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Techniques for Evaluating Effects of Track and Vehicle Wear on Freight-Car Performance

C. Thomas Jones, ENSCO, Alexandria, Virginia
Donald E. Gray, Federal Railroad Administration

Track and vehicle wear affect the dynamic performance and therefore the economic performance of the railcar-track system. A multiphase test program has been designed to determine the relationship between the dynamic performance of freight vehicles and track condition, vehicle-component wear, and variations in track structure. The first part of this program has been completed, i.e., the development of test, instrumentation, and analysis techniques and the determination of their applications to a baseline dynamic-performance test. The test methodology involves dynamic testing of a high-travel car and a reference or low-travel car. Two test tracks at the Transportation Test Center were used, the facility for accelerated services testing track and sections of the railroad test track. The instrumentation for each test vehicle included precision accelerometers to measure accelerations on the car body, bolsters, and trucks and instrumented wheel sets to measure lateral and vertical forces on the wheels. The analysis of the acceleration data is based on the use of six degrees of freedom, or rigid-body modes, for each primary mass (car body and truck). Statistical processing of the computed modal data is used to determine the effects of track structure and condition on vehicle performance. Transmissibility between truck and car body is calculated to determine the effect of component wear on vehicle performance. Finally, statistical processing of wheel-rail forces is used to obtain lateral-to-vertical force ratios and lateral wheel forces as functions of the track section. The instrumentation and data-processing techniques designed for this program proved effective in evaluating freight-car dynamics. Evaluation of the effects of variations in track structure on vehicle dynamics led to the following conclusions: (a) track containing unsupported bonded joints produced the highest car-body accelerations; (b) curves greater than 4 degrees and discrete events such as turnouts produced high accelerations and wheel forces; and (c) variations in track and roadbed such as ballast-shoulder width and depth, spiking patterns, tie material, and rail anchor type had little if any effect on the dynamic response of the vehicle.

The dynamic performance of the railcar-track system has a direct effect on the economics of railroad operations in terms of lading damage and maintenance costs. This performance changes with accumulated use as a result of degradation in the track structure and the vehicle components.

To establish a relationship between the dynamic performance of freight vehicles and the wear of track and components, a multiphase dynamic-performance test program (1) was designed as one part of phase 1 of the facility for accelerated services testing (FAST) program (2). The specific objectives of the dynamic performance test program are

1. To establish the relationship between ride performance and track condition,
2. To establish the relationship between ride performance and vehicle-component wear, and
3. To quantify the dynamic responses of freight vehicles to different track structures.

This paper provides a description of the test, instrumentation, and analysis techniques developed for the program. The results of the first in a series of dynamic performance tests are also presented.

TEST DESCRIPTION

The test methodology compares the dynamic performance of two freight vehicles that have traveled different distances. One car, designated the high-travel car, is operated at an accelerated service rate as a part of the FAST test train and the second car serves as a reference vehicle, or low-travel car, for comparative analysis. Two test tracks located at the Transportation Test Center are used, the FAST track and sections of the railroad test track (RTT). Tests are to be conducted in 80 000-km (50 000-mile) increments up to a maximum of 480 000 km (300 000 miles).

A baseline test was conducted in February 1977. The test consist included a locomotive, two 91-Mg (100-ton) hopper cars, and the Federal Railroad Administration T-5 data-acquisition car. The reference and the high-travel hopper cars were instrumented with precision servoaccelerometers. The reference vehicle was also equipped with instrumented wheel sets for measuring wheel-to-rail vertical and lateral forces. Signals from the instrumentation system were cabled to the T-5 data-acquisition car and recorded in digital form on magnetic tape. The test consist was operated at a constant 48-km/h (30-mph) speed over the 7.7-km (4.8-mile) FAST track, and data were recorded from both cars.

The FAST track has 22 separate sections, each of which has a different track structure and roadbed composition. Hence, operation of the test consist over this track provided data that could be used to quantify vehicle response to differing track and roadbed compositions. Second, the FAST track is subjected to accelerated service (approximately five times that experienced on a typical operating railroad). Thus, the baseline test provides an initial set of reference data for determining the effect of track degradation on ride performance.

Tests were conducted on a section of the RTT at speeds of 16, 32, 48, 64, and 80 km/h (10, 20, 30, 40, and 50 mph). The purpose of this phase of testing was to provide baseline data for determining the relationship between vehicle-component wear and ride performance. The RTT is subjected to relatively light traffic and, therefore, track variation with time has minimal effect. Hence, the subsequent tests will isolate the effects of vehicle-component wear on ride performance.

To correlate the data acquired during the dynamic performance tests with component wear and track degradation, measurements were made of pertinent car-truck wear surfaces and of the track geometry. These measurements will be repeated as travel is accumulated on both the car and track.

INSTRUMENTATION

The instrumentation developed for this test program consisted of

1. Servoaccelerometers,
2. A wheel-force-measurement system,
3. A speed and location system, and
4. A data-acquisition system.

Figure 1 is an overall system block diagram of the instrumentation.

Twenty force-balance servoaccelerometers were used to measure the accelerations on the car body, the bolsters, and the trucks of each test vehicle. Figure 2 shows the location and orientation of the accelerometers. Special mechanical isolators were used to filter out the high-amplitude, impulse-type accelerations that are potentially damaging to these precision instruments. An additional benefit derived from the isolation was that of maximizing the effective resolution of the acceleration measurement.

Two instrumented wheel sets built by the American Association of Railroads were used to measure wheel lateral and vertical forces on the reference car. Strain gauge bridges mounted on the wheel plate provided signals proportional to both lateral and vertical forces. The lateral signal is of a continuous nature, and the vertical signal consists of four outputs per wheel revolution.

The speed of the consist was obtained from an optical shaft encoder mounted on the T-5 data-acquisition vehicle. The encoder, which was mechanically driven by the car wheel, provides a pulse train output whose frequency is proportional to car speed.

The location of the test consist along the track was determined by a capacitive sensor mounted on the test vehicle. Metal targets were attached to the ties marking the beginning and end of each test section, and the passing of the consist over the targets generated a voltage pulse that was recorded on magnetic tape.

The primary elements of the T-5 data-acquisition system are a Raytheon 704 minicomputer, a 2032 bytes/cm (800 bytes/in) tape recorder, signal conditioning and filtering electronics, and an analog chart recorder.

The analog signals routed to the data-acquisition system are conditioned and filtered for compatability with the recording system. The conditioned signals are then converted to a 12-bit digital word and recorded on magnetic tape at a rate of 128 samples/s. For the purpose of visual analysis of data during testing and to ensure that the measurement and recording systems are functioning properly, selected channels of the digital data are passed through a digital-to-analog converter and the resultant time histories are displayed on a six-channel chart recorder.

DATA REDUCTION AND ANALYSIS TECHNIQUES

For the purpose of this study, the measured accelerations were reduced to accelerations with respect to a generalized coordinate system for both the car body and the truck. A right-hand Cartesian coordinate system was used that had its origin located at the geometric centroid. Accelerations with respect to the generalized coordinate system are referred to as modes. The modal representation of accelerations for each of the primary masses (car body and truck) offers two distinct advantages in the analysis of dynamic performance. First, the modes are conceptually easy to visualize,

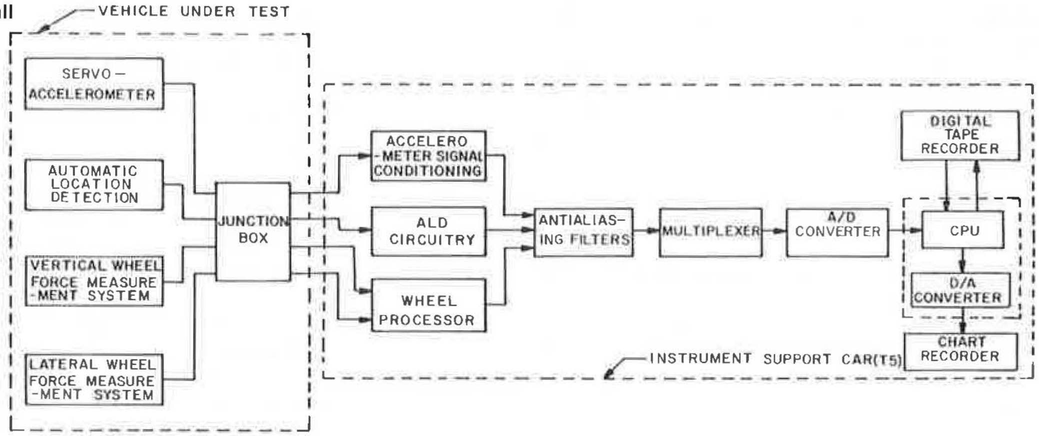


Figure 1. Block diagram of overall system.

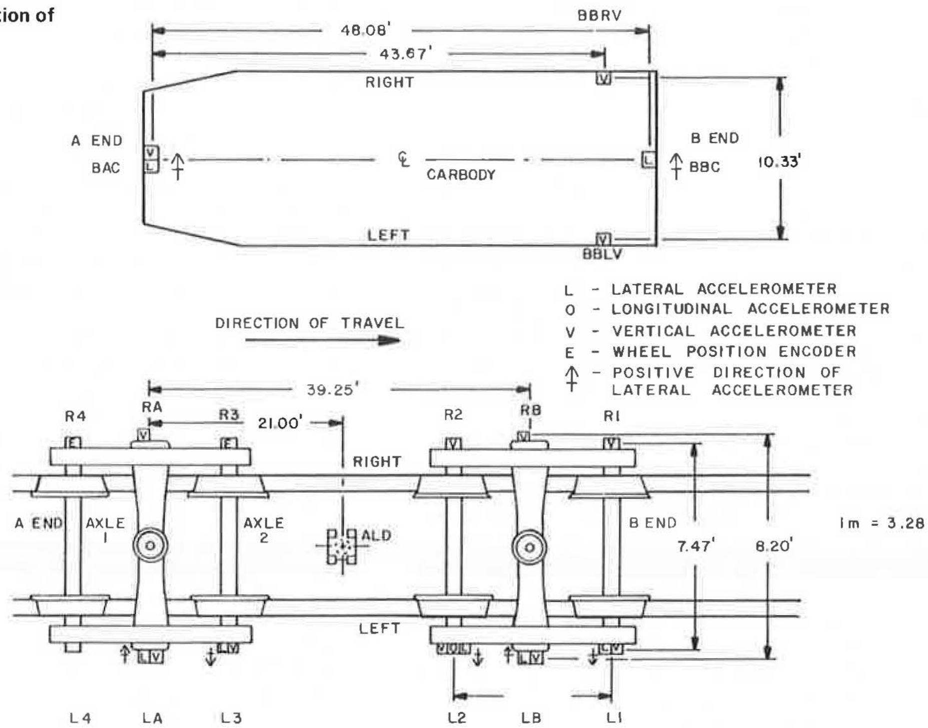


Figure 2. Locations and orientation of accelerometers.

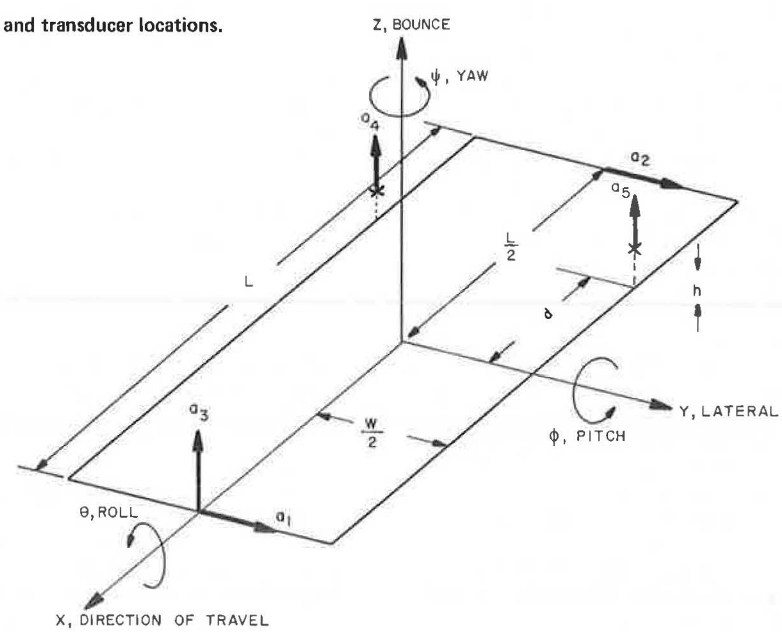


Figure 3. Car-body conventions and transducer locations.

Figure 4. Truck conventions and transducer locations.

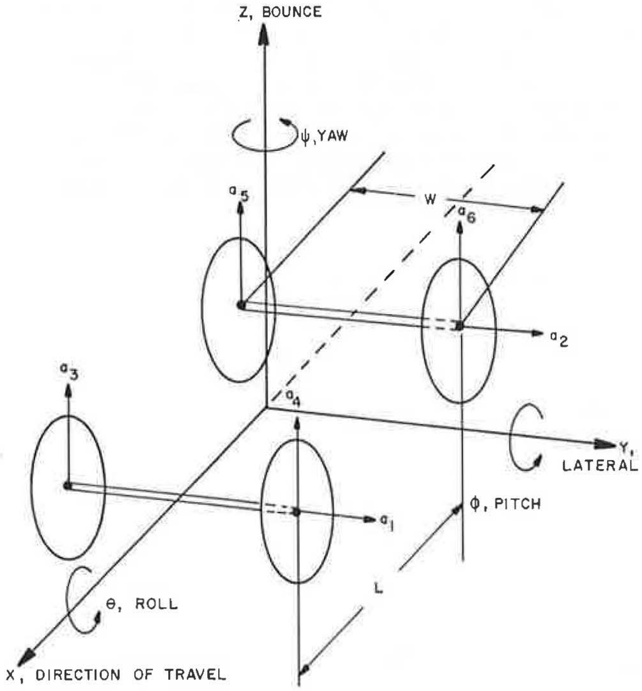
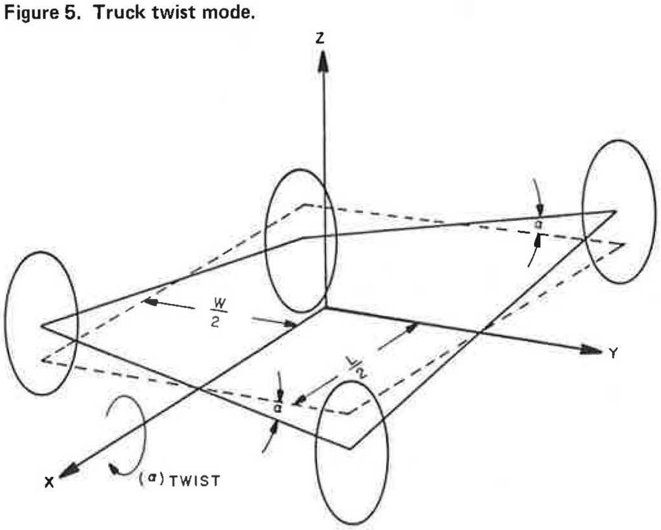


Figure 5. Truck twist mode.



which facilitates subsequent analysis. Second, by assuming that the selected modes account for most of the acceleration experienced by the car body and the truck, a linear combination of modal accelerations can be used to determine the acceleration level at any point on the car body.

The car-body accelerations were assumed to be made up of six modes that correspond to the six rigid-body degrees of freedom. Three of these modes are the linear accelerations along the axes of the Cartesian coordinate system. The remaining three modes are the angular accelerations about each of the three principal axes. The modes are referred to as longitudinal, lateral, bounce, roll, pitch, and yaw and denoted by $\ddot{x}$, $\ddot{y}$, $\ddot{z}$, $\ddot{\phi}$, and $\ddot{\psi}$ respectively (see Figure 3). The double dot above each symbol denotes a double differentiation with respect to time.

Longitudinal acceleration ($\ddot{x}$) is primarily influenced by train handling and is not considered in this study. The remaining modes are to be determined by measurement of the five accelerations indicated by boldface arrows labeled a_i (i = 1, 2, 3, 4, 5) in Figure 3. Note that a_4 and a_5 lie a distance h above the plane of the other measurements. If these measurements are written in terms of their modal components,

$$a_1 = \ddot{y} + (L/2)\ddot{\psi} \tag{1}$$

$$a_2 = \ddot{y} - (L/2)\ddot{\psi} \tag{2}$$

$$a_3 = \ddot{z} - (L/2)\ddot{\phi} \tag{3}$$

$$a_4 = \ddot{z} + d\ddot{\phi} - (W/2)\ddot{\theta} \tag{4}$$

$$a_5 = \ddot{z} + d\ddot{\phi} + (W/2)\ddot{\theta} \tag{5}$$

and, if

$$F \equiv 2d + L \tag{6}$$

then, by solving for the modes, one obtains

$$\ddot{y} = (a_1 + a_2)/2 \tag{7}$$

$$\ddot{z} = [2da_3 + (L/2)(a_4 + a_5)]/F \tag{8}$$

$$\ddot{\theta} = (a_5 - a_4)/W \tag{9}$$

$$\ddot{\phi} = (a_4 + a_5 - 2a_3)/F \tag{10}$$

$$\ddot{\psi} = (a_1 - a_2)/L \tag{11}$$

The definitions and determination of the truck modes were similar to those used for the car body with the addition of the twist mode. As before, a right-hand Cartesian coordinate system was used that had its origin at the geometric center of the truck in the plane of the axles as shown in Figure 4. These modes are directly analogous to those of the car body and are given the same names and symbols. Also shown in Figure 4 are the locations of accelerations measured on the truck denoted a_i (i = 1, 2, . . ., 6).

The trucks consist primarily of two axles and two side frames that behave as rigid bodies within the truck system. These subcomponents can displace angularly with respect to one another, which results in an asymmetric mode referred to as twist. The twist angle (α) is a function of the distance along the x axis as shown in Figure 5 and is expressed in units of radians per unit length. If the small-angle approximation ($\sin \alpha = \alpha$) is made and the convention that twist and roll have opposite signs is remembered, one can write the measured accelerations in terms of the truck modes as

$$a_1 = \ddot{y} + (L/2)\ddot{\psi} \tag{12}$$

$$a_2 = \ddot{y} - (L/2)\ddot{\psi} \tag{13}$$

$$a_3 = \ddot{z} - (W/2)\ddot{\theta} - (L/2)\ddot{\phi} + (WL/4)\ddot{\alpha} \tag{14}$$

$$a_4 = \ddot{z} + (W/2)\ddot{\theta} - (L/2)\ddot{\phi} - (WL/4)\ddot{\alpha} \tag{15}$$

$$a_5 = \ddot{z} - (W/2)\ddot{\theta} + (L/2)\ddot{\phi} - (WL/4)\ddot{\alpha} \tag{16}$$

$$a_6 = \ddot{z} + (W/2)\ddot{\theta} + (L/2)\ddot{\phi} + (WL/4)\ddot{\alpha} \tag{17}$$

This system of equations can be solved for the truck modes, which gives

$$\ddot{y} = (a_1 + a_2)/2 \tag{18}$$

$$\ddot{z} = (a_3 + a_4 + a_5 + a_6)/4 \tag{19}$$

$$\ddot{\theta} = (a_6 - a_5 + a_4 - a_3)/2W \tag{20}$$

$$\ddot{\phi} = (a_6 + a_5 - a_4 - a_3)/2L \qquad (21)$$

$$\ddot{\psi} = (a_1 - a_2)/L \qquad (22)$$

$$\ddot{\alpha} = (a_6 - a_5 - a_4 + a_3)/WL \qquad (23)$$

Based on equations 7 through 11 for the car body and equations 18 through 23 for the truck, the individual measured acceleration can be transformed into 11 mode-acceleration time series. The mode-acceleration time series are then processed by using standard statistical techniques to provide root-mean-square (RMS) values, 95th and 99th percentile levels, histograms, and probability densities. The RMS values of the modes were derived for each of the 22 sections of the FAST track. This technique provides data that can be used to quantify the effect of track and roadbed composition on the dynamic performance of the truck and the car body. Data

Figure 6. Relationship between truck vertical-mode acceleration and track section.

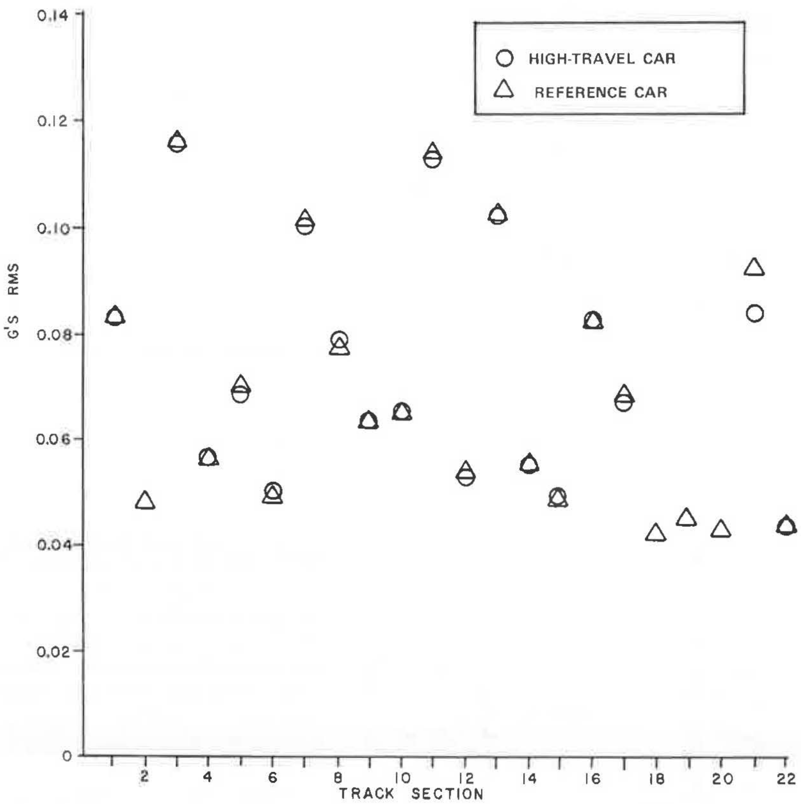
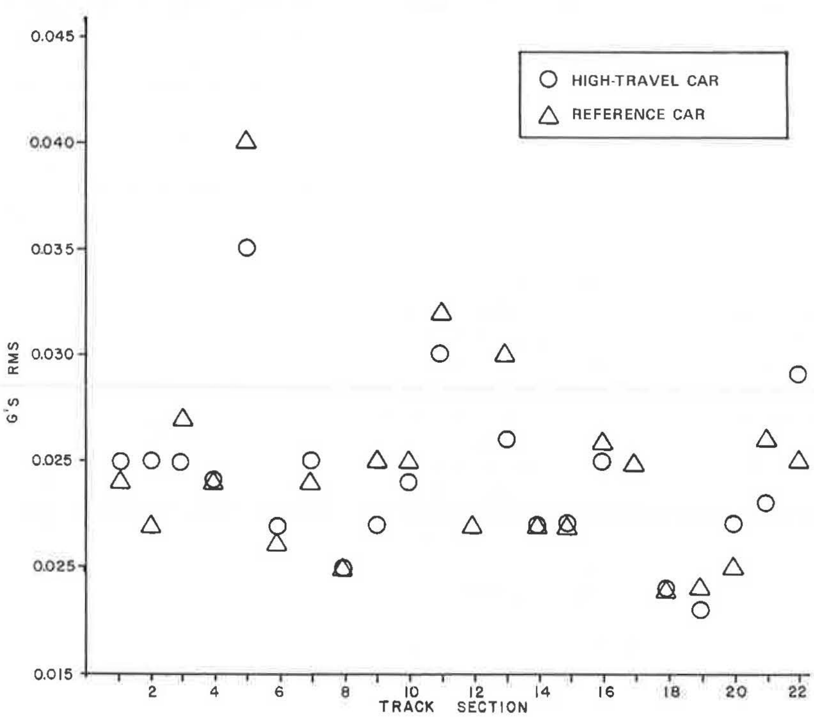


Figure 7. Relationship between car-body vertical-mode acceleration and track section.



presented in this manner will also be used in future tests to determine the effects of track degradation on vehicle performance.

To assess the effect of component wear on the ride performance of freight vehicles, the transmissibility between truck and car-body modes was determined. The transmissibility can be thought of as a characteristic of the freight car system that is independent of the track condition over which the car is operated. The changes in transmissibility characteristics with accumulated travel can, therefore, be directly attributed to changes in the freight-car components.

The transmissibility or gain was formed in the frequency domain by using power spectral densities (PSDs). The mode-acceleration time series were first transformed to a Fourier representation by using a fast Fourier transform. Then the PSD of a given modal acceleration was generated by multiplying the Fourier transform by its complex conjugate. The power associated with each frequency increment of a selected car-body mode was then divided by the power associated with the corresponding frequency increment of a selected truck mode. The result is the spectral distribution of the mean-square gain between the two selected modes.

The primary parameter used in the analysis of wheel-force data was the lateral-to-vertical (L:V) force ratio. This ratio is an important safety index that is used to determine the potential of rail rollover and wheel-flange climb. As discussed above, the lateral wheel forces were measured and recorded continuously, but the vertical forces were measured accurately only four times per revolution. Thus, to construct a continuous L:V time series, the four vertical measurements were averaged over each wheel revolution. The continuous lateral-force time series were then divided by the average vertical force for each wheel revolution. Statistical processing similar to that used for the acceleration modes gave the L:V ratios and lateral wheel forces as functions of track section.

RESULTS AND CONCLUSIONS

The instrumentation and data processing techniques developed for this test program proved highly successful for the evaluation of the dynamic performance of freight vehicles. The use of RMS modal accelerations yielded clear concise results that correlated well with physical phenomena.

A plot of modal acceleration versus track section is shown in Figure 6. The vertical-mode accelerations are shown for both the high-travel and the reference cars. This figure indicates that there is considerable variation in the truck vertical-mode acceleration from one track section to the next but that the accelerations of both trucks are nearly identical. This second observation was anticipated because the accelerometers are mounted on the journal box adapters and, in essence, are directly coupled to the rail. This is an important observation because it implies that the modal acceleration of the truck can be used as an index of track condition.

Car-body vertical-mode accelerations are plotted against track section in Figure 7. Again, variations in modal accelerations are shown for different track sections. As opposed to the truck modes, however, the car-body accelerations of the reference and the high-travel cars exhibited some differences. This was anticipated because car-body modal accelerations are a function of both track geometry and suspension components and the components may differ from car to car. By comparing Figures 6 and 7, one finds attenuation factors of between 2:1 and 5:1 between truck and car-body modes.

Conclusions related to the objective of quantifying the dynamic response of freight vehicles to different track structures are as follows. Variations in track structures (such as ballast-shoulder width and depth, spiking patterns, tie material, and rail anchors) had little if any effect on truck and car-body accelerations or wheel forces. In contrast, curves greater than 4 degrees and discrete events (such as turnouts) had a marked effect on vehicle dynamics. The highest car-body accelerations were those experienced on section 5 of the FAST track, which contains unsupported bonded joints. Because truck modal accelerations were moderate to low over this same section of track, it can be theorized that this particular track structure excites a resonance in the vehicle suspension system.

In summary, the techniques that were developed for this program proved effective in evaluating the dynamic performance of freight vehicles and for determining the effects of variations in track structure on that performance. The use of these techniques in subsequent phases of the test program will provide the necessary input for the evaluation of track and component wear as it affects vehicle dynamic performance.

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