | 0         | 0         |
| Load case 2 | 2011    | 0       | 2011    | 0         | -581179   | 0         |
| Load case 3 | 3733    | 0       | 3733    | 0         | -1078837  | 0         |
| Load case 4 | 0       | 6622    | 6622    | 1913758   | 0         | 0         |

Under conditions (1), the impact load on the wheel shown in Figure 3 of vertical dynamic load curve, k_3 take 2.5. Vertical load on the rear wheels by the formula (1) be regarded as the 9335N, as shown in Table 1. Overall stress contours of rear beam shown in Figure 5, the maximum stress of 213MPa, appears in welding position between rear beam and vertical arm; Longitudinal arm stress contours, as shown in Figure 6, the maximum stress of 385MPa, appears in the bent cross section near the rear beam and trailing arm. Stress contours of absorber bracket as shown in Figure 7, the maximum stress of 274MPa, occur at the transition arc; Hub carrier bracket stress contours shown in Figure 8, the maximum stress of 227MPa, occur in small sectional place; Vertical arm reinforcing plate stress contours shown in Figure 9, the maximum stress 277MPa, appears in the longitudinal arm weld region; Rear beam reinforcing plate stress contours shown in Figure 10, the maximum stress of 225MPa, appears in the curvature region. In this condition, the maximum stresses of the components do not exceed the yield strength of the materials to meet the static strength requirements.

Likewise, the maximum stress value and its distribution position of the components of suspension system in other three typical working conditions are acquired, using the same calculation method applied in condition 1.

By calculating under the four typical operating conditions, the conclusion is: in the maximum lateral force conditions, the maximum stress of the side connecting hub carriers and reinforcing plate of vertical arm is 409MPa, exceeding the yield limits. Under other conditions, the components of suspension systems meet the strength requirements.

4. Comparative Analysis

Results from the overall strength of the suspension system in finite analysis are compared and contrasted with previous records on single component. Figure 11 shows rear beam stress contours in a single piece of analysis. Table 2 shows the maximum stress of the two methods compared.
Figure 5. Stress contours of rear beam

Figure 6. Stress contours of vertical arm
Figure 7. Stress contours of shock absorber bracket

Figure 8. Stress contours of hub carrier bracket
Figure 9. Stress contours of vertical arm reinforcing plate

Figure 10. Stress contours of rear spar reinforcing plate
It is concluded from the charts that stress value on one-piece analysis is generally higher than the overall. The reasons are summarized below: (1) Connections between components are not considered in one-piece analysis, so differences are shown between simulation results and actual load. (2) Fixed constraint and concentrated load are applied in single-piece analysis, resulting in higher stress level. (3) The flexible rigidity of the sleeve connecting is not taken into account in the strength analysis of the suspension system, which has a certain degree of impact on the results.

![Figure 11. Rear beam stress contours in a single piece of analysis](image)

Table 2. The maximum stress of the two methods compared

| Type                  | Single piece | Overall | Maximum stress results (MPa) |
|-----------------------|--------------|---------|-----------------------------|
|                       |              |         | 1  | 2  | 3  | 4  |
| Rear beam             | 265          | 77      | 141 | 386 |
|                       | 213          | 63      | 115 | 365 |
| Vertical arm          | 420          | 105     | 189 | 412 |
|                       | 385          | 86      | 161 | 375 |
| Shock absorber bracket| 331          | 107     | 195 | 237 |
|                       | 274          | 98      | 169 | 218 |
| Hub carrier bracket   | 254          | 103     | 195 | 451 |
|                       | 227          | 90      | 166 | 409 |
| Vertical arm reinforcing plate | 308  | 186    | 293 | 304 |
|                       | 277          | 161     | 279 | 273 |
| Rear spar reinforcing plate | 269 | 97      | 164 | 317 |
|                       | 225          | 73      | 132 | 278 |

5. Conclusion

Stress of connecting bracket of the trailing arm hub seat suspension system exceeds the ultimate strength of the material. It is recommended to flatten depression while increasing the thickness of the
plate. The rest of the components meet strength requirements under the four typical working conditions. It is concluded from comparative analyzing that stress values in single-piece analysis is generally higher than overall. The reasons include ignoring connections between components, the use of fixed constraint and application of concentrated load. Overall, the close-to-fact results from overall analysis, compared with rather conservative single-piece analysis, provide more accurate scientific proof for the design of suspension system.

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