Vibration Analysis Based on Full Multi-Body Model for the Commercial Vehicle Suspension System

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Abstract: - Based on the characteristic of a type of commercial vehicle, a multi-body model of its full suspension system with twenty Degree of Freedom (DOF) was established in MSC.ADAMS. After evaluating relative condition parameters such as stiffness and damping index on the simulation model, the effects of outside exciting on the suspension system have been worked out numerically. The harmonic response of the mass center of the seat at model coordinate space provides reference for the general performance of the suspension system which enable the future design improving.

Key-Words: - Suspension system; Commercial vehicle; Vibration; Simulation; Virtual prototype; MSC.ADAMS; Frequency domain response;

1 Introduction

The vehicle suspension system, carrying the vehicle body and transmitting all forces between vehicle body and road, is responsible for driving comfort and passengers' ride safety while maintaining the car to be safe and stable as the suspension. A develop of the suspension system of commercial vehicle, which is designed to fulfil a strict design specification amd must behave in a safe and predictable manner and pevide a reasonable degree of comfort, requires the parallel design of the actuators, sensors, electronic hardware, software and control algorithms. Each of these sub-system impings on the designs of others, so modeling and simulation techniques can be employed at the initial stages of the project.

A variety of research projects and publications deal with different types of suspension systems have been discussed, as in [l]- [3] apply linear and non-linear optimal control theory to obtain compromised solutions. An active suspension system for vehicles using Neural Networks(NN), Genetic Algorithm(GA) and Linear Quadratic Gaussian(LQG) controls have been proposed, and with a four-degree freedom suspension vibration model described by a nonlinear system, the performance with the NN and GA controller were compared with the performance with the LQG controller [4].The harmonic response of quarter car suspension systems with MR dampers was analyzed, which was a two DOF isolation system with MR damper [5].

With regards to the application of simulation and design techniques to the development of a prototype suspension system, many novel actuation technique within tight cost and time constrains are employed. The common simulation models for the suspension system include the quarter-vehicle model with tow DOF, the half-vehicle model with four DOF and the full-vehicle mode with seven DOF[6]. A multi-body dynamic analysis model for commercial vehicles is developed and the ride sumulation is carried out in [7].

The traditional process of suspension system design, including the conceptual design, project verification, product design, sample testing, and then modification design again is time and money consuming and costly. Conceptual prototyping technology and computer simulation can enable us to obtain the general performance and capability of a component or vehicle system under given loading conditions, such as the frequency response of the system, prior to the prototype manufacturing and testing. Second, it can reduce the time required to investigate and some of the safety risks associated with prototype building and testing. All these factors contribute to improving confidence in a design and achieving a higher added value to the product.

During those general computer simulation analysis methods, finite element analysis (FEA) has been an integral part of the design and development process for vehicles for many years. FEA is applied to ensure that all parts, linkages and systems, which make up a vehicle, are strong enough to withstand the work loads. FEA looks at the stresses and deflections of parts, to ensure that they are strong enough for their intended uses. Whereas multi-body simulation (MBS) software packages such as ADAMS look at the dynamic behaviour and interaction of mechanisms and systems, under loads and conditions that replicate real work condition. ADAMS is a family of software packages for analyzing the dynamic behaviour of physical systems and mechanisms. ADAMS View is the main software package, and can be used to build models of almost any mechanism. This model can then be subjected to numerous tests to analyse and alter the dynamic behaviour of the vehicle.

In this paper, taking the suspension system of a commercial vehicle as the object, vibration simulation i.e. the frequency response of the system under typical load condition is carried out.

2 Modeling of the Suspension System

The degree of accuracy with which these components are modelled depends on the information available to create the model, the degree of accuracy required from the results and the time available for simulations. If, for example, the analysis is being carried out before the detailed design of the vehicle, then there may not be sufficient data to build highly accurate models. This will mean that the absolute accuracy of the simulation in comparison with a real vehicle may not be guaranteed; however, the general behaviour of design concepts can be understood before commitment to a final design. On the other hand, if the analysis is being undertaken on a completed design, there may be highly detailed information to build the model and results may be accurate, but there may be little scope for improving the behaviour and performance of a design.

2.1 General Modelling Strategy and Processing

Before building an ADAMS model of a full vehicle, the available data has to be collated and decisions made about the requirements of the simulation output. The basic data needed for a vehicle model is listed as follows:

1) Design drawings of the suspension system;

2) Compliant component data such as stiffness and damping value, alignment and location;

3) Tyres and road data;

4) Mass/inertia data of the components of the suspension system;

5) Applying relevant measures and outputs from the model;

6) Inputting controls data.

It is essential that attention to detail is paid in all stages of the model building process, for instance: the constraints applied to the model must be correct to ensure no over-constraint; the geometry, orientation, location, mass and inertia of parts must be specified correctly: stiffness and damping values must act in the appropriate direction between contact points.

Moreover, when building a simulation model of the suspension system, care needs to be exercised on the level of detail in the model. If the model is too detailed then the simulation solution times will become excessive and the results may be difficult to interpret. If on the other hand the simulation model has insufficient level of detail, considerable care needs to be exercised in interpreting the results. In respect to the simulation solver, the time step size should be set so that important behavioural characteristics and events are not overlooked.

2.2 Structure of the Suspension System

To build a model of a commercial vehicle suspension system and analyse its dynamic behaviour, the individual components of the vehicle need to be represented. Commercial vehicles have many components that are not found on ordinary road cars. For example, springs such as multi-leaf and air springs; multi axle vehicles may have four or more axles, whereas a car will only have two.

In ADAMS/View, build the models of all parts, assemble each part to another with corresponding joints, adjust several redundant constraints, and define spring and bushing force. The results of the model verification are detailed as the follows:

1) The model is a conceptual vehicle suspension with twenty DOF.

2) The model has twenty moving parts (not including ground) i.e. wheel, link, fl_up_link, fl_wheel and fl_wheel_attach etc.

3) The seat is a cylinder mounted to the chassis with bushing force.

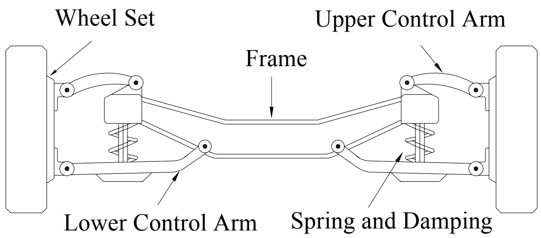

Fig. 1: Schematic of the Suspension Mechanism

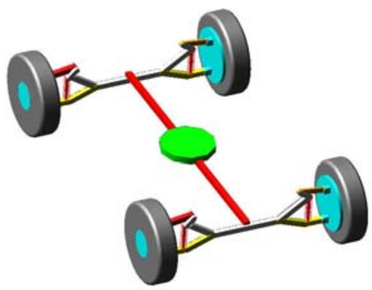

Fig. 2: ADAMS Model

4) Each tire is connected to the ground with bushing force.

Fig. 1 shows the schematic of the vehicle suspension mechanism, while Fig. 2 shows the simulation model in MSC>ADAMS.

2.3 Model Adjustment

It is proposed that each part should be linked to another with a rotation joint, except that the two mid-links are connected to the chassis with fixed joint, namely, there would be twenty revolute joints. However, actually, in order to remove the redundant constraints, several joints have to be adjusted to other types like cylindrical or spherical joint while maintaining the actual number of DOF. It is a requirement for vibration analysis that there should be no redundant constraints in the model[8]. Therefore, the adjusted model holds four cylindrical joints, twelve rotation joints, four spherical joints and two fixed joints.

Table 1 Definition of stiffness and damping coefficients

Items	Design variable	Comments	Items	Design variable	comments
Spring damping	DV_k	All of the four springs' damping coefficients are the same.	Bushing stiffness	DV_k_x, DV_k_y, DV_k_z DV_rk_x,DV_rk_y, DV_rk_z	Translational stiffness coefficients for all the bushings. Torsinal stiffness coefficients for all the four bushings.
Spring stiffness	DV_c	All of the four springs' stiffness coefficients are the same.	Bushing damping	DV_c_x, DV_c_y, DV_c_z DV_rc_x, DV_rc_y, DV_rc_z	Translational damping coefficients for all the four bushings. Torsinal damping coefficients for all the four bushings.

2.4 Parameterized Model

All the damping and stiffness coefficients in the model have been defined into design variables. So this parameterization could facilitate the later research and optimization of the conceptual model. Table 1 shows the details of each design variable:

3 Vibration Simulation Results

For frequency response simulation in MSC.ADAMS, there are several relative concepts detailed as follows [8] [9] [10].

3.1 Sweep sine function

In this paper, the vibration actuator is defined as a swept sine function, which is a constant amplitude sine function being applied to the model.

$$f(\omega) = F \times [\cos(\theta) + j \times \sin(\theta)] \qquad (1)$$

where $f(\omega)$ is the forcing function; F is the magnitude of the force; and θ is the phase angle.

The amplitude and the starting phase angle of the swept sine function here are 1000N and 0 degree respectively.

3.2 Frequent response

Assuming $u(t)$ and $y(t)$ are the input and the output of the system respectively, the linearized model is represented as follows:

$$\dot{x}(t) = Ax(t) + Bu(t) \qquad (2)$$

$$y(t) = Cx(t) + Du(t) \qquad (3)$$

Where, t is the time domain variable. $x(t)$ is the state space vector. Thus the system transfer function can be represented as follows:

$$\mathrm{H}(s) = \frac{y(s)}{u(s)} = C(sI - A)^{-1}B + D \qquad (4)$$

where s is the Laplace variable, H(s) is the transfer function of the model, and I is the identity matrix of dimension equal to the number of system states.
For a given vibration analysis, the system frequency response is given as (5):

$$y(s) = H(s)u(s) \qquad (5)$$

3.3 Forced vibration analysis

First, load the vibration tool through the tools->plug-in. Then, define four input-channels at the mass center of each wheel, three output-channels at the mass center of seat and a swept sine function actuator. They are input-channel-fl, input-channel-fr, input- channel-bl,input-channel-br, output-channel -dis, output-channel-vel, output-channel-acc, respectively. After that, define the actuating frequency range as 0.1 HZ to 200 HZ, and the total simulation steps 2000 in order to grasp the exact response peak. Last, run a static forced vibration analysis.

3.4 Results analysis

Fig. 3 describes the frequency response of the seat mass center, where the solid, dot and dash represent the acceleration response, the velocity response and the displacement response respectively. The four main peaks of the frequency response are at 6.5 Hz, 35.4 Hz, 106.2 Hz, and 150.9 Hz, of witch the magnitude of the system displacement transfer function are 28dB, 55dB, 30dB and 5dB respectively.

Fig. 4 and Fig. 5 show the 7th and 15st frame of the forced vibration analysis respectively, at 35.4Hz.

Also, from modal coordinates table at different frequencies, the information that which mode contributes most can be found out. Only the domain parameters of the first model and second mode are listed in Table 2 and Table 3 respectively.

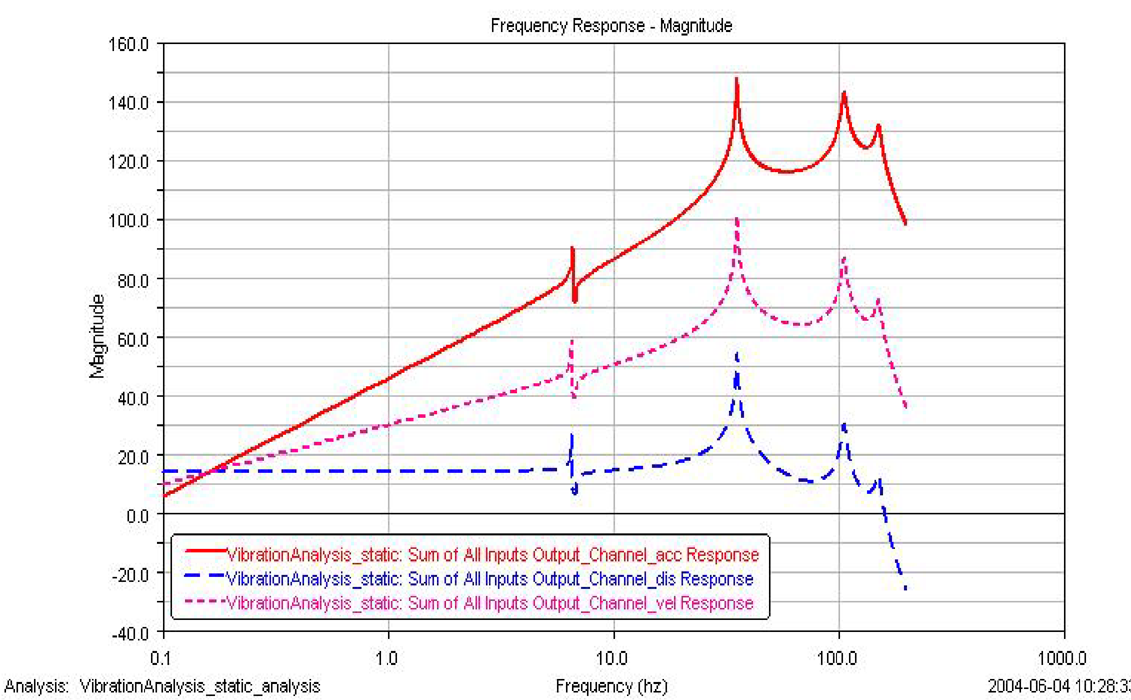

Fig. 3: Frequency Response of the Seat Mass Center

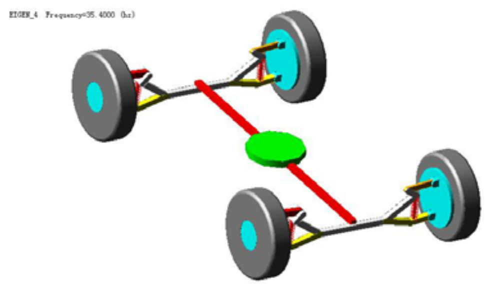

Fig. 4: The 6th Vibration Frame

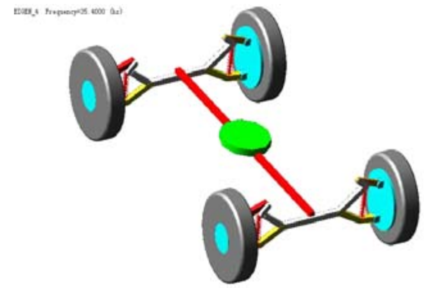

Fig. 5: The 15st Vibration Frame

Table 2: Modal Coordinates of Vibration Analysis (F = 35.4 Hz)

Model	First Order
put_Channel_fl	0.918505
put_Channel_fr	0.905475
put_Channel_bl	0.91852
put_Channel_bl	0.905496

Table 3: Modal Coordinates ofVibration Analysis (F = 106.2 Hz)

Mode	Second Order
put_Channel_fl	0.402594
put_Channel_fr	0.359085
put_Channel_bl	0.403507
put_Channel_br	0.357929

4 Conclusion

The peaks in the frequency response, transfer function, and modal coordinates could be easily analysis achieved through the vibration analysis of the model in the environment of ADAMS/Vibration.

The parameterization of the Springs and bushings could facilitate later optimization to reduce the vibration peaks in the frequence domain response. Since the integration of flexibility analysis and vibration analysis are feasible, future study could take flexibility of the important parts into consideration so as to achieve the results close to the actual model.

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