MODELING SUSPENSION DAMPER MODULES USING LS-DYNA

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ABSTRACT

A suspension damper module is an integration of the vehicle chassis components that control suspension motion. Various analytical tools are currently used in vehicle suspension analyses and component designs. However, these tools are not particularly useful for analysis of proposed damper module designs since they typically do not accurately represent component interactions and force distributions within the module. This paper describes a finite element model of a suspension damper module using an explicit finite element code, LS-DYNA. The modeling techniques used to construct the components within the modular assembly are presented. The LS-DYNA model was first correlated with a side load test of the damper module on an MTS machine. The analysis results were also compared with those from an ABAQUS model of the same module. A parametric study was then conducted to investigate the effects of some design parameters on the piston side load in the damper system.

INTRODUCTION

The primary functions of a damper module are to support the vehicle mass, control vertical vehicle motion, and define suspension travel limits. Damper module assemblies are commonly used in McPherson strut suspensions as well as other suspension types. As illustrated in Figure 1, a strut damper module typically consists of a strut damper, coil spring, spring seats, spring isolators, top mount, steering bearing, and jounce bumper, although component content may change depending on the application.

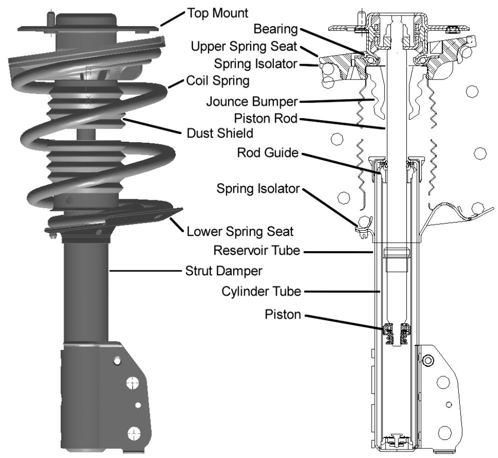


Figure 1. A typical strut damper module assembly.

A major consideration in the damper module design process is the reaction of the various components to severe loading conditions. An example is the reaction of the damper to applied bending moments. The bending moment will impart a normal force, or side load, to the damper piston. An increase in the magnitude of the side load will increase the breakaway friction force necessary to cause the piston to move, which can affect the ride comfort of the vehicle. A high level of side load may result in undesirable noise and vibration and may even affect the life expectancy of the damper module in severe cases. Proper management of these forces is necessary to insure that a given damper module design provides the desired levels of ride comfort as well as long service life.

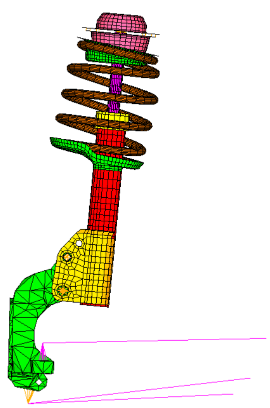
Various analytical tools are currently used to analyze McPherson strut suspension and component designs. However, due to the lacking of accurate representations of complex component interaction, these tools are not particularly useful for analysis of proposed damper module designs. It has become evident that improved damper module analysis techniques will be required for future design efforts to meet anticipated increases in performance and reliability targets.

For a quasi-static structural problem, a finite element analysis software based on an implicit integration algorithm is usually a good choice from the standpoint of computational efficiency. It requires much less time to proceed the analysis due to the large increment allowed in an implicit integration algorithm. However, due to complex interactions within a damper module, contacts between the components at many locations have to be modeled in a damper module analysis. These contacts present significant difficulties in achieving numerical convergence during the solution process, especially in a parametric study when many parameters are varied that will likely complicate the interactions between the components. On the other hand, an explicit finite element code, such as LS-DYNA, does not have convergence problem in solution process, which makes an explicit finite element code more attractive in solving problems that would possess severe non-convergence in an implicit code. In this study, LS-DYNA was used to model a suspension damper module under quasi-static loading and the model was correlated with a damper module test. The results were also compared with those from an ABAQUS model. A parametric study was then performed to investigate the effect of design changes on the piston side load.

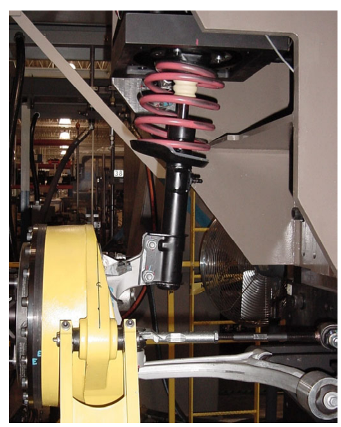
APPROACH

Figure 2(a) shows the LS-DYNA model of the damper module. The model includes major components of a damper module. The strut damper reservoir and cylinder tubes, upper and lower spring seats, and top mount housings were modeled using shell elements. The damper piston rod, piston, and rod guide were modeled using solid elements. The coil spring was also modeled using solid elements. For simplicity, the top mount cushion assembly was modeled as a nonlinear discrete beam (material type 67) that defines stiffness and damping effects in each of six degree-of-freedom. The knuckle was also included in the model as a rigid. The lower control arm and tie rod were represented by three rigid beams. The ball joints that connected the control arm and tie rod to the knuckle were modeled as two spherical joints. Nonlinear discrete beams were also used to represent the stiffness of the three bushings that connect the control arm and the tie rod to the vehicle. The jounce bumper located between the upper spring seat and the damper tube upper surface is an important component in a damper module. It functions as the suspension travel stop in compression. It could be modeled as a nonlinear discrete beam. However, for correlation purpose, the jounce bumper was not included in the results discussed below, because in the test the jounce bumper was not activated due to the test set-up limitation.

An unloaded damper module assembly contains the coil spring in a pre-compressed state. Therefore, the modeling of the damper module has to include an assembling process before the module is subjected to various loading conditions. To simulate this pre-compressed condition, the coil spring was modeled in its free state. After the spring was compressed to the assembled state, a constraint was activated to lock the spring into current position. The damper was then brought to design position. Subsequently, the wheel center was moved to jounce or rebound position to simulate different loading conditions.



(a)



(b)

Figure 2. (a) LS-DYNA model of the damper module and (b) side load test set-up.

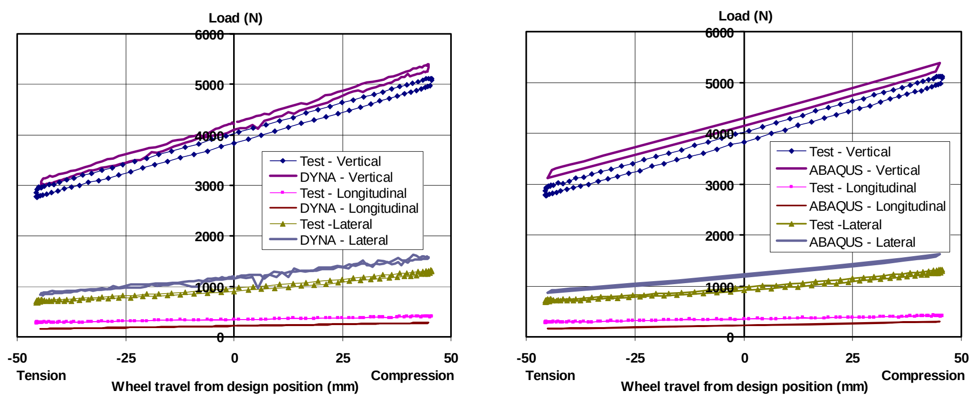
To correlate the analytical model, a side load test was conducted on an MTS329 multi-axial test machine (Figure 2(b)). The load transferred to the tower was obtained by three load cells located on the top of the damper module that measure forces in the vehicle vertical, lateral, and longitudinal directions. The load at the wheel center was measured using a Spinning Wheel Integrated Force Transducer (SWIFT^{TM}). A set of strain gages was instrumented on the piston rod to obtain the side load in the lateral and longitudinal directions. Due to the test set-up limitation, the damper module was loaded in the ±45 mm range from the design position.

The lab test conducted was a quasi-static one in which a 0.005-0.01 Hz loading rate was used. This loading rate requires 100-200 seconds for one cycle of the wheel travel, from the design position to jounce, then to rebound, and then back to the design position. Several trials for the LS-DYNA model revealed that, with an appropriate global damping and mass scaling, this loading sequence, including the assembling process, could be simulated as a 5-second event without introducing significant dynamic effect. Due to the large number of time steps needed for such a long simulation, a double precision version (Version 960) of LS-DYNA was used to minimize the round-off error in solution.

DISCUSSION OF RESULTS

Comparison With Test Data And The Results From ABAQUS Model
The same damper module was also modeled using an implicit finite element code ABAQUS/Standard (version 6.1). The modeling techniques used in the ABAQUS model was discussed elsewhere (Tao, 2000).

Tower Forces. Figure 3(a) shows the comparison between the DYNA model and the test for the forces at the tower where the top mount attached to the vehicle. Figure 3(b) compares the ABAQUS model with the test data for the tower forces. By comparing Figure 3(a) and 3(b), it can be seen that the results from the DYNA model and the ABAQUS model are very comparable. Both finite element models predicted slightly higher forces in the vertical and lateral directions.



(a) (b)

Figure 3. Tower forces vs. wheel travel.

Force At the Wheel Center. Figure 4 shows the vertical force at the wheel center vs. wheel travel. The predictions from the DYNA model and the ABAQUS model are plotted in the figure. Comparing to the test data, it is seen that the DYNA model and ABAQUS model both correlate the test data well for the wheel force.

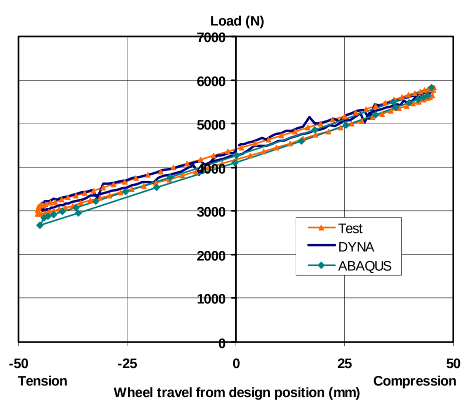
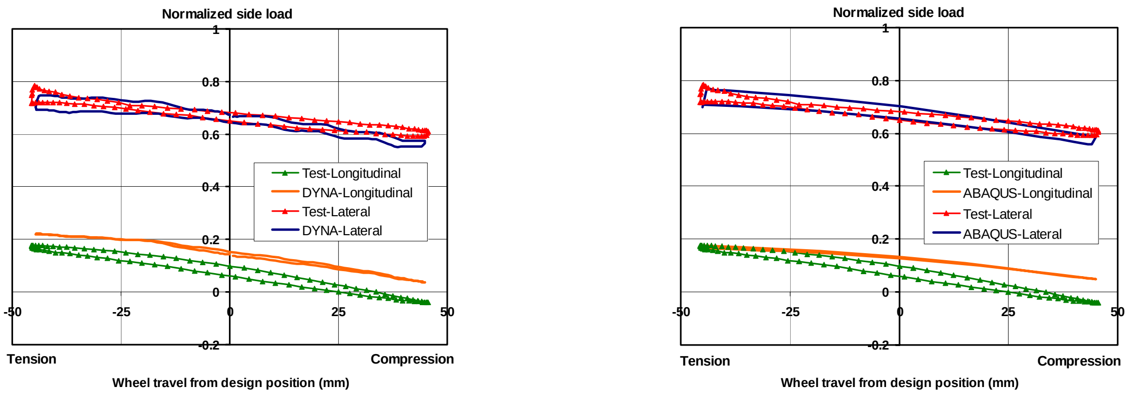


Figure 4. Force at wheel center vs. wheel travel.

Piston Side Load. The bending moment applied to the damper in a McPherson strut suspension results in the generation of a normal force, or side load, at the interfaces between the damper piston and cylinder tube as well as between the piston rod and rod guide. The side load results in a friction force that must be overcome in order for the damper to operate. The piston side load can be decomposed into two components in the vehicle later and longitudinal directions. Figure 5 shows the piston side load predicted by the DYNA model (a) and the ABAQUS model (b), compared with test measurement.

It is interesting to notice the piston side load curve predicted by the DYNA model exhibits some non-smoothness. A mesh density study of the DYNA model revealed that the non-smoothness of the side load curve was mainly caused by the coarse meshes on the circular piston and the cylinder tube that are in contact. It was found that a much finer mesh resulted in a smoother side load curve. However, due to the decrease of the element size and the increase of element number, the analysis with finer mesh requires a significantly longer time. It is seen that the side loads in the lateral direction predicted by the two models match the test measurement. On the other hand, the prediction on the longitudinal direction is higher than the measurement. More investigation is needed to better correlate the finite element models with the lab test or vehicle test.



(a) (b)

Figure 5. Piston side load vs. wheel travel.

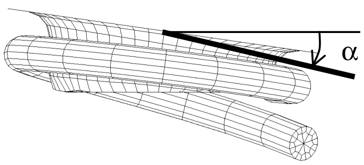
Parametric Study

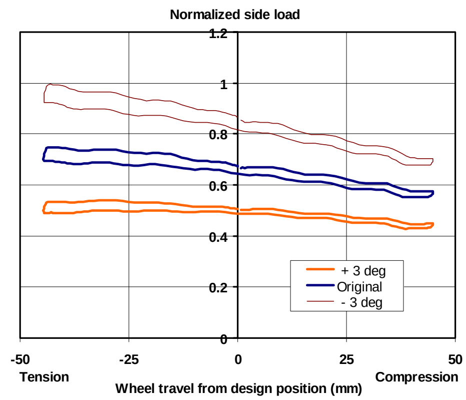
Once a preliminary correlation between the finite element model and the lab test data was established, the LS-DYNA model was used to conduct a parametric study to gain knowledge on the factors that affect the side load and to provide suggestions to the design process. An advantage of the explicit LS-DYNA model over the implicit ABAQUS/Standard model for parametric study of the damper module is the non-existence of convergence problem.

With a specified suspension, varying design parameters of the module can change the characteristics of the damper module. To achieve a lower side load in the damper, the spring seat orientations and the center of the spring seats offset from the damper axis can be adjusted. The shape of the coil spring can also be optimized to reduce the side load in the damper (Gotoh, 2000). In this study, three parameters were varied to investigate their effect on piston side load: the orientation of the upper spring seat in the lateral direction; the orientation of the lower spring seat in the lateral direction; and the translational offset of the lower spring seat in the lateral direction. The reasons for choosing these parameters are based on the following considerations. First, the variation of these three parameters is easier to implement in the FEA model. Second, the lateral side load is higher than the longitudinal one. It is more effective to change the parameters in the lateral direction.

Effect of Upper Spring Seat Orientation. Figure 6(a) shows an illustration of the upper spring seat. The orientation of the upper spring seat was changed 3^{o} counter-clockwise (-) and 3^{o} clockwise (+) from the original design.

The lateral component of the piston side load is shown in Figure 6(b). It is seen that with the increase in the inclination angle, α, the side load is reduced significantly. By increasing the orientation angle of the upper spring seat, the coil spring is compressed more on the inboard side at the top of the spring, thus causing a larger angle of the force line of action with respect to the damper axis. This angle change provides more load compensation to the bending moment resulting from the vertical load at the wheel or tie-patch.



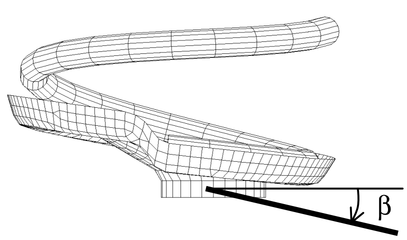


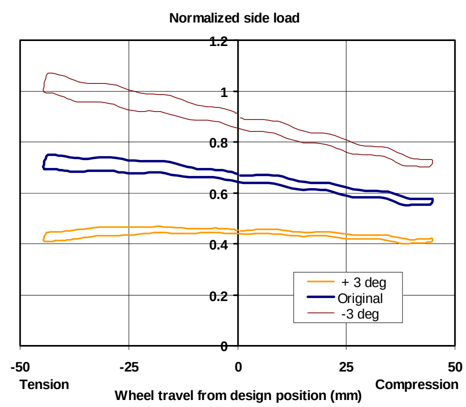
(a) (b)

Figure 6. (a) Orientation of the upper spring seat and (b) its effect on the lateral piston side load.

Effect of Lower Spring Seat Orientation. Figure 7(a) shows an illustration of the lower spring seat. The orientation of the upper spring seat was changed 3^o counter-clockwise (-) and 3^o clockwise (+) from the original design.

The effect of the lower spring seat orientation was shown in Figure 7(b). It can be seen that adjusting the lower spring seat orientation has a similar effect as adjusting the upper spring seat orientation. By increasing the orientation angle of the lower spring seat, it causes the coil spring compressed more on the outboard side at the bottom of the spring, thus also causing a larger angle of the force line of action with respect to the damper axis. Compared to the results of changing upper spring seat orientation, it seems more sensitive to adjust the lower spring seat orientation.





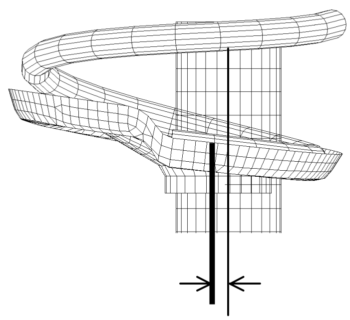
(a) (b)

Figure 7. (a) Orientation of the lower spring seat and (b) its effect on the lateral piston side load.

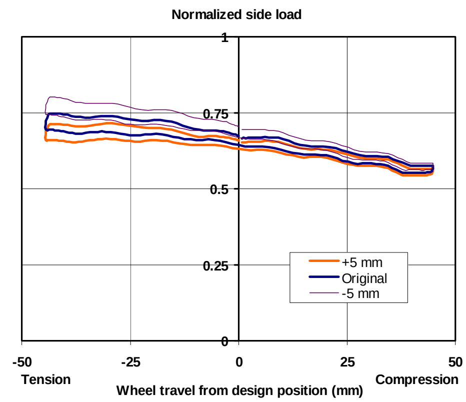
Effect of the Lower Spring Seat Translational Offset. Figure 8(a) illustrates the variation of the lower spring seat translational offset from the damper axis in the lateral direction. The offset was varied +5 mm (outboard) and -5 mm (inboard) from its original position.

The effect of the lower spring seat translational offset is shown in Figure 8(b). With the offset toward outboard, the side load is reduced, vice versa. However, with a 5 mm change, the lateral side load is not altered significantly.

In Figure 6(b), 7(b), and 8(b), only the lateral side loads are plotted. It was found that, in the wheel travel range considered, these three parameters have much less effect on the piston side load in the longitudinal direction.



(a)



(b)

Figure 8. (a) Translational offset of the lower spring seat with respect to damper axis and (b) its effect on the lateral piston side load.

SUMMARY

LS-DYNA was used for modeling a strut damper module under a quasi-static loading. The absence of convergence problem in using LS-DYNA code offers an attractive advantage for a parametric study in which an implicit finite element code would likely encounter severe convergence problems due to complex contact interactions. However, due to the nature of LS-DYNA or other explicit finite element codes, the modeling of a non-dynamic event using an explicit code requires a significantly longer time than using an implicit code.

The results from the LS-DYNA model are comparable with those from the ABAQUS model. Both of them closely match most of the results from a side load test of the damper module on an MTS machine. The piston side loads predicted by the LS-DYNA model exhibit some non-smoothness due to the mesh size and dynamic effect. The parametric study of three design parameters revealed that varying the lower spring seat orientation has similar but slightly stronger effect as varying the upper spring seat orientation. The lateral translational offset of the lower spring seat has relatively less influence on the piston side load. More investigation is needed to better correlate finite element models with lab test or vehicle test.

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REFERENCES

Gotoh, T. and Imaizumi, T. (2000), “Optimization of Force Action Line with New Spring Design on the McPhersion Strut Suspension for Riding Comfort.” SAE 2000 World Congress, paper # 2000-01-0101.

Tao, J.J. and Bishop, T.A. (2000), “System Modeling of A Damper Module.” SAE 2000 World Congress, paper # 2000-01-0727.