

# Calibration Of A Mathematical Vehicle Dynamic Model

*William J. Kenis*

Federal Highway Administration, USA

*Jamel Hammouda*

EBA Engineering, Inc., USA

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## ABSTRACT

This paper describes assumptions, methodologies and observations attained in a preliminary study to calibrate the VSIM2d heavy vehicle simulation model. The model was developed specifically to study the complete spectrum of heavy vehicle wheel forces applied to the pavement for use in policy decision making and for the life cost analyses of pavements. The study involved work in three main areas: road profiling, four-post road simulator (shaker table) testing with a straight 2-axle flat bed truck, and mathematical vehicle simulations. The use of the shaker table is an essential step in the validation and calibration of heavy vehicle models; since the wheels do not spin on the shaker table, and they do not spin in the pitch plane of the VSIM2d Model, then good comparisons might be expected. Calculated statistics of the time histories of tire forces measured in the shaker table tests were compared with those calculated using VSIM2d. Descriptive results are presented and qualitative assessments are made. These suggest the technical feasibility for further refinement and development of the calibration procedure presented and immediate use of the calibrated model to address practical policy and regulatory issues.

## INTRODUCTION

As a vehicle travels on the road surface, road roughness induces vibrations to the vehicle causing the tire loads to fluctuate about a static value. These dynamic tire forces result in an increase in pavement damage causing further deterioration of

the pavement surface and hence increased roughness. Even though there is general agreement that dynamic loading tends to cause an increase in pavement damage, there are a lot of uncertainties surrounding the significance of this phenomenon. Some reasons are given:

- Lack of an understanding of the spatial distribution of dynamic loads on the pavement are dynamic loads distributed randomly or if not to what degree are they repeatable
- Lack of data on pavement primary response and whole life pavement performance due to dynamic loading.
- Most findings provide only partial answers, adopt incompatible assumptions, and are based on partially validated or calibrated vehicle / pavement models, for example, use of the dynamic stress factor to represent pavement damage.

The Federal Highway Administration (FHWA) has established intensive research to determine more precisely the potential of dynamic forces to cause pavement damage. The work will also investigate means of increasing the road friendliness of heavy vehicles with consideration given to trade-offs between increased size and weight and pavement strengthening. The research is comprised of three main elements: field testing, mechanical simulation (DYNTRAC) and computer simulations using VSIM2d. Work, to date, has included (a) development of the FHWA VSIM2d

truck simulation computer program, (b) conduct of an elaborate series of tests involving both the FHWA shaker table and FHWA's 2-axle straight truck, (c) development of mechanistic flexible and rigid pavement damage prediction models and, (d) primary response tests at the FHWA Test Road using 2-, 5- and 6-axle trucks ( DIVINE tests used the NRC 5 axle tractor semi- trailer with air suspensions). Only those portions of (a) and (b) above which are needed to calibrate the VSIM2d model are discussed in this paper.

## ROAD WEAR CONCEPTS

Researchers have reported a broad range of estimates for increases in pavement damage attributable to dynamic loading [1]. In most cases these estimates were based on a modified version of the AASHO fourth power law which portrays road damage in terms of AASHO's 18 kip equivalent single axle load factor ( LEF ). It is directly proportional to the fourth power of the axle weight:

$$LEF = P_j^4 K$$

where  $K$  = inverse of a standard ( in this case 18 kip ) single axle load to the fourth power.

$P_j$  = any static axle load, in kips

One must keep in mind that this estimate was obtained from repeated dynamic loadings of real trucks on a circular test track and that the LEF represents damage in terms of user perception. Although the LEF accounts for dynamic loading, it does not account for mixed traffic since similar trucks with similar steel axle suspensions were repeatedly driven at the same speed on the circular track over a two year period until the road was termed failed (user perception). Thus one might say that the dynamic loadings were distributed in a more concentrated fashion than in a Gaussian fashion.

Nevertheless, Eisenmann [2] derived a quantity known as the road stress factor ( $\Phi$ ) to take into account dynamic loadings. Assuming that dynamic wheel forces are Gaussian he showed that the expected value of the fourth power of the instantaneous wheel force  $p$  (a random variable) is given by:

$$E[p]^4 = \nu P_{stat}^4$$

where:  $\nu = 1 + 6s^2 + 3s^4$

$P_{stat}$  = average tire force

$s$  = coeff. of variation of dynamic tire force called DLC

Multiplying by the constant  $K$  yields

$$\Phi = K \nu P_{stat}^4$$

It is intuitive therefore that

$$\Phi = \nu LEF$$

is another version of LEF always greater than the AASHO LEF depending on the value of DLC. For typical highway conditions of unevenness and speed, DLC's between 0.06 and .28 have been reported [3] depending on suspension. In general DLC varies proportionally with road roughness, vehicle speed and the resonant frequency and damping of the suspension system. It however, varies inversely with the load being carried by the suspension. Air spring suspensions normally have lower resonant frequencies than do steel springs and thus lower DLC's.

Since the AASHO LEF is a measure of "road damage", DLC's between .15 and .35 would then cause road damage 14 to 74% above that estimated by AASHO. Although .35 is relatively high for drive tandem axles, it may not be so for semi trailer tandem axles [1, page 124]. This implies that a road friendly suspension with DLC of .15 could carry an additional 1 1/2 to 2 t above that of a steel suspension with DLC of 0.35 in order to cause the same amount of damage.

Based on calculations using the above assumptions, it was reported [4] that widespread replacement of steel or rubber suspensions with air suspensions in England would reduce pavement damage by 8 - 20 %.

Hahn's research [5] enabled introduction of legislation in 1984 to allow two axle busses with rear air axle suspensions and dual wheels to carry 11 t rather than 10 t provided the sprung mass natural frequency measured in a step test is less than

1.5 Hz and the damping ratio greater than 0.25.

The European Union has ruled that for all vehicles produced after Jan 1993 with 11.5 t max. weight be fitted with dual tires and air suspension or equivalent, otherwise the max. weight is 10.5 t. Vehicles produced up to Jan 1993 were exempt.

Cebon [6] suggested that road damage is governed by damage at the worst locations rather than by an average value over the total road surface, and that this worst location exists because the largest ( peak ) dynamic loads always occur at the same locations ( spacial repeatability ). Values of increases in fatigue damage between 200 and 400 % ( not including walking beam ) are given.

Actual measurements of spatial repeatability [7] on medium and rough road sections, of TRL's test track facility, imply increases in road damage of 41% and 75% respectively.

Further research dealing with actual measurements of spatial repeatability is being conducted on real roads in The UK, France, and Canada.

## ROAD SIMULATOR TECHNOLOGY

Plans to construct and operate FHWA's DYNTRAC Shaker Table were initiated in the late 1970's. Procurement was initiated to construction and erection of the current facility at the Turner-Fairbank Highway Research Center (TFHRC) December 1985. Construction was completed in 1989. In 1988 FHWA held it's second in a series of Load Equivalency Workshops. The paper by Hu [8] and support from Workshop participants provided FHWA the impetus to continue its Truck Pavement Interaction research program in accordance with the plans presented by Kenis [9]. In his paper Hu Described the PACCAR Technical Center's road simulator as a 6 post servo - hydraulic system equipped with remote parameter control ( RPC ) software from MTS. At that time PACCAR would measure axle accelerations over real roads and then attempt to reproduce these same accelerations with the vehicle sitting on the shaker table. Here the actuator drive signal is designed to reproduce the accelerations but not the

road profile that caused them. The RPC software automatically selects the set of actuator amplitudes and frequencies that reproduce the accelerations measured in the field. They were not able at that time to measure wheel forces directly. Keep in mind that prior to this time shaker table work was largely geared to study suspension vibrations or to conduct vehicle component and endurance testing.

Other known shaker tables in the US include those at the Fruehauf Co and at Navistar. The Fruehauf facility consists is a six post servo hydraulic unit with RPC and it is equipped with load cells installed under the tires and it was primarily used for durability testing. The Navistar facility has similar attributes as Fruehauf except that the system contains ten servo hydraulic actuators.

A methodology for rating suspensions for road friendliness using a two post servo hydraulic rig was developed by de Pont. [10]. He used his own version of RPC and likewise attempted to reproduce axle accelerations as measured on real roads. Wheel forces were measured directly using load cells attached to the actuators.

Currently work is on-going at the National Research Council of Canada. The Canadian unit is a four post servo hydraulic unit with RPC and load cells attached to the actuators. They are currently conducting intensive work under the auspices of OECD's DIVINE research program.

The four-post road simulator at TFHRC is usually referred to as DYNTRAC ( **DY**Namic **T**ruck **A**ctuation ). The shaker table permits measurement of forces from different types of vehicles, suspensions, tires, etc. It is a 4-post system equipped with four 35-kips computer controlled servo hydraulic actuators to dynamically excite the wheels in a vertical plane (simulating the dynamic action of a truck traveling along a section of highway). During testing a vehicle rests on wheel pans (load scales) attached to the piston's of the hydraulic actuators. A road profile is input to the computer and the hydraulics are controlled so as to cause actuator piston displacements that are similar to those of the road profile. Frequency changes of the actuator piston's are intended to simulate vehicle speed. Kulakowski [11] has compared the power spectra density of a white

noise input signal (with a 50 hz frequency cutoff ) to that of the actuator's displacement and suggests that the good match to 20 Hz should allow the profile harmonics responsible for generating tire forces to be reproduced by the actuators with out significant error. This simply states that good agreement between actuator displacement and road profile displacement is attained as long as frequencies are kept below 20 Hz. and that therefore there is no need to install RPC software.

### THE VSIM2D COMPUTER PROGRAM

The VSIM2d computer program is a vehicle dynamic simulation model that was developed to reflect state of the art modelling capabilities. It is comprehensive enough to simulate all possible configurations of vehicles on the road and to allow for easy modification and user input subroutines to accommodate futuristic geometrics. Attributes of the VSIM2d simulation package include

- Minimum input and preprocessor generation of any additional parameters.
- Input of separate left and right profiles
- scaling factor for automatic change of units
- automatic suppression of roll
- switches for presentation of any state variable, its first or second derivative.
- Arbitrary large number of bodies
- Pitch, heave, and roll degrees of freedom
- Single, Tandem, and Tridem Suspension: Leaf spring, Air bag
- Tandem Suspension: Walking beam
- Rocker: Viscous Damping, Hysteresis
- Shock Absorbers: Non Linear
- Non Linear point or adaptive tire models

Complete descriptions of the model and results of sensitivity and verification studies are given elsewhere [12]. They prove that VSIM2d generates and solves correctly the equations of motions. i.e. every component in the model behaves as modelled and as desired.

### INSTRUMENTED TRUCK

#### VEHICLE INSTRUMENTATION

The FHWA test vehicle is a Navistar International straight flatbed 2-axle truck having constant rate steel leaf springs on the front and

progressive rate leaf springs on the rear (the rear steel springs may be replaced with air springs) [13]. Although it is not necessary to install instrumentation on the axle housing for measurement of wheel forces when some other more reliable method is available, the strain gages must be installed when wheel forces are to be measured on real roads. When using DYNTRAC, we need only the wheel scale placed between the tire and actuator for wheel force measurements; however, because the truck was to be used on real roads, axle strain gages were installed.

Strain gages are cemented to the axle housing to measure the shearing forces transmitted to the axles. Accelerometers are installed on the axle above the strain gages. By adding the force accelerating the mass outboard of the strain gages to the shear force sensed by the strain gages, the force applied at the tire/road interface can be determined. The main disadvantage of this method is that every vehicle requires a completely new installation and calibration. Figure 1 shows the DYNTRAC assembly and Figure 2 shows the scheme of the truck instrumentation .

In this study, the wheel forces obtained from the wheel scales placed under the tire and attached to the hydraulic actuators are used to calibrate VSIM2d.

#### INSTRUMENTATION EQUATIONS

The signals from the load scales and strain gages were converted from voltages to engineering units using calibration equations supplied by Penn State University.

Intercomp Scales: Several different types of scales were evaluated and the Intercomp scale, having adequate stiffness and negligible off axis bending, was selected. The tire rests directly on the scale which is bolted directly to the actuator. A detailed description is presented by Straight [13]. Static calibration of the scales determined a linear relationship between weight and voltage:

$$f_{scale} \text{ (lb)} = A * X \text{ (volts)} + B$$

where  $f_{scale}$  is scale weight in pounds, X is voltage reading, A and B are calibration constants.

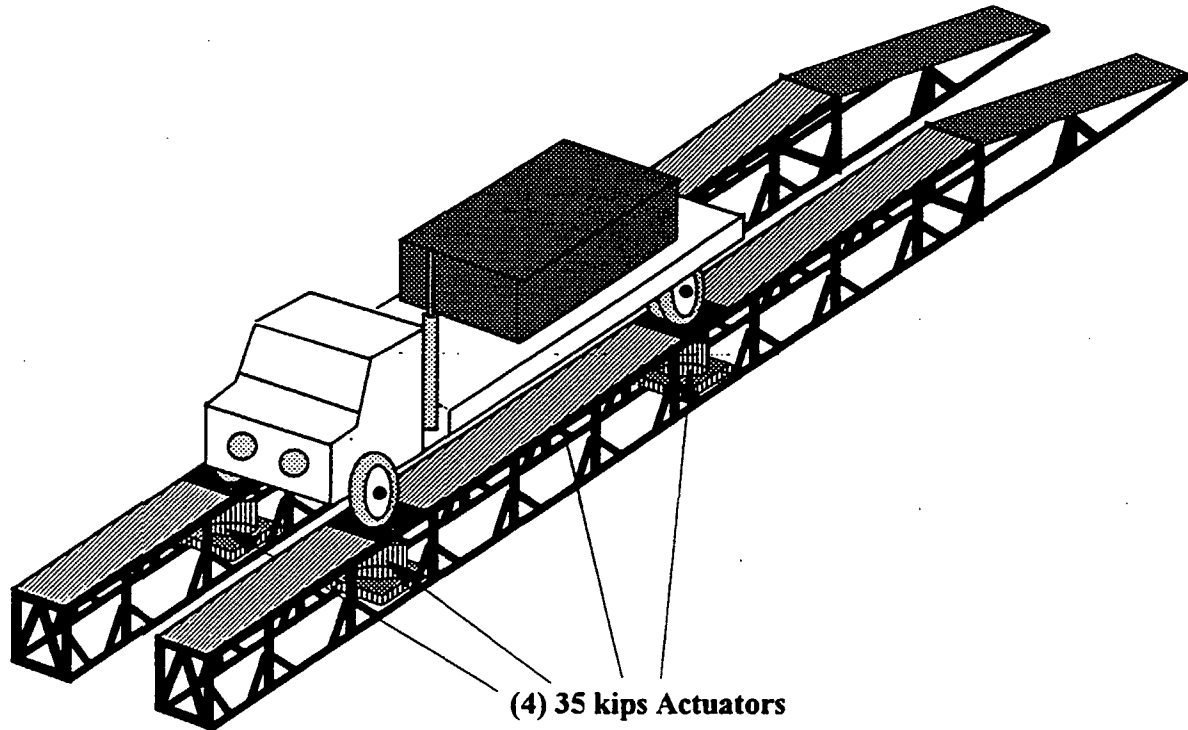
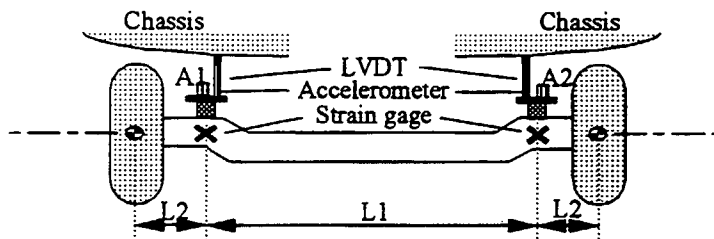
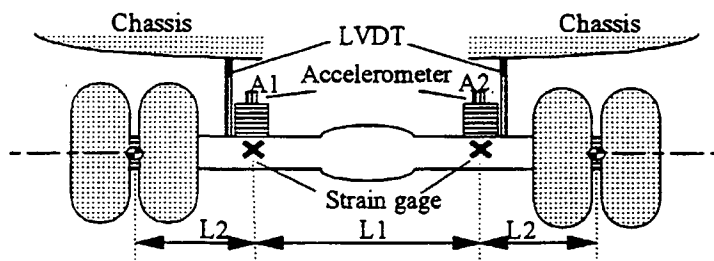


Fig 1. DYNTRAC



**Instrumentation of Front Axle**



**Instrumentation of Rear Axle**

Fig 2. Instrumentation of Test Truck

**Axle Strain Gages:** Axle strain gages are first calibrated statically. As weight is applied to the truck bed, strain gage readings in volts (V) are correlated to the scale readings in pounds yielding (  $f_{\text{gage stat}}$  ). It was found that a linear relation exist:

$$f_{\text{gage stat}} = K V + B$$

where K and B are constants.

The dynamic calibrations are iterative and were obtained by exciting the truck with in-phase harmonics at each tire. The total dynamic tire force (  $f_{\text{dyn}}$  ) is calculated from the strain gage component of the load (  $f_{\text{gage stat}}$  ), by adding an inertia term (  $f_i$  ) computed by multiplying the out board masses ( 175 and 300 for the front and rear axles respectively ) by their accelerations:

$$f_{\text{dyn}} = f_{\text{gage stat}} + f_i$$

The current value of  $f_{\text{dyn}}$  is compared to the load measured by the scale. If there is no match, then the K and B are changed iteratively until  $f_{\text{dyn}}$  closely matches the scale value. The calibration process was conducted each day the DYNTRAC testing was performed. These calibration equations were also provided by PSU. Leblanc's [ 14 ] calibration process is slightly different and involves iteration for the inertia component.

## MODEL CALIBRATION PROCEDURE

It is important that the initial calibration of VSIM2d be conducted against the simplest vehicle configuration possible. Using knowledge gained here will allow more complex calibrations with tandem and triple axles and with semi trailer combinations at a later time. The test matrix used to calibrate VSIM2d was selected in accordance with on-going shaker table testing in the laboratory:

- 1 two-axle straight truck with front and rear steel leaf springs
- front axle with single, rear axle with dual tires [ all tires radial 11R22.5 ]
- 1 tire pressure [ 95 psi ]
- 4 load conditions [ EMPTY, 4000, 8000, & 14,000 lbs. ]
- 4 speeds [ 15, 30, 45, & 55 mph ]
- 2 harmonic excitations

- 3 measured road profiles

The two harmonic excitations, 2.2 and 3.3 Hz ( in. amplitudes ), were used initially to see if measured wheel forces are in any way comparable to predicted values.

The 3 measured road profiles were used as input to DYNTRAC. The profiles were measured using FHWA's PRORUT2 non contact laser profiler. Three different road profiles representing low, medium, and high roughness roads were obtained from the three roads respectively: RT. 15 N, Clara Barton Park, and RT. 659. All roads are in the immediate vicinity of McLean, VA near TFHRC.

TABLE 1 identifies all of the vehicle related parameters which are required as input to VSIM2d. Most of the values listed are based on measurements made at the Bus Test Center in Altoona, Pennsylvania as reported in interim reports by Pennsylvania State University (PSU). The parameter names displayed in unbolded print in TABLE 1 are considered as precise measurements and fixed in the analysis. Obviously, the pitch (and roll ) moment of inertia must be changed in order to accommodate change in payload. The names displayed in bolded print in TABLE 1 were varied in the iterative calibrations.

Wheel force - time histories using VSIM2d were then calculated for all EMPTY truck test conditions of the matrix. Both calculated and measured wheel force - time histories were used to calculate the dynamic load coefficient ( DLC ), and power spectral density ( PSD ) function. Comparisons between the measured and calculated statistics were made.

During the conduct of the calibration process, little could be learned from comparisons between measured and calculated DLC's. Although the statistic is simple to interpret it was cumbersome and somewhat meaningless for the task at hand. Using the PSD proved to be quite convenient, easy to interpret and yielded sufficiently good calibrations. The comparison of measured and calculated wheel force PSD spectra indicated clearly at what

TABLE 1. INITIAL VEHICLE PARAMETERS \*

PARAMETER NAME	VALUE
wheel base	254 in.
front/rear suspension curb weight	7420/8460 lb.
roll/pitch moment of inertia sprung mass	40,000/105,000 in-lb.-sec <sup>2</sup>
<b>front suspension spring rate</b>	1530 lb./in./side/axle
<b>front coulomb friction</b>	800 lb./side/axle
front/rear lateral dist. between suspension springs	35/40.5 in.
front/rear track width	80.5/72.0 in
front/rear unsprung weight	1475/1975 lb.
<b>decay constant BETA front/rear suspension</b>	Not measured
<b>rear suspension spring rate</b>	3759 lb./in./side/axle
<b>rear coulomb friction</b>	1375 lb./side/axle
tire load radius	19 in.
<b>tire damping</b>	50 lb./ft./sec.
<b>tire stiffness</b>	4557 lb./in.
dual tire separation	13.5 in.
distance from vehicle sprung mass center to rear axle	121.483 in.

\* Bolded parameters were varied in this analysis.

frequencies discrepancies might exist.

Those parameters ( bolded names in TABLE 1 ) which have a significant effect on the spectrum where the discrepancies exist were altered as input to VSIM2d and a new PSD spectrum calculated. For instance, if the discrepancy is at the tire hop mode, it was the tire parameters that were changed to produce the desired spectrum; for body modes it was the suspension characteristics that were tuned. The process was continued until reasonable agreement was reached for the empty truck. The iterative parameters with their initial and final values are listed in TABLE 2. The values in TABLE 2 were then input to VSIM2d along with the actuator's output drive signal to predict the tire loads for the various test conditions given above.

## COMPARISONS OF VSIM2D AND DYNTRAC

Over several hundred computer plots were generated in the conduct of the calibration process. Included herein are 11 typical plots presented in figures 3 through 13. All of the curves (in figure 4 through 13) generated by VSIM2d, used calibrated final parameter values except for the VSIM2d curve in Figure 6. Note that in all figures, left front, right front, left rear and right rear are denoted as LF, RF, LR and RR, respectively. In addition, normalized tire load is defined as:

$$\frac{TF_{dyn} - \overline{TF}}{\overline{TF}}$$

where  $TF_{dyn}$  is dynamic tire force and  $\overline{TF}$  is mean tire force.

Figure 3 depicts the performance of the left

TABLE 2. ITERATIVE VEHICLE PARAMETERS \*

PARAMETER NAME	VALUE
front suspension spring rate	1530 ( <b>1280</b> ) lb./in./side/axle
front coulomb friction	800 ( <b>250</b> ) lb./side/axle
decay constant BETA front/rear suspension	(.004/.001)
rear suspension spring rate	3759 ( <b>2925.7</b> ) lb./in./side/axle
rear coulomb friction	1375 ( <b>500</b> ) lb./side/axle
tire damping	( <b>50</b> ) lb./ft./sec.
tire stiffness	4557 ( <b>5037</b> ) lb./in.

\* Bolded values were final values used in this analysis.

front actuator and shows excellent agreement between the input profile and output drive signals.

Figures 4 and 5 are plots of VSIM2d for harmonic input. Here excellent agreement is obtained both in the time domain and in the frequency domain. Note also in figure 5 that the PSD of the normalized force measured by the gage is identical to that measured by the scale.

Figure 6 shows that when the initial measured vehicle parameters are used in VISM2d, extremely large discrepancies between measured and simulated tire load PSD's are introduced. When the final calibrated parameters are used in VSIM2d, reasonable agreement between measured and simulated tire load PSD's are obtained. Such typical agreement is shown in Figure 7. It is important to remember that the calibration as described earlier does not lead to the optimal values of the parameter rather it was performed to lead to a general qualitative agreement to assess the feasibility of calibration of the model.

Figures 4 and 5 compare measured with simulated tire loads for harmonic excitation. For this type of excitation the response tends to be much simpler because it is dominated by the mode close to the frequency of excitation. Consequently a close match was obtained between measured and simulated tire loads. Figure 4 shows the normalized tire load in the time domain for the front left tire at 3.33 hz. and figure 5 is the PSD of the normalized tire load of the left rear tire at

frequency of 2.22 hz. The excellent agreement observed leads us to believe that a single mode of vibration will easily produce these excellent comparisons.

Figure 8 compares time traces of the normalized left front tire load of the scale with those of the gage for a payload of 4000 lb, at 15 mph on low roughness road. Figure 9 compares scale with simulation output for the same testing condition. It can be seen from these typical plots that the simulation is capable of reproducing the general behavior of the load both in frequency and amplitude.

Figure 10 shows typical comparisons of scale, gage and simulated normalized tire load PSD. Figure 11 shows the time traces of the normalized right front tire load for the scale and simulation for the empty truck at 15 mph on medium roughness road. These show that the simulation accurately predicts the modes of vibration both in frequency and amplitude.

Figure 12 displays the PSD of the normalized rear tire load at 8000 lb payload at 30 mph on a high roughness road. It shows that at high roughness and a moderate speed, the simulation agreed reasonably in predicting the lower modes (body modes) but overestimated the wheel hop.

Figure 13 clearly displays that the model is still capable of predicting the frequency-amplitude content of the tire load at moderately high speed - 45 mph, on a medium roughness road.



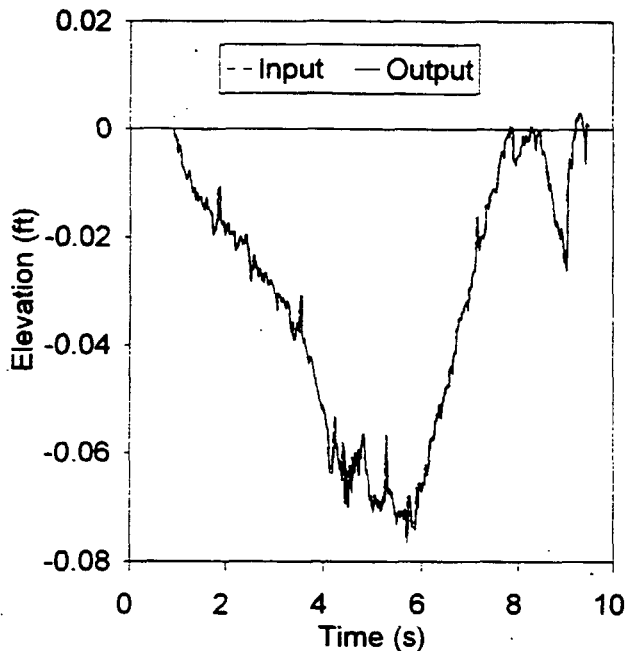


Fig 3. Actuator Performance  
LF, 15 MPH Rt. 15N (Empty)

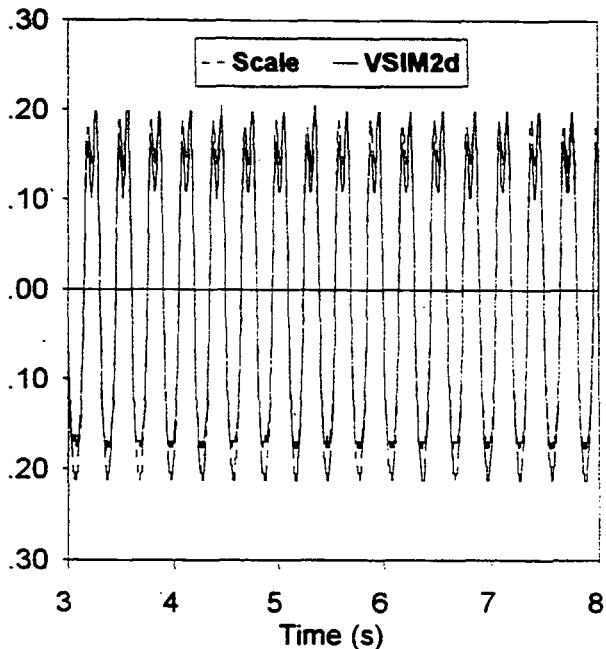


Fig 4. Normalized Dynamic Tire Load  
LF, Harmonic 3.3 Hz (Empty)

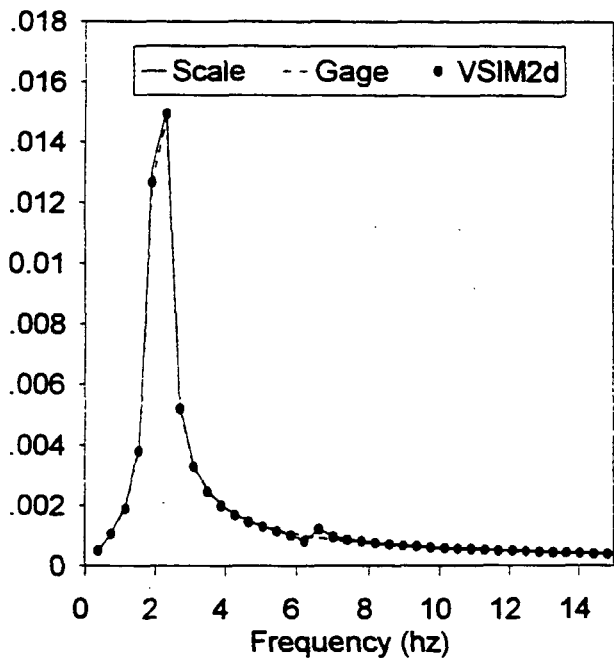


Fig 5 PSD Normalized Tire Load  
LR, Harmonic 2.2 Hz (Empty)

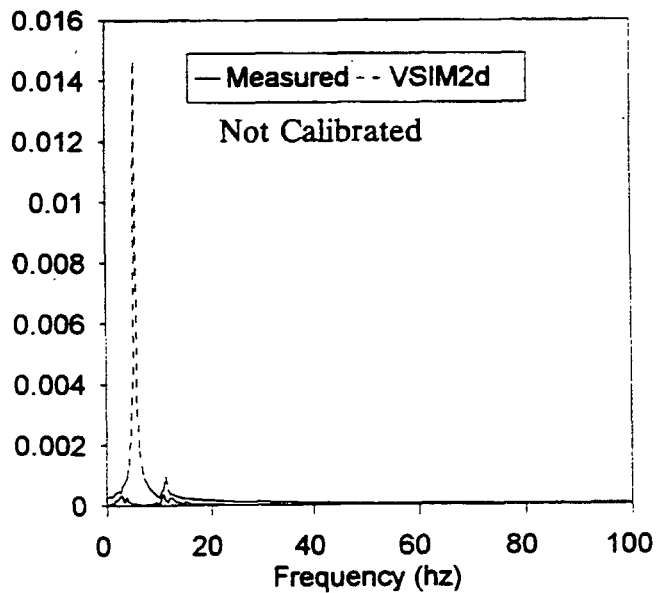


Fig 6. PSD of Normalized Tire Load  
RF, 45 MPH Rt. 15N (Empty)

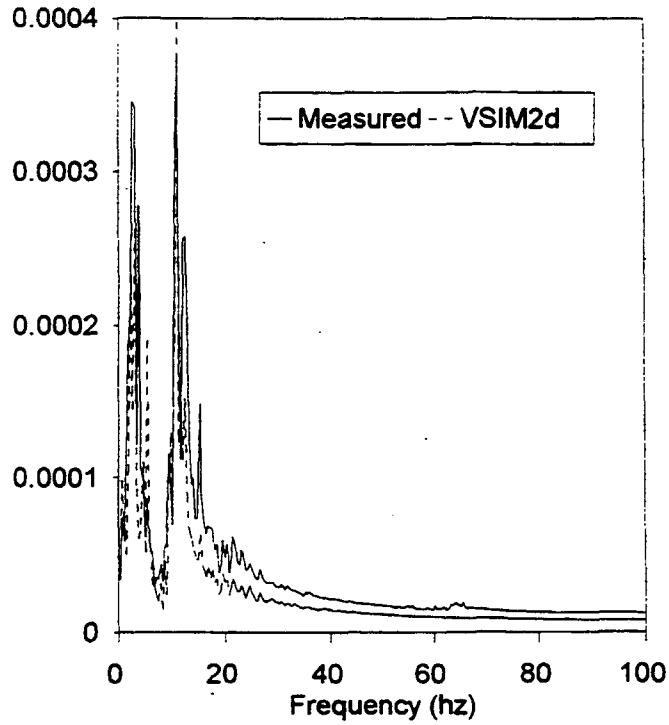


Fig 7. PSD of Normalized Tire Load  
RF 45 MPH, Rt. 15N (Empty)

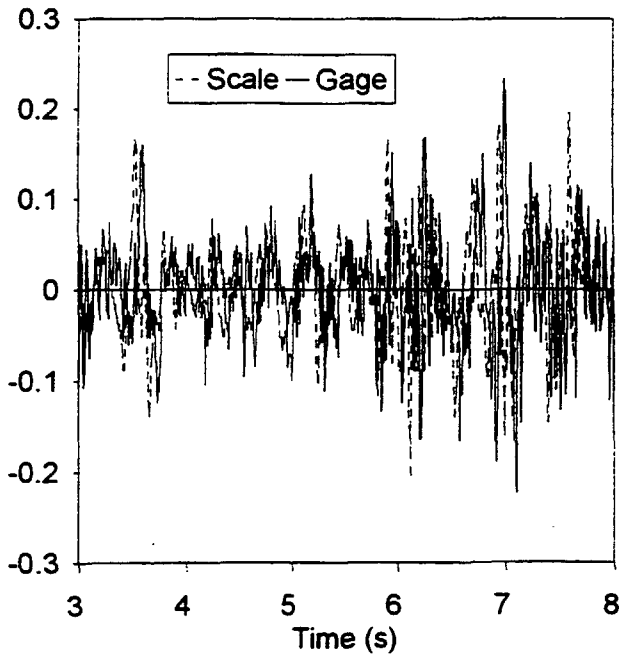


Fig 8. Normalized Tire Load  
LF, 15 MPH Rt. 15N (4000 lb payload)  
Scale vs. Gage

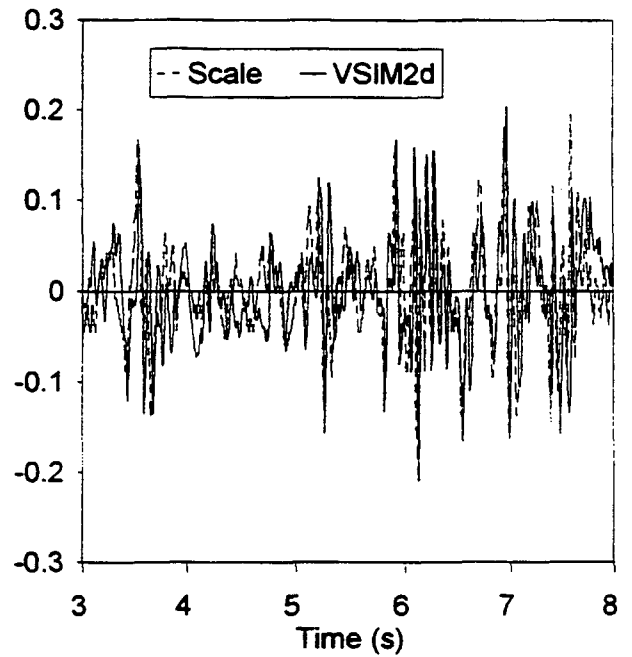


Fig 9. Normalized Tire Load  
LF, 15 MPH Rt. 15N (4000 lb payload)  
Scale vs VSIM2d

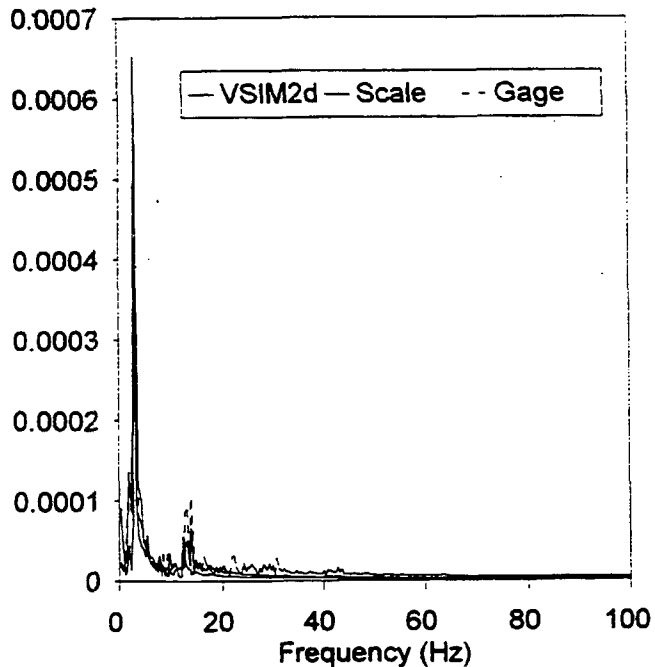


Fig 10. PSD of Normalized Tire Load RF, 15 MPH Clara Barton (Empty)

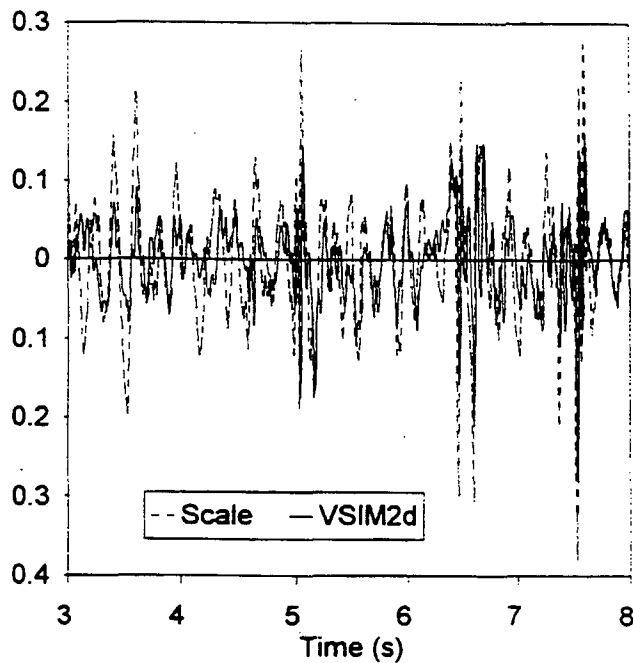


Fig 11. Normalized Tire Load RF, 15 MPH Clara Barton (Empty)

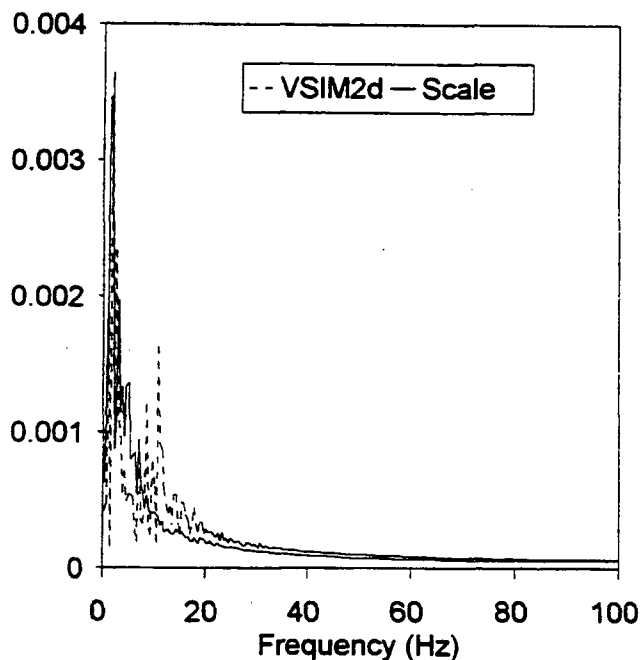


Fig 12 PSD of Normalized Tire Load LR, 30 MPH, Rt 659 (8000 lb payload)

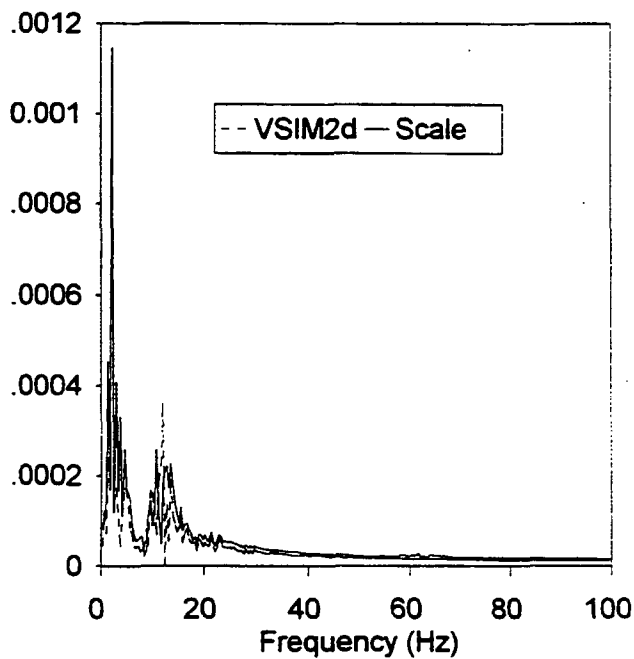


Fig 13. PSD of Normalized Tire Load LF, 45 MPH Clara Barton (4000 lb payload)

TABLE 2. ITERATIVE VEHICLE PARAMETERS \*

PARAMETER NAME	VALUE
front suspension spring rate	1530 ( <b>1280</b> ) lb./in./side/axle
front coulomb friction	800 ( <b>250</b> ) lb./side/axle
decay constant BETA front/rear suspension	( <b>.004/.001</b> )
rear suspension spring rate	3759 ( <b>2925.7</b> ) lb./in./side/axle
rear coulomb friction	1375 ( <b>500</b> ) lb./side/axle
tire damping	( <b>50</b> ) lb./ft./sec.
tire stiffness	4557 ( <b>5037</b> ) lb./in.

\* Bolded values were final values used in this analysis.

### COMMENTS AND RECOMMENDATIONS

The VSIM2d model is capable of predicting the general trends of the tire force PSD representing the different modes of vibrations. The comparison should be viewed qualitatively and it indicates that reasonable comparison between measurement and simulation could be achieved for most of the test conditions. The comparison tended to deteriorate for higher roughnesses and higher speeds.

Wherever there is reasonable agreement of the PSD one might also expect reasonable agreement of the time traces of the tire loads. Similarly for the DLC.

The DLC is inappropriate for model calibration purposes. The PSD function consists of distinct peaks corresponding to the resonant modes, and it is much easier to detect which parameter needs to be altered for a given mode.

Vehicles normally vibrate in 3 major modes as frequency increases (bounce, pitch, and wheel hop). Conduct of shaker table harmonic sweep tests to identify these different resonant frequencies will enable quick and accurate backcalculation of the different vehicle parameters.

### CONCLUSIONS

This work accesses the possibility of developing methodologies for calibrating a nonlinear dynamic truck mathematical model using real road profile shaker table data. Excellent agreement was reached for harmonic excitation; reasonable agreement was reached for the real road profiles. The final vehicle parameters attained are reasonable estimates of what the parameters should be. The trial and error method used can not be considered to yield optimal solutions but it does yield a set of parameters that reasonably represent the real behavior of the vehicle. The results are very encouraging, they prove that a frequency sweep test scheme for calibration purposes together with an appropriate optimization scheme will allow the calibration process to be performed quickly and efficiently. Immediate use of the model for more complex vehicle configurations is warranted however, only if one exercises "educated" extensions of the calibrated parameters attained herein.

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