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ANALYTICAL AND EXPERIMENTAL ASSESSMENT OF HEAVY TRUCK RIDE

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ABSTRACT. This study is the second phase in a combined analytical and experimental effort to characterize and improve the ride quality of the Department Of Energy tractor/trailer. The discussion includes a brief overview of the finite element model of the vehicle and experimental road and modal test results. A novel system identification approach is used, employing both lab-based modal tests, and modal data derived using the Natural Excitation Technique (NExT), a scheme that utilizes the roadway surface as a natural forcing function. The use of a cab isolation system is investigated with the computer model for purposes of improving the ride quality of the vehicle. To validate these analytical predictions, an engineering prototype vehicle was developed, which included a cab isolation system, to experimentally assess ride quality. Ride quality improvements due to the addition of the isolation system are then assessed both experimentally and analytically, and the results are compared.

NOMENCLATURE

c	Scaling coefficient in analytical road surface model
f	Frequency in Hertz (Hz)
g	Unit of acceleration
rms	Root-mean-square
v_o	Constant velocity assumed in road surface model
$CMIF$	Complex mode indicator function
CSD	Cross spectral density function
DOE	Department of Energy
FRF	Frequency response function
$L(f)$	Insertion loss factor
MAC	Modal assurance criterion
$NExT$	Natural excitation technique
RSI	Ride severity index
$S(f)$	Power spectral density (PSD) used in RSI calculation
$\Phi(f)$	PSD used as analytical road surface input

1. INTRODUCTION

The Department of Energy (DOE) is committed to researching ideas for improved heavy truck ride, because a smoother ride is less fatiguing to the operators and, hence, safer to everyone on the road. Previous work funded by the DOE in this area [3], demonstrated that the addition of a cab isolation system may provide considerable ride improvements to DOE's existing fleet of tractor/trailers.

Designing tractor/trailer style trucks that exhibit good ride characteristics has been a challenge to automotive engineers for decades [7]. Hence, the use of a cab isolation system is not a new concept. Previous methods to improve cab-ride quality have included: altering the frame bending stiffness, introducing softer primary suspensions, and improving tire stability and driver seats [4]. The idea of a cab suspension system was first employed in heavy truck design more than three decades ago and consisted of an independent leaf spring assembly placed at each of the four corners of the cab [5]. Since then, the design of the typical cab suspension system has evolved into a fixed-front pivot combined with a spring/damper rear mount. It is this configuration that is commonly seen on the road today.

Previous conclusions about the possibilities of ride improvement discussed in [3] were based solely upon a model that was validated with laboratory-based modal tests. These tests used a very low level of excitation; it was known at the time that the tires and leaf springs added nonlinear behavior to the overall dynamics of the vehicle and would react quite differently when subject to the much larger level of excitation provided by the road surface. Hence, it was expected that the truck would behave much differently when exposed to an operational level of excitation. Laboratory modal tests, however, were the only means available to validate the computer model at the time. Dynamic modes of the vehicle have since been extracted using the Natural Excitation Technique (NExT) [9]; this dynamic state is more representative of the truck's operational environment. Correlating the computer model to these test results will allow a much more realistic prediction of the vehicle's ride characteristics.

The purpose of this work is threefold: (1), to illustrate the combined use of laboratory modal test results, modes extracted from the operating environment of the vehicle using NExT, and the truck's PSD response to validate the computer model, (2), to make a comparison between analytical predictions and experimental measurements of ride quality, and (3), to broaden the knowledge base in the areas of vehicle dynamics and proper testing techniques.

2. VEHICLE DESCRIPTION

Figure 1 illustrates a schematic of the DOE tractor, including some of the terminology commonly used in the heavy truck community. The driver is located in the cab, and any off-duty personnel lie on the bunk in the sleeper compartment. The

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Figure 1: The Department of Energy tractor.

vehicle is configured in a separate cab / sleeper arrangement, as compared to an integrated cab / sleeper design.

The MSC/NASTRAN finite element model of the DOE tractor is shown in Fig. 2. A standard, two-axle trailer connected to the tractor at the fifth wheel is also modeled, but not shown in the figure. See [3] for a detailed description of the finite element model development.

3. EXPERIMENTAL TESTING

In an effort to characterize the ride characteristics of the vehicle, two sets of experimental tests were performed. The first was the typical laboratory-based modal test, indicative of how the vehicle responds to low-level excitation. The second set of tests using NExT, were performed under the truck's normal operating conditions. The following two sections provide detailed descriptions of the two tests.

3.1 Modal testing

A low level, continuous-random force excitation signal was input to the vehicle through electrodynamic shakers at two locations on the vehicle; a 100 lb shaker was located at the

Figure 2: MSC/NASTRAN model of the DOE tractor.

Figure 3: Experimental FRF from modal tests.

front/passenger side of the tractor, and a larger 250 lb shaker was placed at the passenger side of the trailer, halfway back. The signals sufficiently excited the structure while keeping the dynamic response within the linear regime about the static equilibrium. Frequency response functions (FRFs) between the applied excitation forces and the measured accelerations were calculated and recorded. Modal frequencies, damping factors, and mode shapes were then estimated from the recorded FRFs. Prior to each test, the vehicle was rolled forward and back to overcome any stiction in the wheels and bearings. Figure 3 contains experimental FRFs for this test configuration.

3.2 Natural excitation testing

The tractor was instrumented with thirty one accelerometers, the majority of which were placed along the passenger side of the vehicle. Five displacement sensors were located along the passenger side of the vehicle at each of the five axle locations. These sensors measured the relative displacement between an axle and the frame directly above it. During all road testing, the vehicle was driven along Interstate 25 (I-25) south of Albuquerque at a constant velocity of 55 mph.

The cross spectral densities (CSD's) of the accelerometers were computed for various choices of reference accelerometers. A total of 1024 spectral points were used in the computation, along with a Hanning window function and overlap processing. Treating the CSD's as transfer functions, the Complex Mode Indicator Function (CMIF) was computed and is shown in Fig. 4 for several reference accelerometer choices. Here, the largest singular value is plotted along the y-axis, illustrating that there are relatively lightly damped structural modes at frequencies of 2.75 Hz, 3.75 Hz, and 5.50 Hz. There is a very heavily damped structural mode at 1.25 Hz, and a periodic response near 8.0 Hz, which is related to the tire rotation frequency.

A peak-picking strategy was adopted and applied to the CSD's to determine frequency and mode shape information.

Figure 4: Experimental complex mode indicator function from natural excitation tests.

Specifically, it is the magnitude and phase information of the CSD's that lead to the frequency and shape information. To illustrate, consider the mode at 3.75 Hz. Clearly from Fig. 4, the CMIF has the largest amplitude at 3.75 Hz when channel 22 is selected as the reference channel. Thus, at 3.75 Hz, the magnitude and phase information of the CSD's associated with channel 22 selected as the reference channel are used to determine the associated mode shape. This process is repeated with a different reference channel to obtain the magnitude and phase information related to the other frequencies. The magnitude gives the nodal amplitudes, and the phase is used to determine the sign of the amplitude: if the amplitude is near 0° then the amplitude is taken as positive; if the amplitude is near $\pm 180^\circ$, then the amplitude is taken as negative. A judgement call must be made for phase angles that significantly deviate from either of these extremes.

4. TEST/ANALYSIS CORRELATION

Before the computer model can be utilized to predict ride characteristics, correlation between measured test data and predicted analysis data is required. Comparing the model predictions to laboratory-based modal test results is the usual procedure, but this proved to be inadequate due to a number of nonlinear elements evident in the vehicle. One such element, the leaf spring suspension assembly located at the steer axle and trailer axles, is known to exhibit hysteretic behavior. In addition, a typical tire produces less spring rate when rotating than when motionless [8]. As a result, correlating the model to laboratory-based modal test results alone is not an acceptable approach due to the presence of these and other nonlinear qualities. Furthermore, using NExT test data as the only information to verify the computer model is not a viable alternative. Cross-spectra were measured between different locations on the vehicle using the road

surface as the input. The unknowns in the computer model cannot be determined, however, because these data sets do not contain sufficient information. The solution is to treat the vehicle as a quasi-linear system, behaving linearly in one regime under laboratory-level excitation, and linearly in a second regime under normal operating levels of excitation.

The computer model, correlated to the modal test results, is used as a starting point. Herein it is assumed that only those parameters behaving in a highly nonlinear manner will deviate from how they behave during laboratory modal tests. As a result, the NExT procedure is used to complement the modal tests, providing sufficient information to validate the model so it can predict responses due to natural operating conditions.

4.1 Correlating to modal tests

In order to obtain good agreement between the finite element model and modal test results, it was necessary to adjust the values of some uncertain parameters in the model. These included stiffnesses of the tires, airsprings, and leaf spring assemblies.

Updated parameter values were obtained using a Sandia in-house code PESTDY [1] (Parameter Estimation for STRuctural DYnamics). Refer to [3] for a more detailed discussion of this procedure. The corresponding modal assurance criteria (MACs) and frequency agreement are shown in Table 1. Because the correlation was performed using modal test data, the updated parameters are associated with small amplitude motions. It is expected that some stiffnesses, such as those of the leaf springs, would be reduced in over-the-road environments where stiction effects are overcome.

4.2 Correlating to natural excitation tests

Because the NExT method also provides modal frequency, damping, and shape information, the techniques discussed in §4.1 can be employed to the road data. However, and as noted above, neither the laboratory or road test data can be used alone to update the model. Therefore, using the model correlated to the modal test as a starting point, certain parameters were adjusted until the entire model matched the NExT results. Those elements in the model suspected of highly nonlinear behavior (*i.e.*, tire and leaf spring stiffnesses) were

Mode Description	Test/Analysis Frequency (Hz)	Measured damping (%)	MAC
bounce	2.04 / 2.01	3.36	0.867
tractor pitch	3.19 / 3.22	2.91	0.939
counter pitch	4.16 / 4.12	1.15	0.835
frame bending	5.96 / 6.06	2.10	0.830

Table 1: Results of model / modal test reconciliation.

Mode Description	Test/Analysis Frequency (Hz)	Measured damping (%)	MAC
bounce	1.25 / 1.98	26.0	0.661
tractor pitch	2.75 / 2.75	9.1	0.934
counter pitch	3.75 / 3.75	3.5	0.986
frame bending	5.50 / 5.50	3.1	0.783

Table 2: Results of model / NExT test reconciliation.

the only parameters changed in the effort to correlate to NExT results.

Table 2 illustrates the results of the model / NExT test reconciliation effort. Note that there is a large discrepancy in the modal frequency of the bounce mode. As illustrated by Fig. 4, the bounce mode was not sufficiently excited by the road surface, thereby making the extraction of the modal frequency, damping, and shape not a well defined task. The level of excitation is directly related to the vehicle's speed. Therefore, future road tests will be conducted at different speeds to ensure that all dynamic modes are sufficiently excited. For all the tests discussed here, the vehicle was operated at 55 *mph*; little effort went into correlating the computer model to the test results for the bounce mode for this reason.

The test and analytical mode shapes characterizing the vehicle under operating-level excitation are included in Figs. 5-8. An interesting concept to notice is how the model parameters change when correlating to modal or NExT tests. As illustrated by Table 3, the leaf springs on both the tractor and trailer are much stiffer when subject to the low-level excitation of a modal test. When the level of excitation increases, however, the leaves start sliding with respect to one another, and the stiffness of the entire assembly decreases. In addition, the slight decrease in the vertical tire stiffness shown in Table 2 is expected, because a rotating tire is slightly less stiff than a stationary one [8].

Figure 5: Bounce mode, experimental, (a), and analytical, (b).

Figure 6: Tractor pitch mode, experimental, (a), and analytical, (b).

Figure 7: Counter-pitch mode, experimental, (a), and analytical, (b).

Figure 8: Frame bending mode, experimental, (a), and analytical, (b).

5. ENGINEERING PROTOTYPE VEHICLE

For experimental ride quality studies, an engineering prototype vehicle was established by modifying one tractor in the current DOE fleet. A drawing of the prototype is included in Fig. 9. As illustrated in Fig. 10, a sub-frame assembly was added beneath the cab and sleeper. This approximates an integrated cab configuration, because the sub-frame ties the cab and sleeper together; they behave as a single unit on top of the vehicle's main frame.

The Cab-mate™ cab isolation system, provided by Link Manufacturing, Inc., is also shown in Fig. 10. The objective of this system is to isolate the cab occupants as much as

Parameter	Correlated to modal tests	Correlated to NExT tests
Tire vertical stiffness	6342, $\frac{lb}{in}$	6131, $\frac{lb}{in}$
Tire lateral stiffness	1988, $\frac{lb}{in}$	1982, $\frac{lb}{in}$
Tractor leaf spring stiffness	27320, $\frac{lb}{in}$	8063, $\frac{lb}{in}$
Air bag stiffness	1141, $\frac{lb}{in}$	1124, $\frac{lb}{in}$
Trailer leaf spring stiffness	47190, $\frac{lb}{in}$	23900, $\frac{lb}{in}$

Table 3: Parameter changes due to model update.

Figure 9: The engineering prototype.

possible from vibrations due to road surface roughness. Typically, a 2 dB reduction in the vibrational response of the cab has been observed. To help determine how vital the location of this system is, the sub-frame was assembled so as to allow the fore/aft position of the Cab-mate™ to vary between the aft

Figure 10: The cab sub-frame and isolation system.

Figure 11: Predicted cab pitch mode of prototype vehicle.

Figure 12: Predicted bounce mode of prototype vehicle.

Figure 13: Predicted tractor pitch mode of prototype vehicle.

Figure 14: Predicted counter-pitch mode of prototype vehicle.

Figure 15: Predicted frame bending mode of prototype vehicle.

end of the sub-frame and 33 inches forward. This vehicle was then used for all ride quality studies that follow. Specifically, the ride characteristics of three configurations were assessed: (1), *nominal* (cab is hard mounted to the frame), (2), *isolation #1* (cab isolation system is installed at the rear of the sleeper), and (3), *isolation #2* (cab isolation system is installed 33 inches forward of the rear of the sleeper).

Because it is this prototype vehicle that is to be used for all ride quality studies, appropriate changes were made to the finite element model to accurately represent all three configurations of this prototype vehicle. Analytical predictions of the dynamic modes of the prototype (*isolation #1* configuration) under operational-level excitation are shown in Figs. 12-15. A summary of the modal frequencies is shown in Table 4.

Mode Description	Iso #1 Analysis Frequency (Hz)	Iso #2 Analysis Frequency (Hz)
cab pitch	1.90	1.80
bounce	2.55	2.30
tractor pitch	3.04	2.89
counter pitch	3.77	3.75
frame bending	6.29	6.28

Table 4: Predictions of the prototype vehicle's over-the-road response.

6. RIDE QUALITY ASSESSMENT

The ride severity index (RSI), a weighted measure of the power of the output spectrum, was introduced in [3] and can be used to gauge the ride quality of the vehicle. Consider the following relation

$$RSI = \left[\int_{f_1}^{f_2} L(f)^2 S(f) df \right]^{\frac{1}{2}}, \quad (1)$$

where $S(f)$ is the output power spectral density function and $f \in [f_1, f_2]$ denotes the frequency band of interest. Note that the RSI is a scalar quantity that is simply a weighted sum of the output PSD. The insertion loss factor, $L(f)$, is a quantity indicative of the human body's sensitivity to vibration and is applied as a filter to the output response. The higher the loss factor, the more sensitive the human is to the excitation [2].

6.1 Experimental results

Three different tests were conducted, and within each test, eight different runs were performed. In the odd numbered runs of each test, data was collected as the vehicle traveled southbound along I-25, from mile marker 175 to 141. In the even numbered runs, data was collected as the vehicle traveled northbound along I-25, from mile marker 141 to 174. In each run of a particular test, time series ensembles of the sensor signals were averaged to obtain auto and cross correlation functions. The auto correlation functions were then used to compute power spectral densities (PSD's), from which a measure of the ride quality at the driver seat and bunk were determined.

6.2 Analytical results

To assess the ride characteristics of the prototype vehicle, the procedure discussed in [3] was employed here. The mass, damping, and stiffness matrices representing the vehicle from the finite element model were imported into the MATLAB environment. Transfer functions representing the output accelerations at the driver seat and bunk due to input excitation from the road surface at the axles were then computed.

An analytical model of the frequency spectrum of the road surface at a constant speed v_o , given by

$$\Phi(f) = \frac{v_o^c}{(2\pi f)^2}, \quad (2)$$

was then applied to recover PSD responses at the output locations of interest [6,10]. There was some discrepancy in the literature, however, about the value of the scale factor, c , for a paved road surface. In addition, the amount of correlation between the excitation at the left and right tracks of the vehicle was also an unknown quantity. This must be determined, because it dictates how much asymmetrical behavior will occur while the vehicle is in operation. In short, model validation can be inexact when the form of the input is unknown. Therefore, part of the reconciliation effort went into identifying the form of the road surface input so that it matched the measured road surface of I-25.

The ride severity was calculated both experimentally and analytically, for the three vehicle configurations. The results are tabulated in Tables 5-7. Experimentally, it is evident that the ride quality at the driver seat has improved by approximately 20% due to the addition of the cab isolation system, but remains unchanged at the bunk. Furthermore, the location of the isolation system does not affect the experimental measurements. Similar results are predicted by the computer model only when the isolation system has been moved forward. While the isolation system is positioned at the rear of the integrated cab, no improvement in ride is predicted.

	RSI at driver seat (mg_{rms})	RSI at bunk (mg_{rms})
Analytical	57.69	49.73
Experimental	59.32	50.99

Table 5: Ride quality for nominal configuration.

	RSI at driver seat (mg_{rms})	RSI at bunk (mg_{rms})
Analytical	57.49	55.26
Experimental	50.05	48.30

Table 6: Ride quality for Iso #1 configuration.

	RSI at driver seat (mg_{rms})	RSI at bunk (mg_{rms})
Analytical	51.18	49.01
Experimental	50.50	47.86

Table 7: Ride quality for Iso #2 configuration.

Figure 16: Measured PSD response at driver seat.

Figures 16 and 17 illustrate the vertical PSD response at the driver seat from the measured test results and analytical predictions, respectively. There are several discrepancies between the experimental results shown in Fig. 16 and the analytical results shown in Fig. 17. First, the predicted bounce mode occurs at a higher frequency than measured in the test. As discussed above, this is because the test results for this mode were somewhat suspect, due to insufficient excitation of the bounce mode for accurate measurement. Second, all predicted dynamic modes increased in frequency when the isolation system was introduced (see Table 4). This was not the case, however, for the experimental test results. As illustrated in Fig. 16, the cab isolation system reduces the tractor pitch mode at 2.75 Hz, but leaves the other modes unaffected, implying that the response of the cab isolation system is uncoupled from the dynamics of the rest of the vehicle. In

Figure 17: Predicted PSD response at driver seat.

addition, the experimental frequencies appear independent of the physical location of the isolation system. In the model, the modes near 2 Hz shift down in frequency as the isolation system is moved forward; this makes sense because as the system moves forward, the resultant load on the isolation system increases. Third, the experimental results reveal a 5 dB reduction in the amplitude of the tractor pitch mode due to the cab isolation system. The analytical model, however, only predicts a 3 dB reduction.

7. CONCLUSIONS

A computer model of the DOE tractor/trailer has been validated using a combination of modal and road test data. Experimental results from each test were shown to be quite different due to the high degree of nonlinearity present in the dynamics of heavy trucks. As a result, model reconciliation efforts proved to be anything but straightforward. Even so, the computer model proved invaluable in explaining many complex features of heavy truck dynamics.

Ride quality has been measured both experimentally and analytically, with and without the use of a cab isolation system. When considering ride quality, the pitch mode at 2.75 Hz is the only major contributor, as evidenced by the magnitude of the PSD response in Fig. 16. The analytical model does a good job of predicting the over-the-road response of this mode, leading to an adequate prediction of ride quality. However, while the computer model predicts an overall ride severity index comparable to what was experimentally measured, it has limitations when predicting the overall PSD response. This can be attributed to: uncertainty in the road input, inadequate excitation of the bounce mode during road testing, and the presence of nonlinearities in the vehicle response.

As a general observation, the work presented here makes a strong argument about the need for operational testing. Without it, the only means of verifying a computer model would be the laboratory-based modal tests. The level of disparity shown here between responses due to low- and high-level excitations makes this approach to model validation clearly inadequate. Moreover, this observation should not be confined to the study of heavy truck dynamics. Automobiles, rockets, missiles, etc. are subject to the same degree of nonlinearity; this should be addressed in the modeling effort of these systems as well.

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