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# Matching of Suspension Damping and Air Spring Based on Multi-Body Dynamic Model

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**Abstract**

By using multi-body dynamics simulation software SIMPACK, the paper completed full vehicle modeling of bus air suspension. In four typical working conditions, body vertical acceleration, suspension working space and dynamic tire load were selected as performance indexes to analyze the matching of suspension damping and air spring. On the basis of comparison of simulation data, a suitable damping was selected for a good matching with air spring in corresponding condition and the performance of the suspension system was improved.



*Keywords:* air suspension,*SIMPACK*,damping,matching

## 1. Introduction

There was a large displacement motion and existed a space nonlinear relationship of various components in air spring system. Thus, there were significant limitations to study the vehicle dynamic characteristics by using a mathematical model simplified greatly from full vehicle. Therefore, establishing a whole vehicle model close to realistic system could contribute to more accurate researches on vehicle dynamic

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characteristics[1][2]. Analysis software *SIMPACK*, based on multi-body system dynamics, could be used to obtain the whole vehicle modeling.

During vehicle's working, its operating conditions changed frequently and the matching of suspension stiffness and damping would affect the suspension performance. By selecting a bus with type YBL6891H as the research object, the paper completed multi-body dynamic modeling of full vehicle and studied the matching analysis of suspension damping and air spring under four typical operating conditions to improve the system's performance.

## 2. Multi-body dynamic modeling of full vehicle

### *2.1. Multi-body system dynamics theory*

In the research areas of multi-body dynamics system, an inertial system is often considered as a global reference system $I$ . Two objects determine the location of point $P$ . One is the original coordinate of connecting body coordinate system, and the other is connecting body coordinate system's azimuth parameter in the global coordinate system.

Coordinate transformation is represented in the form [3]

$$r^P = r + As'^P \quad . \tag{1}$$

Where, $r^P$ is the coordinate of point $P$ in the global coordinate system $oxyz$ , $r \equiv [x_0, y_0, z_0]^T$ is the origin $o'$ of the connecting body coordinate system $o'x'y'z'$ in global coordinate system $oxyz$ , also known as rigid position coordinate. $s'^P$ is the coordinate of point $P$ in the connecting body coordinate system $o'x'y'z'$ , $A$ is a direction cosine matrix of the connecting body coordinate system $o'x'y'z'$ relative to the global coordinate system $oxyz$ , also called as rotation transformation matrix.

According to Euler Theorem, from the coincident starting point of connecting body coordinate system and global coordinate system, rigid body azimuth can be determined by rotating successively connecting body coordinate axes $o'x'$ , $o'y'$ and $o'z'$ by limited angles $\alpha$ , $\beta$ and $\gamma$ , namely rotation angels of connecting body coordinate system relative to global coordinate system. $\alpha$ , $\beta$ and $\gamma$ are Euler angles, respectively called as precession angle, nutation angle and self-rotation angle, which constitute a rigid body rotation coordinate $\varphi = [\alpha, \beta, \gamma]^T$ . Hence, the direction cosine matrix $A$ becomes

$$A = \begin{bmatrix} \cos\alpha\cdot\cos\beta - \sin\alpha\cdot\cos\beta\cdot\sin\gamma & -\cos\alpha\cdot\sin\gamma - \sin\alpha\cdot\cos\beta\cdot\cos\gamma & \sin\alpha\cdot\sin\beta \\ \sin\alpha\cdot\cos\gamma + \cos\alpha\cdot\cos\beta\cdot\sin\gamma & -\sin\alpha\cdot\sin\gamma + \cos\alpha\cdot\cos\beta\cdot\cos\gamma & -\cos\alpha\cdot\sin\beta \\ \sin\beta\cdot\sin\gamma & \sin\beta\cdot\cos\gamma & \cos\beta \end{bmatrix} , \tag{2}$$

By combining the rigid position coordinate $r \equiv [x_0, y_0, z_0]^T$ with the rigid rotation coordinate $\boldsymbol{\varphi} = [\alpha, \beta, \gamma]^T$ , a generalized Cartesian coordinate can be obtained as

$$q = \begin{bmatrix} r \\ \boldsymbol{\varphi} \end{bmatrix} = \left[x_0, y_0, z_0, \alpha, \beta, \gamma\right]^T \quad . \tag{3}$$

Therefore, the generalized coordinate vector of the component $i$ can be composed by connecting body coordinate system's origin coordinate $r_i$ and Euler parameters, expressed as

$$q_i = \begin{bmatrix} r_i \\ \boldsymbol{\varphi}_i \end{bmatrix} = \left[x_{0i}, y_{0i}, z_{0i}, \alpha_i, \beta_i, \gamma_i\right]^T \quad . \tag{4}$$

The rigid position coordinate $r \equiv [x_0, y_0, z_0]^T$ and the direction cosine matrix $A_i$ can be expressed as functions of the generalized coordinate vector $q_i$

$$r_i = r_i(q_i)\ ,\ A_i = A_i(q_i)\ . \tag{5}$$

According to the reference system $I$ , the speed of the centroid $P_i$ can be calculated as

$$v_i = \frac{dr_i}{dt} = \frac{\partial r_i}{\partial q_i}\dot{q}_i \cdot \tag{6}$$

The displacement rate equation of centroid can be described by a Jacobi matrix $J_{Ci}$ , and similarly, the angular velocity can be calculated by a Jacobi rotation matrix $J_{\omega i}$ , namely

$$v_i = J_{ci}\dot{q}_i\ ,\ \omega_i = J_{\omega i}\dot{q}_i \tag{7}$$

where

$$J_{Ci} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \end{bmatrix},\quad J_{\omega i} = \begin{bmatrix} 0 & 0 & 0 & s\beta s\gamma & c\gamma & 0 \\ 0 & 0 & 0 & s\beta c\gamma & -s\gamma & 0 \\ 0 & 0 & 0 & c\beta & 0 & 1 \end{bmatrix}.$$

A motion equation of multi-body systems composed by $p$ rigid bodies and a complete ring can be expressed by Second Lagrange Equation[4]

$$\frac{d}{dt}\left(\frac{\partial T}{\partial q}\right)^T - \left(\frac{\partial T}{\partial q}\right)^T = Q\ , \tag{8}$$

where $T$ is the kinetic energy of multi-body system, $q$ is the generalized coordinate vector of system, and $Q$ is the generalized force vector, represented in the form

$$Q = \sum_{i=1}^{p}(J_{Ci}^T F_i + J_{\omega i}^T M_i)\ , \tag{9}$$

where $F_i$ is the application force of a rigid body $i$ and $M_i$ is the torque of the body.

### 2.2. Full vehicle modeling

YBL6891H type bus was chosen as the study object in this paper, using a four-link independent air suspension with stabilizer bar and the similar structure of the front and rear suspensions. The guiding mechanism was composed by two longitudinal rods and two V-shaped thrust rods. First, a three-dimensional entity assembly drawing was completed by 2D drawing application software *CATIA*[5]. In the process, the measurement of the critical hinge points' positioning coordinates and the calculation of every part's quality, centroid and rotational inertia were the keys to the foundation of an accurate multi-body dynamics model. Second, a full vehicle topology was established according to the connection of various components and motion analysis. Lastly, the multi-body dynamic vehicle model was obtained in *SIMPACK*[6], shown in Fig.1.

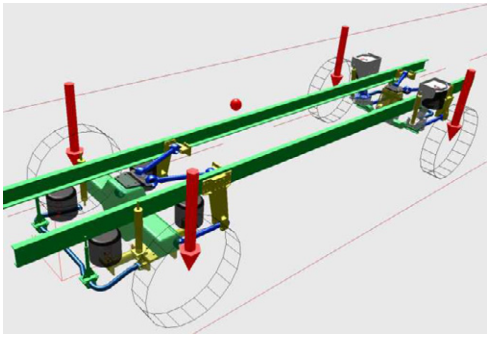


Fig.1. Full vehicle model in SIMPACK

### 3. B-class random road model

Although *SIMPACK* provides several random road models, no road model conforms to our standard B-class random road model entirely. Hence, on the basis of the data of B-class road generating from *AR* model, the paper obtained B-class random road model by means of defining a road file according to the model format in *SIMPACK*. The generated B-class road excitation is shown in Fig.2.

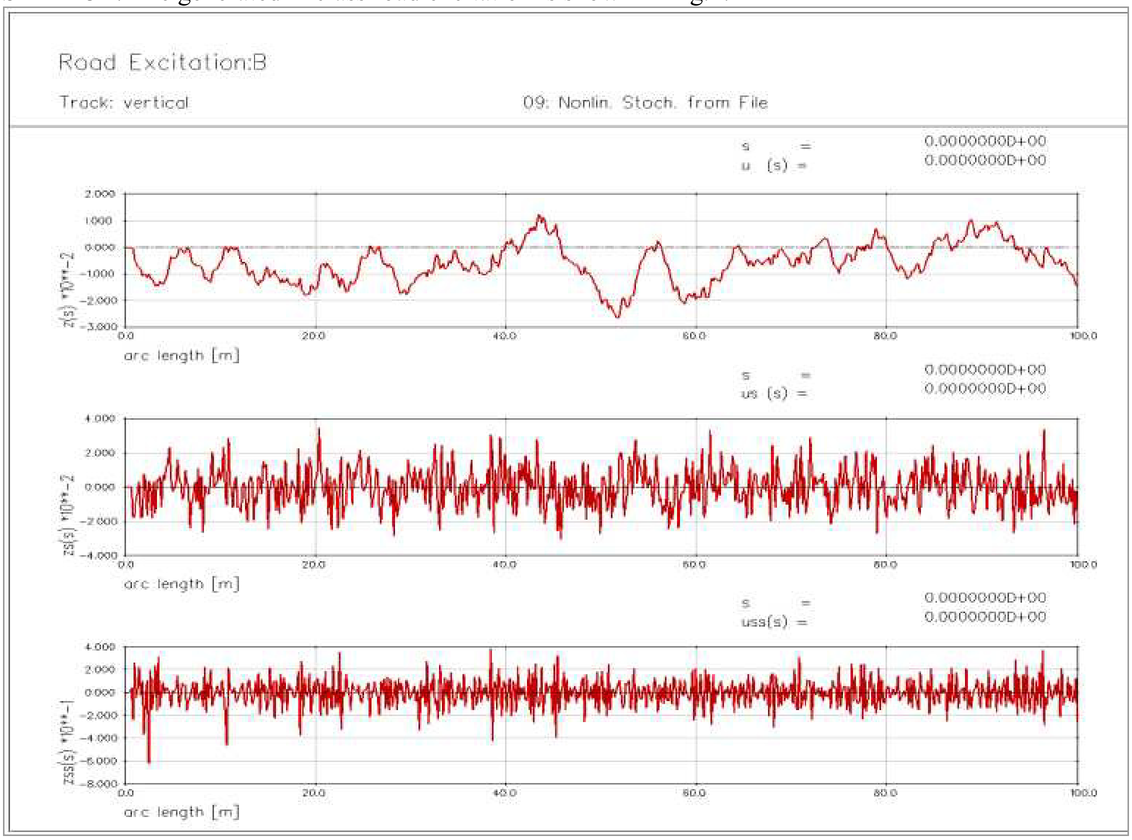



Fig. 2. B-class road surface

## 4. Matching of suspension damping and air spring in multiple loading conditions

During the operation of the vehicle under various conditions, the air spring stiffness changes as conditions change. In order to improve the performance, the suspension damping should be changed to match the changing stiffness of air spring. So it's important to realize the matching of suspension damping and air spring. According to vibration theories and engineering practices, the suspension damping performance can be measured by a relative damping ratio $\zeta$ [7]

$$\zeta = \frac{C}{2\sqrt{m_s K}} = 0.2 \sim 0.45 \cdot \tag{10}$$

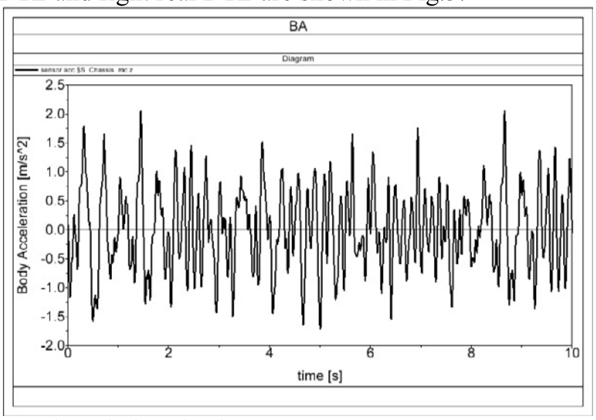
The ranges of front and rear shock absorbers' damping are determined by the full vehicle basic parameters and the formula(10). The damping range of front shock absorber is *6000-11000(N·m/s)*, and that of rear shock absorber is *8000-18000(N·m/s)*. According to the specific parameters of *YBL6891* type bus and some statistics [8], the following four typical working conditions shown in Table 1 were selected to study the matching of suspension damping and air spring.

Table 1. Four typical working conditions

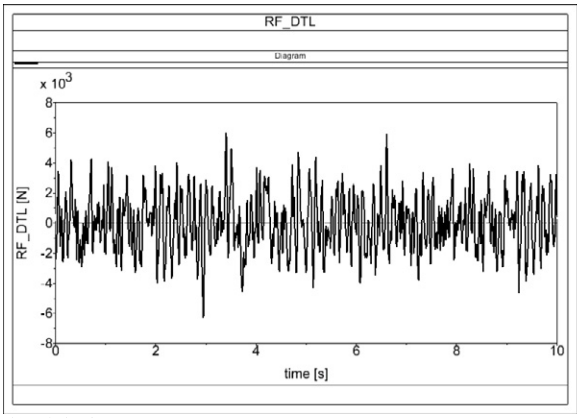
| Working condition | Description | Working condition | Description |
|---|---|---|---|
| Case 1 | No-load, B-class road, 50*km/h* | Case 3 | Full-load, B-class road, 50*km/h* |
| Case 2 | No-load, B-class road, 80*km/h* | Case 4 | Full-load, B-class road, 50*km/h* |

Because of the space constraint, the paper only introduced the matching of the front shock absorber damping and air spring. In the paper, three performances, the acceleration of the body representing riding comfort, the suspension working space (*SWS*) affecting body posture and related with structural design and layout, and dynamic tire load (*DTL*) representing tire grounding performance, were selected as the key performance indicators to study the matching.

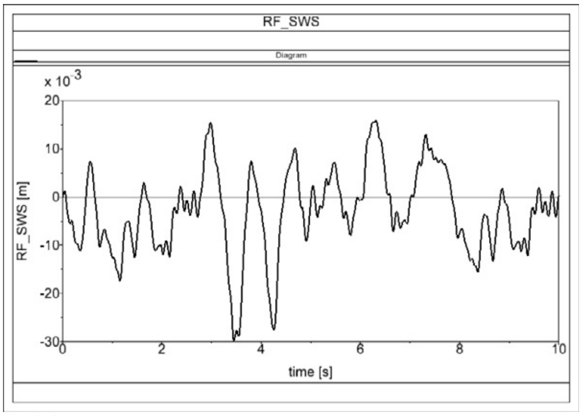
On the basis of full vehicle multi-body dynamics model, the rear shock absorbers' damping remained the initial value *16600(N·m/s)*, and the front shock absorbers' damping was changed to match the damping and air suspension. Under the condition of case 1, in which the rear shock absorbers' damping was set *6000(N·m/s)*, the simulation curves of body vertical acceleration, right front *SWS*, right rear *SWS*, right front *DTL* and right rear *DTL* are shown in Fig.3.

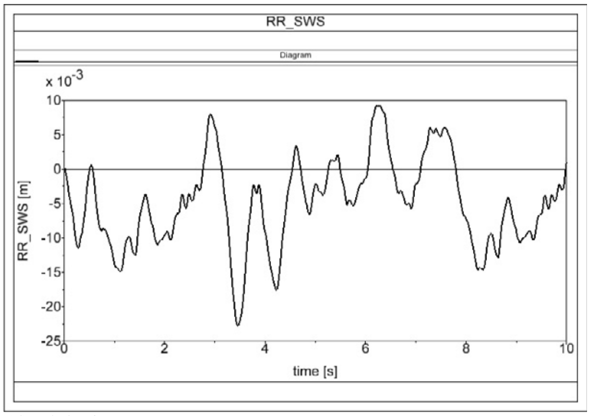



(a)Body vertical acceleration

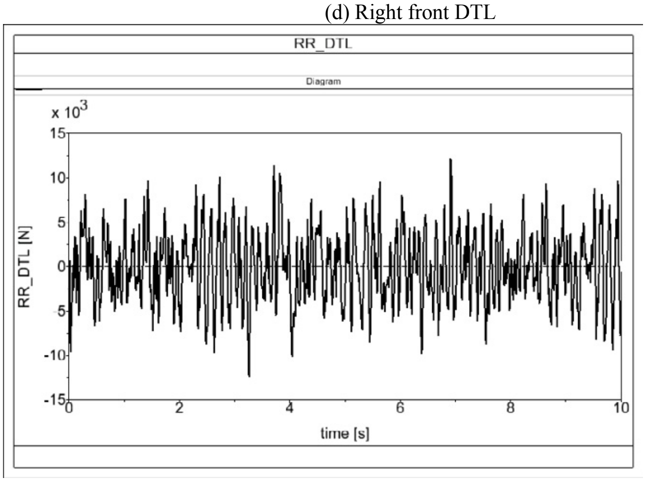



(b) Right front SWS




(c)Right rear SWS (d) Right front DTL




(e) Right rear DTL

Fig.3. Simulation curves

The simulation curves under different conditions could be obtained in the same way but not listed here respectively. The root mean square(*RMS*) values from the simulation curves' data are shown in Table 2. From the data in the table, a conclusion could be obtained that a smaller front suspension damping could reduce the body vertical acceleration, but increase the suspension *SWS*. On the basis of comparison of simulation data, a suitable damping was selected for a good matching with air spring under corresponding conditions.

## 5. Conclusions

The paper designed 2D drawings by *CATIA* to obtain assembly drawings first, accurately measured the vehicle positioning parameters and mass parameters to lay the foundation for the establishment of accurate vehicle multi-body dynamics model, and then analyzed the connection of various components and motions to construct the vehicle topology. Lastly, the paper established the full vehicle multi-body dynamics model in *SIMPACK*.

On the basis of the vehicle multi-body dynamics model, under four typical operating conditions, different damping was calculated to match air spring, and a suitable damping was selected for a good matching to improve the performance of the suspension system.

Table 2. Compared with RMS values in different conditions

| Working condition | Parameter | Front damping ($N \cdot m/s$) | | |
|---|---|---|---|---|
| | | 6000 | 8500 | 11000 |
| Case 1 | Body vertical acceleration ($m/s^2$) | 0.7174 | 0.7477 | 0.7772 |
| | Right front *SWS* (*m*) | 0.0082 | 0.0077 | 0.0074 |
| | Right rear *SWS* (*m*) | 0.0072 | 0.0070 | 0.0069 |
| | Right front *DTL* (*N*) | 1884 | 1776 | 1720 |
| | Right rear *DTL* (*N*) | 4082 | 4102 | 4122 |
| Case 2 | Body vertical acceleration ($m/s^2$) | 0.8680 | 0.8777 | 0.8937 |
| | Right front *SWS* (*m*) | 0.0109 | 0.0099 | 0.0091 |
| | Right rear *SWS* (*m*) | 0.0083 | 0.0079 | 0.0077 |
| | Right front *DTL* (*N*) | 2347 | 2263 | 2237 |
| | Right rear *DTL* (*N*) | 4789 | 4771 | 4759 |
| Case 3 | Body vertical acceleration ($m/s^2$) | 0.4939 | 0.5165 | 0.5398 |
| | Right front *SWS* (*m*) | 0.0138 | 0.0133 | 0.013 |
| | Right rear *SWS* (*m*) | 0.008 | 0.0078 | 0.0078 |
| | Right front *DTL* (*N*) | 2062 | 1959 | 1913 |
| | Right rear *DTL* (*N*) | 4113 | 4132 | 4149 |
| Case 4 | Body vertical acceleration ($m/s^2$) | 0.6294 | 0.6242 | 0.6253 |
| | Right front *SWS* (*m*) | 0.0161 | 0.0151 | 0.0144 |
| | Right rear *SWS* (*m*) | 0.0089 | 0.0085 | 0.0082 |
| | Right front *DTL* (*N*) | 2589 | 2499 | 2476 |
| | Right rear *DTL* (*N*) | 4807 | 4806 | 4806 |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