Modal Impact Testing of Ground Vehicle Enabling Mechanical Condition Assessment

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ABSTRACT

Military tactical wheeled vehicles must operate on wide ranging terrains under varying payload driving conditions, which lead to high dynamic loads in the wheels and suspension. High loads can, in turn, lead to mechanical and structural failures. The ability to quickly detect damage in a vehicle in the field can be extremely advantageous for condition-based maintenance. In this paper, the use of modal impact testing to characterize the modal properties of an H1 chassis in its healthy and faulty conditions was studied. The results show that five modes of vibration – bounce, pitch, roll and two flexible body modes of the frame – can be identified using the Complex Mode Indicator Function. Frequency response functions are measured by applying modal impacts to the chassis and measuring the response using a roving set of accelerometers on the wheel, body, and chassis. The frame of the vehicle was impacted at three locations and response accelerations were measured at 19 points on the chassis, vehicle body, suspension and wheels. The sprung mass vibrations in roll, bounce, pitch, beaming and torsional shake were estimated to have modal frequencies of 2.0 Hz, 3.5 Hz, 4.1 Hz, 9.8 Hz and 13 Hz, respectively. The experiments also indicated that modal impact testing has the ability to detect faults in the vehicle, such as low tire pressure. The results suggest that modal impact testing on ground vehicles is a feasible means of identifying the operational deflection shapes and certain types of mechanical damage. Furthermore, this knowledge could be used by onboard structural health monitoring systems to effectively analyze operational data to identify damage in operation.

INTRODUCTION

The ability to accurately characterize the free dynamic response characteristics of a vehicle is invaluable in the development of a ground vehicle or the modification of a vehicle suspension system. Modal hammer impact testing is often thought to be limited in its ability to provide accurate results for large, heavily damped structures, such as an H1 military vehicle. However, this study shows that good estimates of the modal properties can be gleaned from impact testing of a large vehicle in a relatively quick and inexpensive fashion. Modal impact testing was conducted on a military, Hummer H1 vehicle. Faults were induced in the Hummer H1 to determine if the FRFs would suggest that there was some sort of fault within the vehicle. These tests indicated that faults could be observed by analyzing the discrepancies in the faulted and un-faulted vehicle frequency response functions. To obtain the baseline frequency response, modal impact testing was employed using an impact sledge hammer at three specific points on the H1. This work will be useful long term in the creation of a vehicle health monitoring system, which has the potential to reduce the maintenance costs of the vehicle and increase asset readiness.

Operating Deflection Shapes

In the article by Schwarz and Richardson on operating deflection shapes [1], an operating deflection shape (ODS) is defined as any forced motion of two or more points on a structure. In other words, the specification of the motion of two or more points defines a shape; more generally, a shape is the motion of one point relative to all other points. Operating deflection shapes are measured to identify the modes of vibration, which superimpose to produce the operating shapes when a structure is driven by various forcing functions. Mode shapes are inherent properties of a system, whereas ODSs depend on the forces or loads applied to a system. The ODSs are measured in the time domain using a multi-channel data acquisition system to simultaneously acquire the responses at certain degrees of freedom, including reference (fixed) responses across all of the measurements.

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Complex Mode Indicator Function

The Complex Mode Indicator Function (CMIF) is a simple algorithm based on singular value decomposition (SVD) methods applied to multiple reference frequency response measurements [2]. CMIF properly identifies the existence of real normal or complex modes and the relative magnitude of each mode, particularly when there are closely spaced or repeated modal frequencies by maximizing the use of spatial data. The SVD approach does not require multiplication of the FRF matrix by the Hermitian as in $[H(\omega)]\bullet[H(\omega)]^H$. By taking the SVD of the FRF matrix at each frequency, the following expression is obtained [2]:

$$[H(\omega)] = [U(\omega)]\left[\sum(\omega)\right][V(\omega)]^H \tag{1}$$

where:

$[H(\omega)]$ is the FRF matrix of size N_o (number of response points) by N_i (number of excitation points)

$[U(\omega)]$ is the left singular value matrix (unitary)

$\left[\sum(\omega)\right]$ is the singular value matrix (diagonal)

$[V(\omega)]$ is the right singular vector matrix (diagonal)

Vehicle Vibrations

Common low frequency vibrations of the sprung mass of a vehicle as a rigid body consist of pitch, bounce, and roll. Pitch is the angular component of vibration of the sprung mass, or vehicle body, about the vehicle y-axis [3]. Bounce is the translational component of ride vibrations of the sprung mass in the direction of the vehicle z-axis, while roll is the angular component of ride vibrations of the sprung mass about the vehicle x-axis. Vibrations of the sprung mass as a flexible body include torsional shake and beaming. Beaming is a mode of vibration involving predominantly bending deformations of the sprung mass about the y-axis. Torsional shake is a mode of vibration involving twisting deformations of the sprung mass about the x-axis.

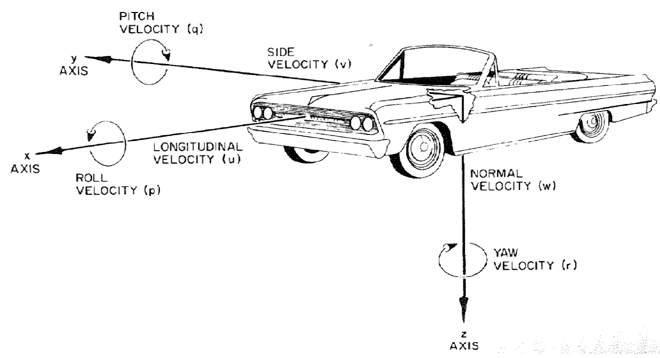

Fig. 1 Local pitch, bounce, and roll angles [3]

THEORY

The quarter-car model provides an excellent basis to analyze and estimate the body bounce and wheel hop; it consists of a quarter of the vehicle body, m_s (sprung mass) and one wheel of the vehicle, m_u (unsprung mass), where each mass is permitted to undergo vertical motion only. The spring stiffness k_s and shock absorber with damping coefficient c_s represent the stiffness and damping of the independent suspension system. The unsprung mass, m_t, k_t and c_t represent the stiffness and damping of the tire. The theoretical model is different from typical quarter car models due to a unique experimental setup. The governing equations of motion and transfer functions when modal-impacting a quarter-car are:

$$m_s\ddot{x}_s + c_s\dot{x}_s + k_s x_s - c_s\dot{x}_u - k_s x_u = f_{hammer} \tag{2}$$

$$m_u\ddot{x}_u + (c_s + c_t)\dot{x}_t + (k_s + k_t)x_u - c_s\dot{x}_s - k_s x_s = 0 \tag{3}$$

$$\frac{X_s}{F_{hammer}} = \frac{m_u s^2 + (c_s + c_t)s + k_s + k_t}{m_u m_s s^4 + m_s(c_s + c_t)s^3 + m_u c_s s^3 + m_s(k_s + k_t)s^2 + m_u k_s s^2 + c_s c_t s^2 + (c_t k_s + c_s k_t)s + k_s k_t} \tag{4}$$

$$\frac{X_u}{F_{hammer}} = \frac{c_s s + k_s}{m_u m_s s^4 + m_s(c_s + c_t)s^3 + m_u c_s s^3 + m_s(k_s + k_t)s^2 + m_u k_s s^2 + c_s c_t s^2 + (c_t k_s + c_s k_t)s + k_s k_t} \tag{5}$$

For this set of equations, the road input z_{in} is zero, since the external excitation only occurs on the body of the HMMWV. However, it should be noted that this model does not accurately represent the real operating conditions of the vehicle.

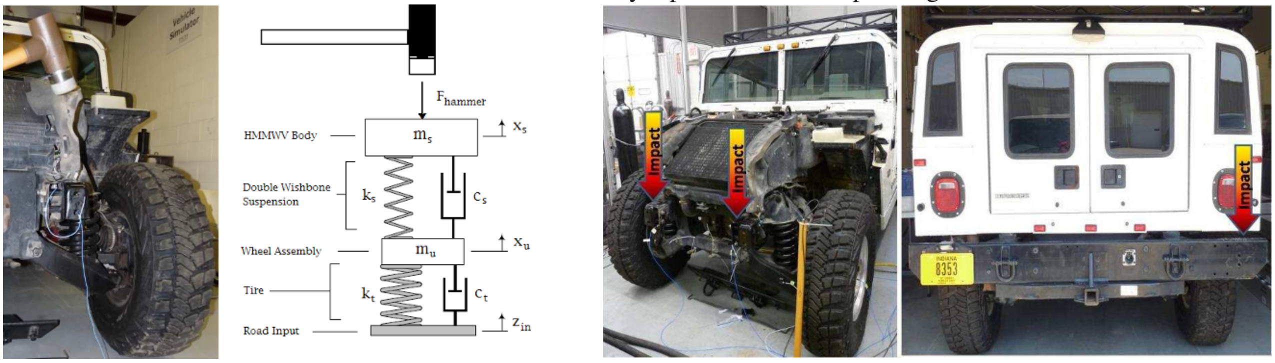

Fig. 2 Theoretical quarter-car model of vehicle and impact locations

Since it was deemed too difficult to excite the wheels directly using the equipment on hand, it was decided to use a modal sledge hammer to excite the vehicle frame instead. Methods using the road simulator to apply initial conditions to the vehicle wheels are also being explored for possible use in impulse testing. The wheels were kept in contact with the floor. Assuming reciprocity, three impact points were identified on the vehicle frame: left and right front frame rails and the rear bumper. The hood of the vehicle was removed to expose the front frame rails so they could be impacted with the hammer in Fig. 2.

EXPERIMENTAL APPLICATION

Six DC accelerometers were placed at various locations on the vehicle because the expected vehicle vibration modes were low in frequency (< 20Hz). Super glue was used to attach the sensors. The vehicle frame was then impacted at each of the three locations described above and the responses were recorded. The ODS measurement method was applied and the accelerometers were moved to different positions on the vehicle and the impacts were repeated to obtain more spatial data. Overall, three sets of data were collected. There was also one unique point from the fault testing portion of the experiment, yielding a total of 19 spatial response points. The locations of the accelerometers in each set are listed in Table 1 and Fig. 3.

TABLE 1 Sensor locations and descriptions for each measurement set

Location Number		Description	Coordinates		
			X	Y	Z
Data Set 1	1	Right front frame	0	-27.25	0
	2	Vehicle body (next to left front body isolation mount)	-50	0	0
	3	Left front lower control arm (near shock connection pt)	-9.5	7	-10.5
	4	Left rear - back bumper	-180	0	0
	5	Left front wheel spindle	-16	25	-12
	6	Left front frame	0	0	0
Data Set 2	7	Left front frame near shock tower	-14.5	0	0
	8	Right front frame near shock tower	-14.5	-27.25	0
	9	Bottom of LH side frame rail - mid vehicle	-72.75	0	-6
	10	Right rear frame near shock tower	-154	-27.25	0
	11	Right front wheel spindle	-16	-50	-12
	12	Left rear frame near shock tower	-154	0	0
Data Set 3	13	Left front upper control arm	-20	11	3
	14	Bottom Frame middle cross member	-88	-17	-5
	15	Right rear lower control arm	-159	-40	-11
	16	Right rear body - next to body isolation mount	-176	-39	2
	17	Right rear wheel spindle	-149	-56	-12
	18	Right rear frame	-102	-27.25	-6
Fault Analysis	19	Left front shock tower (outside top of spring)	-12	9	5

- The origin for all coordinates listed above is at response point 6, which is the left front frame rail (Impact Point 1)
- Positive x points towards forward vehicle driving direction, y points towards driver side, z points up toward ceiling
- Right hand coordinate system and all measurements in INCHES

Fig. 3 Sensor locations for each measurement set

To excite the vehicle suspension, the modal sledge hammer was used to impact the frame. After experimenting with different hammer tips, a soft rubber tip was used to obtain the most uniform low frequency energy input [4]. A force threshold of 5 lbs was also applied to the input force. This threshold caused the data acquisition system to ignore any forces that are less than 5 lbs resulting in a smoothed input power spectrum, as shown in Fig. 4. The figure also shows the desired shape of a typical hammer impact. The roll off in the frequency domain should be smooth and consistent to indicate a clean impact. An effort was also made to impact with a consistent amplitude. The data acquisition software was programmed to screen for double hits and to only accept impacts that fell within a specified range. All impacts fell between 1,850 and 2,150 lbs. A total of five impacts were averaged at each location. Since only the vertical motion of the vehicle was measured, care was taken to ensure that each hit was as "square" as possible to avoid exciting modes in other degrees of freedom (other than vertical).

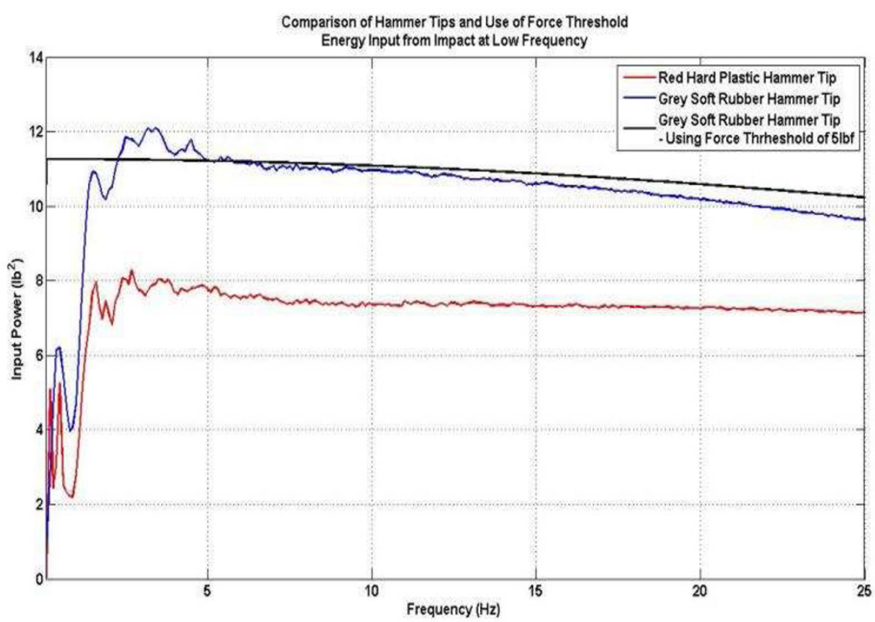

Fig. 4 Hammer tips and force threshold comparison

The CMIF was used to analyze the closely spaced and heavily damped modes of vibration of the vehicle as shown in Figure 5. The function also provided an estimate of the mode shapes contained in the left singular matrix in Eq.(1). The estimated modal vectors were obtained from the left singular matrix at each corresponding frequency and input (reference) location. The mode shapes were then animated using the modal vectors to more easily interpret the motion at each frequency. The modes listed in Table 2 were animated and still images of the deflection shapes were plotted in Figure 7 for the pitch mode at 4.125 Hz. Figure 6 is the un-deformed shape of the vehicle based on the response points collected for each measurement set.

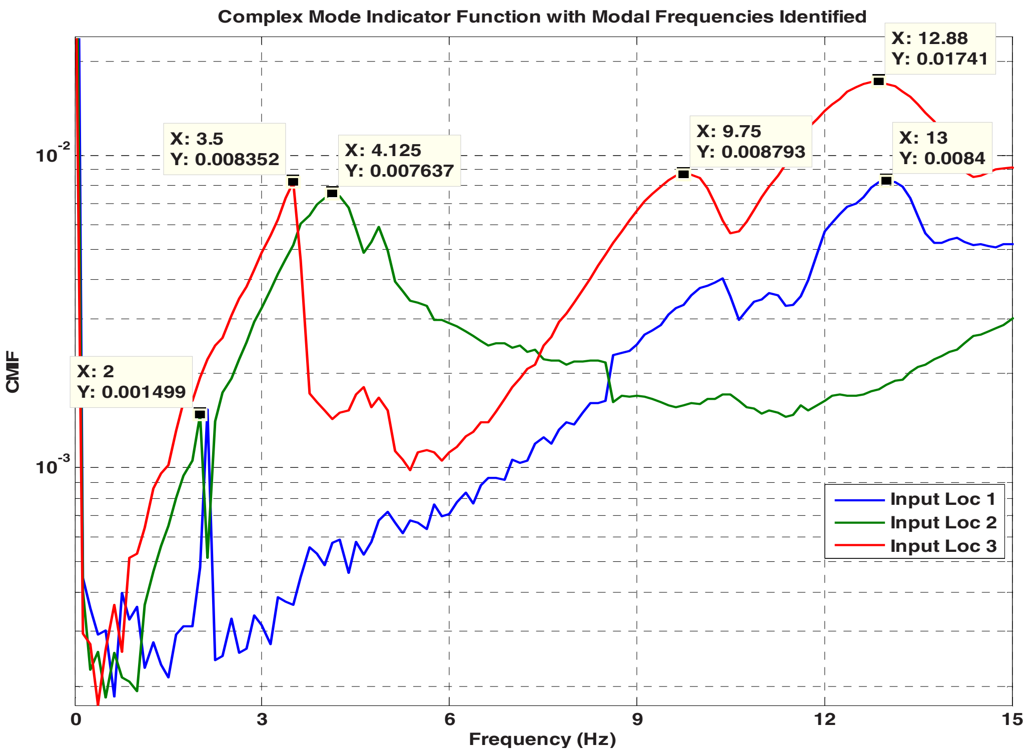

Fig. 5 CMIF plot for three reference points and identified modal frequencies

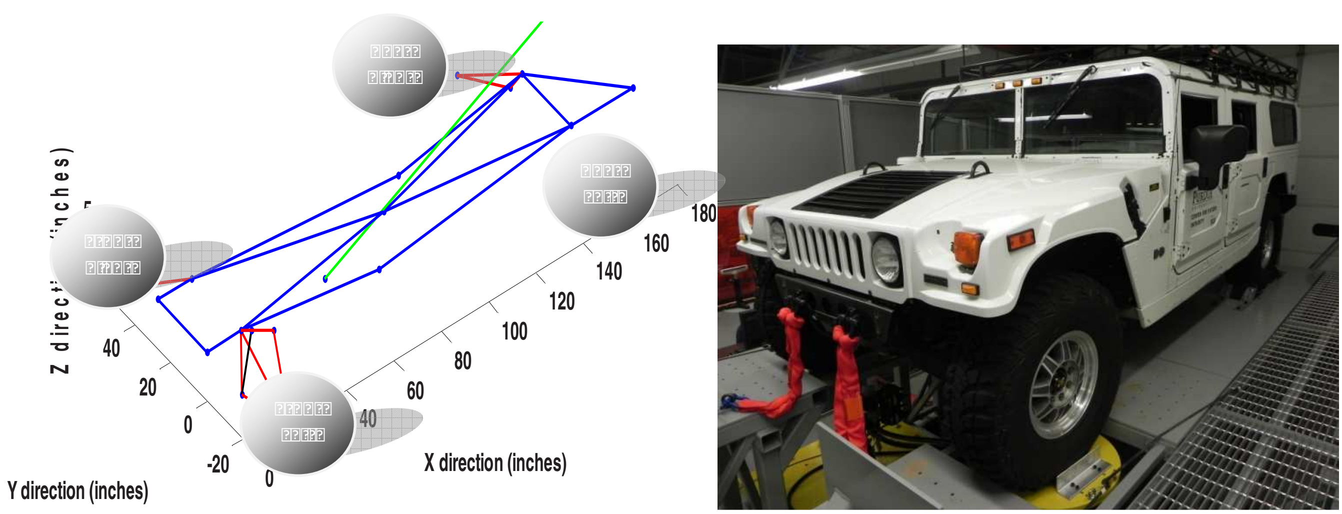

Fig. 6 Representation of un-deformed geometry of the vehicle

In Figure 6, the blue lines represent the vehicle frame or chassis. The red lines represent the suspension, the black line represents the left front shock absorber and the green line represents the vehicle body or cab structure.

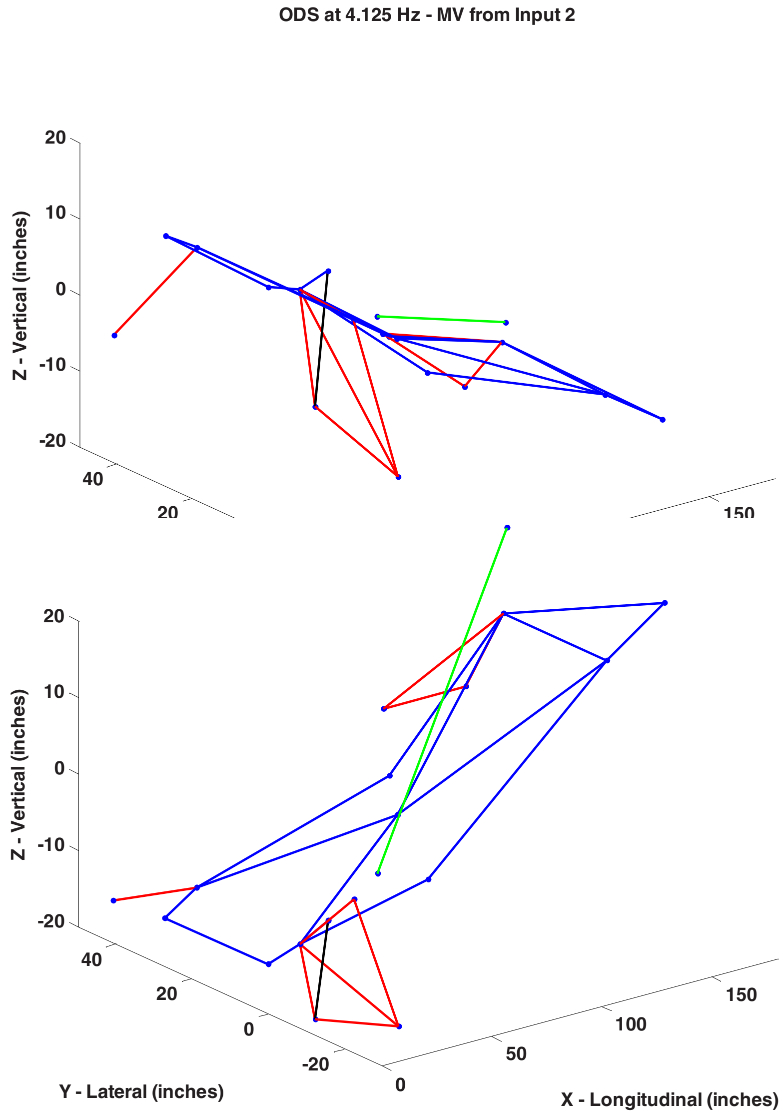

Fig. 7 Pitch mode at 4.125 Hz

Five modes of vibration of the vehicle sprung mass were identified in Table 2. These five modes determine the ride and shake characteristics of the vehicle as defined by SAE J670e [3].

TABLE 2 Identified vehicle modes of vibration

[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]

Due to the location of the modal impact excitation on the vehicle frame, the theoretical model was analyzed using typical vehicle parameters to determine whether or not differences existed for the impact location and road input location. For comparison, Figure 9 revealed that the second peak of the green line, which corresponded to the tire degree of freedom (wheel hop), was non-existent in the FRF of the modal impact location (X_{sprung}/F_{hammer}). These observations indicated that the wheel hop mode of vibration would also not be evident in the experimental data.

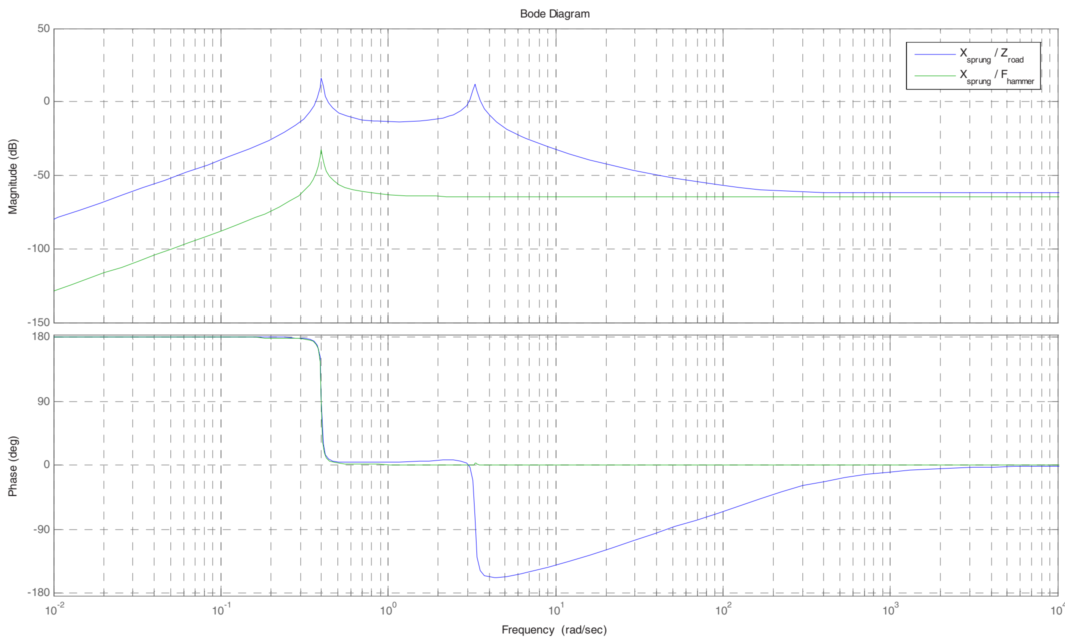

Fig. 8 Theoretical model comparing modal impact and road input locations

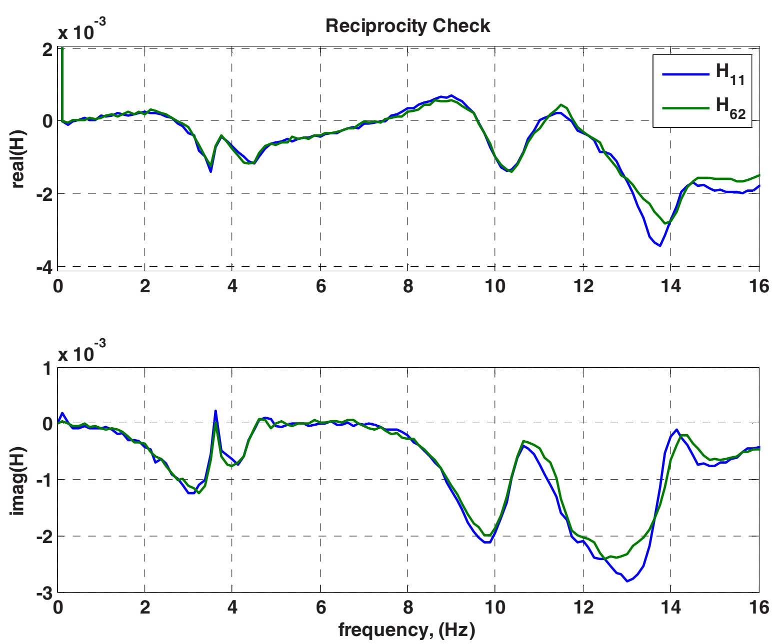

Fig. 9 Verification of reciprocity assumption for vehicle chassis

Figure 10 shows the FRF for the input at location 1 and response at input location 2 as well as the FRF for the input at location 2 and response at input location 1. The fact that these plots lay on top of one another verifies the assumption of reciprocity across the vehicle chassis.

FAULT IDENTIFICATION

In the second half of the experiment, a fault was induced on the vehicle and the change in its response to a frame excitation was measured. All the accelerometers were mounted at points around the front driver side wheel and suspension as shown in Table 3 and Figure 11. The modal hammer was used to impact the frame in the manner described previously. Again, all three impact points were used. However, in this case the tire pressure was reduced from 30 psi to 10 psi in both front tires. Since the two front tires are connected to a central air pressure system, it was not possible to reduce the pressure on one tire only. The rear tires were left unchanged. The vehicle was impacted again to study the effect of low tire pressure on the response.

Table 3 Fault analysis sensor locations

Location Number		Description	Coordinates		
			X	Y	Z
Fault Analysis	13	Left front upper control arm	-18	9.5	4
	7	Left front frame near shock tower	-14.5	0	0
	3	Left front lower control arm (near shock connection pt)	-9.5	7	-10.5
	6	Left front frame - Impact pt 1	0	0	0
	5	Left front Wheel Spindle	-16	25	-12
	19	Left front shock tower (outside top of spring)	-12	9	5

- The origin for all coordinates listed above is at response point 6, which is the left front frame rail (Impact Point 1)
- Positive x points towards forward vehicle driving direction
- Positive y points towards the driver side of vehicle
- Positive z points up toward the ceiling
- Right hand coordinate system and all measurements in INCHES

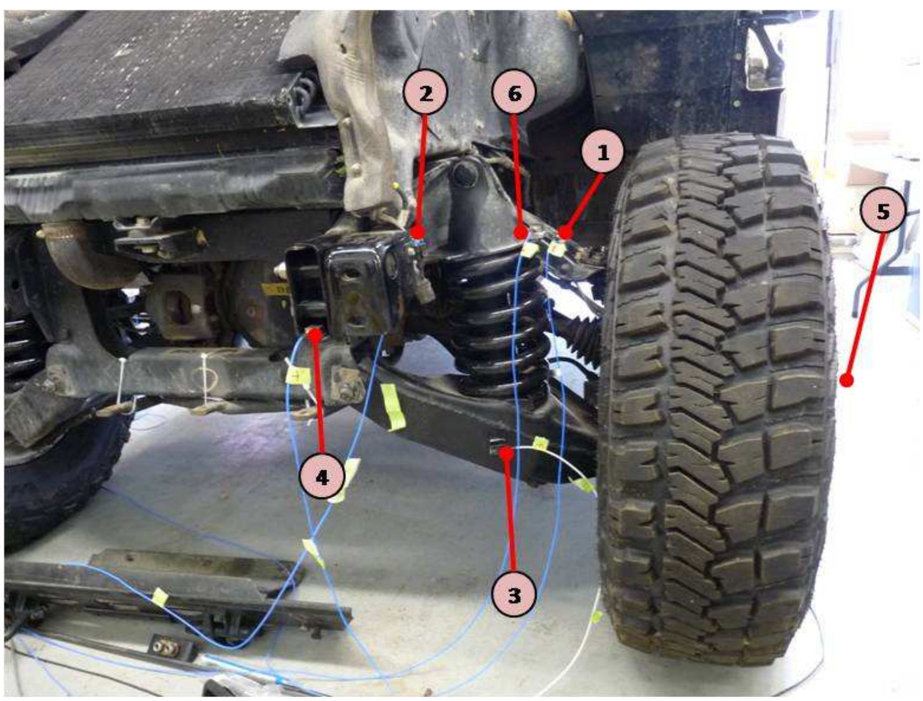

Fig. 10 Sensor locations for low tire pressure fault analysis

Figure 12 illustrates a comparison between estimates of modal frequencies with the front tires at the standard 30 psi pressure in contrast to the 10 psi faulted pressure. This plot indicates that several modes especially the vehicle bounce mode were sensitive to changes in tire pressure because the bounce mode shifted from 3.5 to 3.0 Hz.

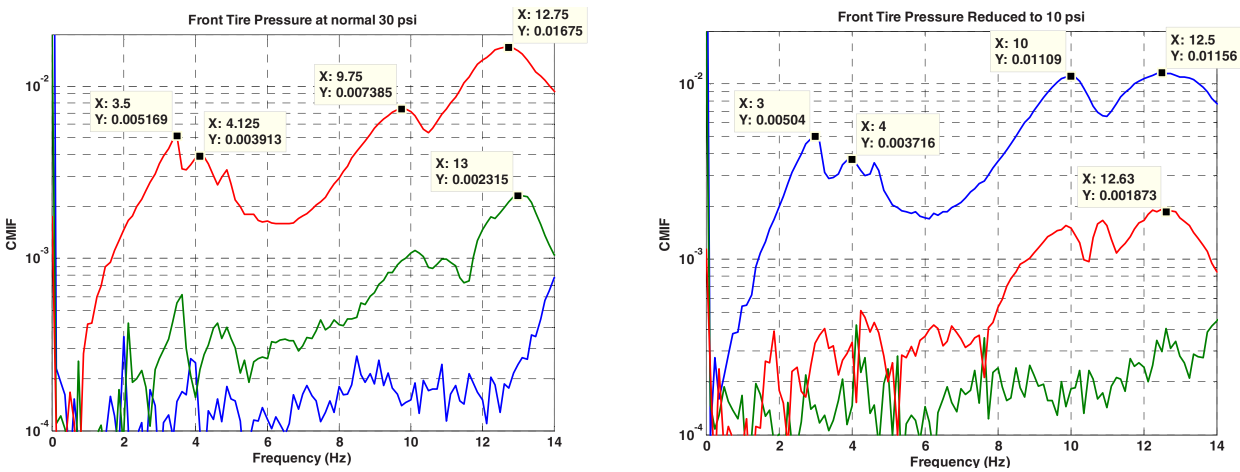

Fig. 11 Comparison of CMIF and modal frequencies with front wheels at 30 psi vs. 10 psi

CONCLUSION

One of the fundamental assumptions that was made at the beginning of the experiment was that inputs on the tire footprint would produce the same dynamic response characteristics as inputs on the frame. However, the experimental data and the model showed that this assumption does not hold across the suspension system. The fact that the wheel hop mode was not seen in the mode shapes indicates that the vehicle suspension may be isolating the un-sprung mass and preventing enough energy from reaching the tire to excite wheel hop.

This experiment showed that although the first choice would probably have been to use a shaker test, the modal impact method still produced meaningful results. The data collected revealed bounce, pitch, roll, and two flexible body modes of the frame. Modal impact testing can be used on a vehicle to quickly investigate the modal properties of the suspension and frame.

The data indicates that modal testing may also have the ability to detect faults in the vehicle suspension. When the front tire pressure was reduced, the frequency of the bounce mode dropped from 3.5Hz to 3Hz. This result is logical because a reduction in the tire pressure should reduce the overall stiffness of the suspension. This reduction in stiffness, in turn, would lower the natural frequency as was observed experimentally in this study.

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