Intermodal Cars -- New Developments

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ABSTRACT

Intermodal railcars used for "piggyback" trailer and container transportation are becoming the subject of increasing design interest. In response to increasing fuel costs and competitive pressures, a number of new car designs have been developed, some to the hardware stage. From the designs it is apparent that two common goals are the reduction of tare weight and aerodynamic drag. The ride quality and dynamic stability characteristics of lighter weight cars are key issues which are being investigated. This paper describes the features of several new intermodal railcar concepts. It also reports on a cooperative government industry test program designed to quantitatively define and measure the ride vibration characteristics of current and experimental intermodal flatcars. Under the program, the procedures, equipment and analytical techniques suitable for evaluating car performance under both controlled test and actual service conditions were developed and successfully employed. Preliminary results indicate that substantial weight reductions can be achieved without adversely affecting ride quality or dynamic stability.

INTRODUCTION

The objectives of this paper are to review developments in the area of intermodal railcar design which represent the beginning of a new cycle in the evolutionary process and to describe a method of testing and evaluating the dynamic performance of new car prototypes under actual service conditions.

CAR DESIGN OBJECTIVES

As the backbone of the intermodal hardware system the flatcar has become the subject of increasing design interest due to a number of trends. In combination, these trends have produced several distinct objectives for all new flatcars which are:

- Reduced train resistance
- Ability to carry 45 foot trailers
- Improved service capabilities
- Reduced acquisition and maintenance costs
- Reduced man-machine interaction
- Reduced clearance requirements

The driving forces behind these objectives can be readily traced to:

- Increased fuel costs
- Shortages of motive power
- Increased cost of investment capital
- Tightened competition within and between modes
- Need to extend service in northeast metropolitan areas

Consideration of design changes is appropriate at this time because there is a need to produce new intermodal cars to replace those that have reached the end of their useful life and to provide the fleet expansion needed to handle an annual 10 to 12% growth rate in intermodal carloadings. A production rate of 6000 new cars a year may be necessary.

INTERMODAL IMPORTANCE

The importance of intermodal traffic to the railroads is based on its revenue potential. Shippers of high value manufactured goods pay the highest revenue rates, but they demand prompt, reliable damage free service in return. Dedicated, run-through intermodal trains appear to be the only way the railroads can meet the service requirements.

Such service is proving to be well worth the effort, for its revenues can assure the long term economic viability of the railroads. It is likely that within a few years many railroads will derive 20 percent or more of their total revenues from intermodal traffic. Recently it was reported that one major railroad has already attained the 25% mark. Noting that in terms of traffic volume, intermodal loads account for only 15% of the total shows that its revenue potential is remarkably high. As a result of its earning power the investment in new intermodal equipment appears attractive. The opportunity will soon exist to introduce designs that are better suited to the needs of the times. Although the design requirements have yet to be defined, and probably will not be until the Federal Railroad Administration (FRA) Intermodal Systems Engineering Program has been completed, a number of concepts have been formulated within the industry toward achieving the basic objectives cited above.

DESIGN CONSIDERATIONS

The design of an intermodal car is a challenging task. The current car presents a deceptively simple image. It has been doing its job well for the past twenty years and when it is superseded the new equipment must be superior in meeting the needs of the industry.

Will the new design be evolutionary or revolutionary? For good reason the industry has been cautious and slow to accept revolutionary hardware. At the present time the new concepts being proposed range from alterations of the current design to radical new approaches to the idea of moving trailers and containers by rail.

In connection with FRA's Intermodal Systems Engineering Program new ideas are being solicited for study and evaluation. Each will be subjected to a comprehensive assessment of its merits based on the interrelationships between the car and the other components in the intermodal system.

The systems aspect must be emphasized. In order for the railroads to benefit from the intermodal potential, the cost of providing the service must be minimized. A well-matched, efficient, and cost effective hardware system will be essential. Each element's characteristics must be established in concert with those of the other equipment toward maximizing the system's overall performance in terms of service and return on investment. This means that all the equipment, facilities and sub-systems used between the shipper's and consignee's loading docks must be considered. The intermodal rail car is certainly one of the major elements of the system. Typical considerations for a new intermodal car include:

° Size(s) of trailers/containers to be carried

° Number of loads per car

° Method of loading

° Tare weight per load

° Aerodynamics (resistance and stability)

° Number of axles

° Ride quality and stability

° Method of connection between cars

° Vertical and lateral clearances

° Automation of load securement devices

° Cost of acquisition and maintenance

° Durability

° Compatibility

° Interchangeability

NEW CONCEPTS

A review of the proposed concepts has disclosed a variety of ideas concerning these design considerations. Variations in the ranking of priorities is apparent reflecting tradeoffs made by the respective designers. However, without commenting on each concept's merits with respect to the design considerations, it is interesting to observe in what areas changes have been proposed in a few of the better known concepts.

The Santa Fe "Six Pack" concept utilizes articulated joints with 2 axle trucks at the five intermediate points of the six-unit car sets. This arrangement reduces by two thirds the number of couplers and brake reservoirs in a 60 trailer train. A box type structure serves as the longitudinal member and supports cantilevered aprons for the trailer wheels. The slenderness of the center structure allows it to be straddled by the landing legs of the trailers eliminating the need for their adjustment. Car section length will correspond to the length of trailer to be carried resulting in a minimal spacing between successive trailers and lower aerodynamic drag. Twenty eight inch diameter wheels contribute to a 16 inch reduction in vertical loaded height. Compared to conventional equipment, the Santa Fe design has achieved a weight reduction of 34%. The cars are intended only for captured service and require lift type loading capability at the terminals.

The Trailer Train Company is con sidering modifications to their conventional car which will permit the carriage of two 45 foot trailers. They have also reportedly been looking at a new configuration that would carry single 45 foot trailers. In both cases, lift type loading operations appear to be required.

Pullman-Standard utilized a skeleton type longitudinal frame in their design for a light weight intermodal container car which resulted in the construction of two experimental light weight cars in 1969. Both cars carried two 40 foot or four 20 foot containers. They incorporated several innovations in load securement devices but their primary achievement was a 17% to 31% reduction in weight compared to the conventional cars.

The Southern Pacific has received for testing, a unique design wherein two 40 foot containers are carried in a stacked configuration such that their overall height is 18'-6" above the rails. A car length of 63 feet saves 26 feet of train length for each pair of containers. Using articulated joints between cars a weight savings of 40% may be acheived over conventional cars

The Bimodal Corporation's concept is to eliminate the need for a rail car completely by constructing trailers with a rail wheel system included in addition to the highway suspension components. Trains would be made by coupling numbers of such trailers directly together using devices included in the trailers. With elimination of the flatcar, the net to tare ratio for the rail mode would improve dramatically and there be no problem with tunnel or bridge clearances.

The Paton Corporation has a concept which reduces the rail-based equipment to a minimum configuration. Low tare weight, high utilization rate, flexibility and the ability to operate thru the restricted clearance areas of the Northeast region have been cited as its predominant features.

LIGHTNESS vs. STABILITY

A common attribute of all the new concepts is an improvement in the net to tare weight ratio toward the goals of improving fuel efficiency, increasing locomotive productivity and increasing component service life.

Of particular concern to the industry are the ride quality and dynamic stability characteristics of lighter weight rolling stock. The reduced mass of the carbody, combined with the possiblity of a higher loaded center of gravity, could render such a car unacceptable.

However, because of the major payoffs in intermodal efficiency that could result from lighter weight rolling stock, the effort to acquire and provide reliable, quantitative information on the relationship between car weight and dynamic performance to those who will be working on new car concepts and performing system trade-off studies is very important. The capability to measure and characterize the ride vibration environment of intermodal cars will also aid in the evaluation of specific design alternatives once a basic concept or configuration has been established.

Under a cooperative FRA/industry research program the procedures, equipment and processing techniques have been developed and tested toward providing this capability. Intended to yield information about car behavior in the operational environment, the techniques can be applied to the evaluation of all of the new car designs.

FRA-INDUSTRY TEST PROGRAM

The program was originally planned to evaluate the performance of two experimental lightweight intermodal cars produced by Pullman-Standard in 1969. For comparison, a conventional Trailer Train TTAX car was included in the evaluation. Both lightweight cars were originally configured to carry containers only with landbridge operations in mind. At a later date, one of them was modified to carry trailers by adding fixed stanchions and support plates for the trailer wheels. Both cars carry two loads, as does the conventional flatcar. However, one of the lightweight cars weighs 47,800 pounds compared to 69,300 pounds for the conventional car resulting in a savings of 31%. The lightweight trailer car weighs 57,100 pounds for a savings of 17%.

DESCRIPTION OF TESTS

The Lightweight Flatcar Evaluation program consisted primarily of two types of tests whose geographic locations are shown in Figure 1. The first test, referred to as the Vehicle Dynamic Characterization (VDC) test, was conducted in a controlled environment to provide information on the dynamic or elastic nature of flatcars. The second test was conducted in revenue service environments to quantify the acceleration environment actually experienced by the flatcars and loads. This test is referred to as an Over-the-Road (OTR) test.

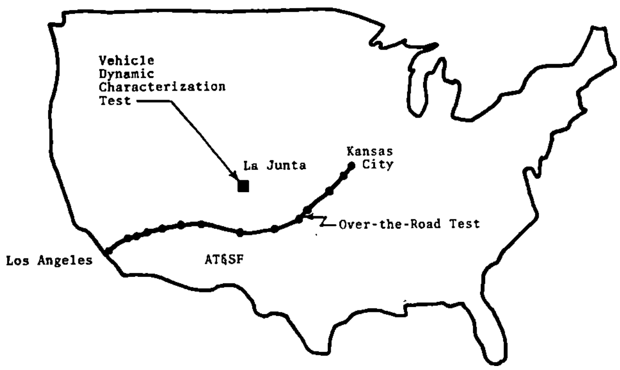

Figure 1. Test Zone Locations

The test consist was comprised of the three instrumented test flatcars and the FRA/DOT Data Acquisition Vehicle T-5. The test vehicles included a conventional TTAX (973799) and two lightweight flatcars designed by Pullman-Standard. One lightweight flatcar (TLDX 62) was designed for container service only and the other (TLDX 61) was modified for trailer service only. The test matrix included empty, half loaded, and fully loaded configurations for both the VDC and OTR test series.

The VDC test was conducted on two test zones near La Junta, Colorado. The test zones were established on a 1 mile section of class 3 tangent track and a 3 mile section of class 5 tangent track. Accelerations were measured on the vehicle and recorded at consist speeds of 10, 15, 20, 30, and 40 mph over the first test zone and at 40, 50, 60, 70, and 79 mph over the second zone.

The OTR test series was conducted on main line track between Argentine Yard, Kansas City, Missouri and Hobart Yard, Los Angeles, California. Data were recorded in twelve 10 mile test zones representing a cross section of track class and structures, and accelerations were measured and recorded while the consist was passing through these zones. During the OTR test, there was no control over consist speed, and as a result, speed varied from 20 to 79 mph during measurements. This method of testing includes the effects of train handling which were not included in the VDC test.

DESCRIPTION OF INSTRUMENTATION

Signals recorded during the VDC and OTR tests consisted of speed, automatic location detection (ALD), and up to 120 accelerations. The latter of these required a number of ancillary components in addition to the accelerometer transducer as shown in Figure 2.

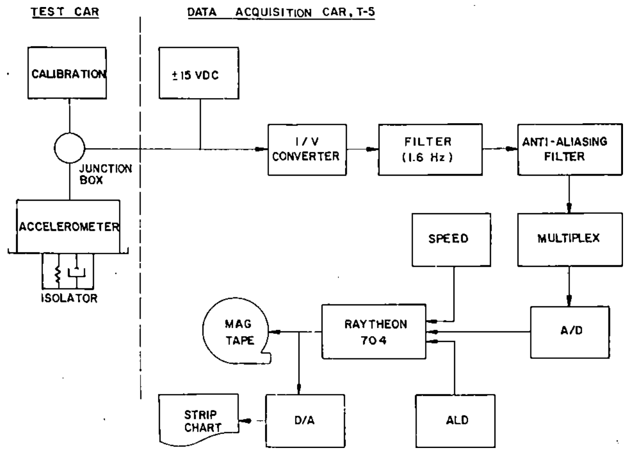

Figure 2. Schematic of Instrumentation and Recording System

For the purpose of this evaluation program 120 precision servo-accelerometers were mounted on the carbody, loads, and axle journal bearings of the three test flatcars. Accelerometers on the car and loads were 5g accelerometers while 30g accelerometers were used on the axles. An excitation voltage was supplied from T-5 and calibration signals were input at junction boxes located on each test vehicle.

Each accelerometer was mounted in a mechanical isolator for protection from high frequency accelerations of large amplitude present in the rail environment. Mismatched rail joints, for example, can produce impulses as large as a hundred g's. The isolators with a natural frequency of 150 Hz were designed to low pass accelerations. The signal from the accelerometer transducer was transmitted along the consist through a maximum of 250 feet of shielded cable in a current mode to avoid voltage drop due to line resistance.

In the Data Acquisition Car the signal was converted to a voltage. This signal was then filtered using a low pass single pole filter (-6dB/octave) with a corner frequency of 1.6 Hz. This filter is used to offset the effect of acceleration amplitude increasing with frequency and thereby produced an increase in the system resolution. Next the signal was anti-aliased using a 4-pole (-24dB/octave) Bessel filter with a corner frequency of 30 Hz. This type of filter provides a linear phase shift which is essential to data processing requiring phase synchronization.

The fully conditioned signal was multiplexed and converted to a 12 bit digital word at a rate of 128 samples per second. The digitized signal was stored in the on-board mini-computer (Raytheon 704) and buffered onto a magnetic tape. Selected channels were passed through the D/A converter and displayed on a strip chart recorder for real time examination of data. This system was also used to verify the data tapes after tests.

DATA REDUCTION AND ANALYSIS

In order to analyze the acceleration environment of the flatcar/load system use is made of the technique of superposition. This method presupposes that the acceleration, a, at any point on the flatcar may be thought of as being the sum of contributions of rigid, a_r, and elastic body, a_e, accelerations.

$$a = a_r + a_e \quad (1)$$

The components of a_r and a_e are referred to as the modes or modal coordinates. The use of modal coordinates offers a number of distinct advantages in the analysis of rail vehicles. First, the modal coordinates are conceptually easy to visualize and as a result are a great aid to the design engineer. Second, since these coordinates by definition are orthogonal or independent, phenomena such as cancellation and reenforcement do not obscure details of analysis. Thirdly, modal coordinates may be used to obtain the actual acceleration level at any point on the vehicle.

The vehicle subsystems treated in this evaluation program are the load, carbody, and axle. The acceleration response of each subsystem is modeled as a separate freebody. A Cartesian coordinate system is established at the geometric centroid of each subsystem with positive x in the direction of travel, positive y to the left when viewed in the direction of travel, and z positive upwards. The rotational coordinates are θ, ϕ, and ψ about the x, y, and z axes respectively. Figure 3 illustrates the sign conventions for the carbody coordinate system. The load and axle coordinate systems are identical with the origin at their respective centroids.

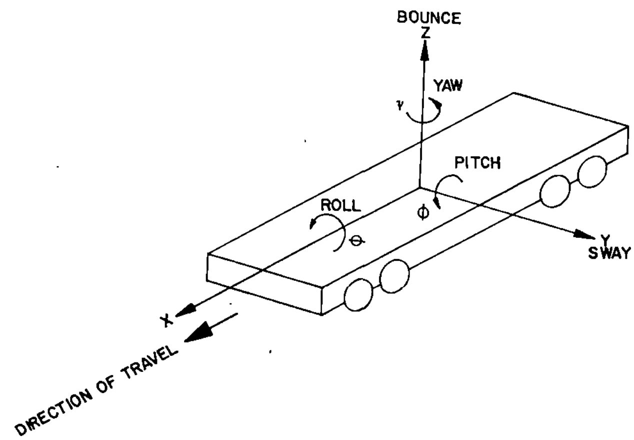

Figure 3. Rigid Body Modal Coordinates

The rigid body modal coordinates are composed of three linear and three angular acceleration elements. Linear accelerations are parallel to the axes defined above and angular acceleration are about these same axes. For the purposes of identification, the acceleration along or about a given axis is indicated by a double dot over that coordinate, used to show double differentiation with respect to time. Furthermore, modal coordinates are subscripted with $_0$. Thus the longitudinal modal coordiante is $\ddot{x}_0$; the lateral modal coordinate, referred to as sway, is $\ddot{y}_0$; and the vertical modal coordinate, referred to as bounce, is $\ddot{z}_0$. The angular modal coordinates are roll, pitch and yaw denoted $\ddot{\theta}_0$, $\ddot{\phi}_0$, and $\ddot{\psi}_0$ respectively.

The contribution of rigid body modes to linear accelerations may be written as the modal coordinate in the given direction plus the contribution due to angular modal coordinates about the two remaining axes. Thus, denoting the rigid body acceleration components with the subscript r:

$$\ddot{x}_r(y,z,t) = \ddot{x}_0(t) + \ddot{\phi}_0(t)z - \ddot{\psi}_0(t)y \quad (2)$$

$$\ddot{y}_r(x,z,t) = \ddot{y}_0(t) + \ddot{\theta}_0(t)z + \ddot{\psi}_0(t)x \quad (3)$$

$$\ddot{z}_r(z,y,t) = \ddot{z}_0(t) + \ddot{\theta}_0(t)y - \ddot{\phi}_0(t)x \quad (4)$$

Note that the small angle approximation, cosine of the angle of deflection is approximately unity, has been made.

Next certain assumptions concerning the elastic body modal coordinates must be made. In the most general case an elastic body may bend and twist about any axis. Associated with each elastic deformation is an infinite set of modes or harmonics. Experience has shown that the more important contributions to the rail vehicle vibration environment occur below 30 Hz. Thus based on structural considerations and experience, it was determined that for the carbody it was necessary to include only the first and second bending modes about the y-axis and first and second twist mode, more commonly referred to as torsion, about the x-axis. These are denoted α_1, α_2, β_1 and β_2 respectively and are illustrated in Figure 4. The loads were found to require only the first bending mode about the z-axis. This mode is called lateral bending to distinguish it from the previous bending modes. The axle is treated as a purely rigid body so that the set of modal coordinates associated with the axle contain neither bending or torsion modes.

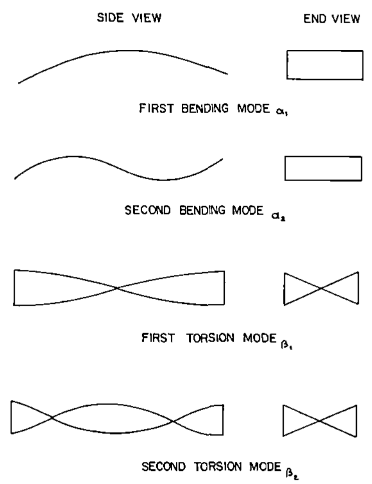

Figure 4. Elastic Body Modal Coordinates

The shapes of the elastic modes are described by power law expressions. This is considered reasonable since the elastic deformations of rail vehicles are small in amplitude and result in relatively smooth shapes. Three terms are required to model the carbody first bending mode while two terms were required for all other modes. Based on considerations of symmetry these are written as:

$$f_1(x) = 1 + a_1x^2 + a_2x^4 \quad \text{for } \alpha_1; \tag{5}$$

$$f_2(a) = X + b_1x^3 \quad \text{for } \alpha_2; \tag{6}$$

$$g_1(x) = X + c_1x^3 \quad \text{for } \beta_1; \tag{7}$$

$$g_2(x) = 1 + d_1x^2 \quad \text{for } \beta_2 \tag{8}$$

where the coefficient a_1, a_2, b_1, c_1, and d_1 are referred to as the mode shape coefficients and are to be determined. The mode shape of the load lateral bending mode is similar to Equation 5 but with a_2 set to zero. Elastic body contributions to carbody linear acceleration subscribed e are:

$$\ddot{y}_e(x,z,t) = [\beta_1(t)g_1(x) + \beta_2(t)g_2(x)]z \tag{9}$$

$$\ddot{z}_e(z,y,t) = [\alpha_1(t)f_1(x) + \alpha_2(t)f_2(x)] + [\beta_1(t)g_1(x) + \beta_2(t)g_2(x)]y. \tag{10}$$

Because of the assumption of small amplitude deflection, cross sections taken normal to the x-axis are not deformed. As a result elastic body deformations make no contribution to longitudinal accelerations.

Equations 2 through 4 are combined with Equations 8 and 9 to obtain expressions for the linear accelerations along the principal axes in terms of the modal coordinates. An example of this is illustrated by the vertical carbody acceleration

$$\ddot{z}_m(x,y,t) = A_0(t) + A_1(t)x^2 + A_2(t)x^4 + B_0(t)x + B_1(t)x^3 + C_0(t)xy + C_1(t)x^3y + D_0(t)y + D_1(t)x^2y \tag{11}$$

where

$$A_0(t) = \ddot{z}_0(t) + \alpha_1(t)$$

$$A_1(t) = \alpha_1(t)a_1$$

$$A_2(t) = \alpha_1(t)a_2$$

$$B_0(t) = \alpha_2(t) - \ddot{\phi}_0(t)$$

$$B_1(t) = \alpha_2(t)b_1$$

$$C_0(t) = \beta_1(t)$$

$$C_1(t) = \beta_1(t)c_1$$

$$D_0(t) = \ddot{\theta}_0(t) + \beta_2(t)$$

$$D_1(t) = \beta_2(t)d_1$$

and the subscript m denotes a modeled acceleration as opposed to a measured or observed acceleration. Expressions for the remaining components and masses can be written similarly. The time dependent coefficients defined in Equation 11 are determined from measured accelerations.

Each mass was individually instrumented with a specified number of servo-accelerometers: 17 on each carbody, 8 on each load, and 5 on each axle. The transducer location and orientation are shown in Figures 5, 6, and 7. A summary of the modes and measurements of each subsystem is given in Table 1. Each measurement of acceleration represents a single equation in terms of modal coordinates. Thus the set of 10 modal coordinates for the carbody is found using 17 equations. Similarly, the 7 load modal coordinates are found using 8 equations.

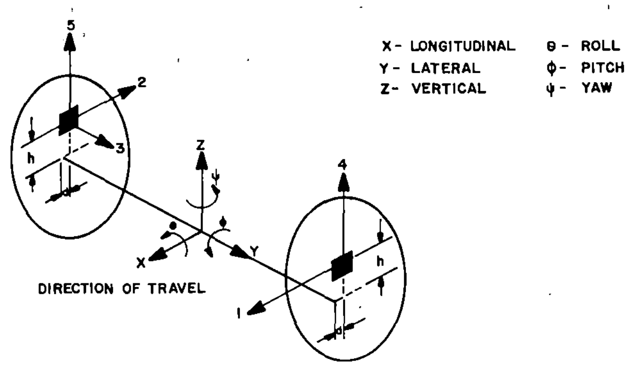

Figure 7. Axle Transducer Locations

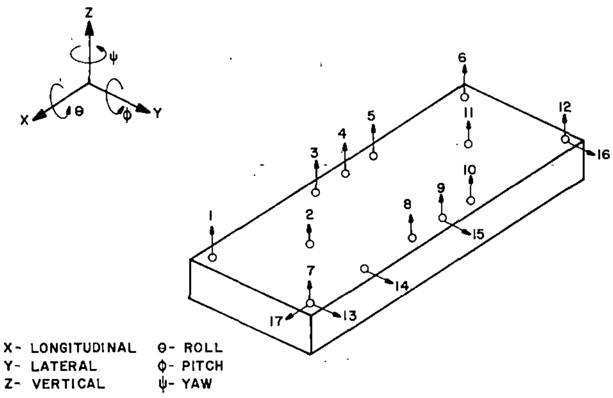

Figure 5. Carbody Accelerometer Locations

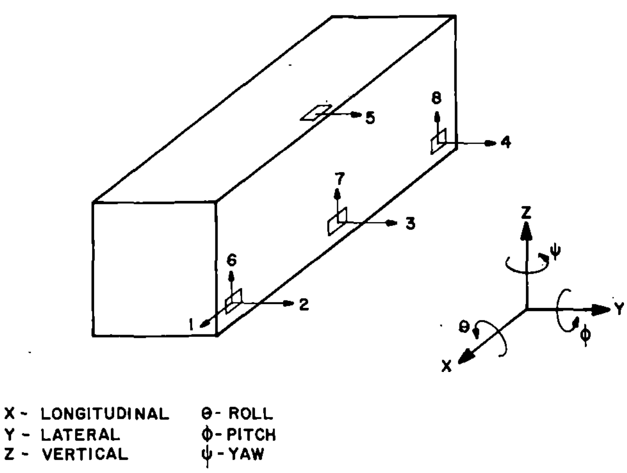

Figure 6. Load Accelerometer Locations

Table 1. Subsystem Modes and Measurements

SUBSYSTEM	RIGID BODY MODES	ELASTIC BODY MODES	TOTAL MODES	MEASURED ACCELERATIONS
TRAILER	6	1	7	8
CONTAINER	6	1	7	8
CARBODY	6	4	10	17
AXLE	5 *	0	5	5

These systems of equations are redundant, and the solution set of modal coordinates will not, except in the ideal case, satisfy the set of equations. For this reason the method of least squares is used to find that solution set which satisfies most nearly all the equations. The criterion of this method is that the sum of the squared errors be a minimum. This is accomplished by differentiating the expression for the sum of squared errors with respect to each modal coordinate individually and setting the result equal to zero. The result is a set of N equations in N unknowns which is solvable and yields the desired result. The set of equations corresponding to the number of measurements made is written as:

* Note that the axle pitch mode is not included since the axle itself coincides with the y-axis. Thus, pitch is simply wheel rotation.

$$[X] \ \{A\} = \{Z_m\} \tag{12}$$

where X is the coordinate matrix of the polynomial, A is the vector of the coefficients defined in Equation 11 and Z_m is the modeled acceleration vector. The condition of least squares is fulfilled by the expression

$$[X]^T \{Z_m\} = [X]^T \{Z\} \tag{13}$$

where Z is the measured acceleration vector.

Substitution of Equation 12 into Equation 13 yields

$$[X]^T[X]\{A\} = [X]^T \{Z\} \ . \tag{14}$$

Introducing the following definition

$$[Q] = [X]^T [X]$$

Equation 14 becomes

$$[Q]\{A\} = [X]^T \{Z\} \text{ , or finally} \tag{15}$$

$$\{A\} = [Q]^{-1}[X]^T\{Z\}. \tag{16}$$

Equation 16 thus defines the matrix operation required to convert measured accelerations on the carbody and load to the corresponding modal coordinate. The 5 axle modal coordinates are solved for using 5 equations in closed form requiring no fitting technique.

The A-vector for the carbody contains elements which are the sum of two modal coordinates (see Equation 11), one rigid and one elastic. In order to uncouple these the condition of dynamic equilibrium is used. This condition states that the net force or moment due to an elastic deformation is zero. The result is an equation for each coupled element of the A-vector of the form

$$R_0(t) = \sum_{n=0}^{2} \frac{Mn}{Mo} A_0(t) \tag{17}$$

where R_0 is the coupled rigid body modal coordinate and A_n is the corresponding subset of the A-vector and M_n is defined by

$$M_n = \int_{-L/2}^{L/2} x^{2n} \ \rho(x) dx$$

Here $\rho(x)$ is used in a more general sense to denote either a mass or polar mass moment of distribution. Detailed knowledge of the vehicle structure permits the uncoupling of rigid and elastic modal coordinates.

The time histories of accelerations measured on the carbody are thus reduced to modal coordinate time histories. As mentioned at the outset of this section, modal coordinates are useful in the analysis of rail vehicle performance, and the following data processing schemes are employed.

1. Goodness of Fit - A time series of the difference between each measured and modeled acceleration, $\ddot{z}$ and $\ddot{z}_m$, is created. This series, as well as the z series, is Fourier transformed and the power spectral density (PSD) calculated. The power in the two PSD's is calculated from 0 to 30 Hz, and the ratio of residual to measured power obtained. This parameter indicates the percent error incurred in the model.

2. Power Spectral Density (PSD) of Modal Coordinates - The entire time series of modal coordinates are also Fourier transformed and PSD's formed. The root mean square (rms) is calculated from 0 to 30 Hz.

3. Positive Zero Crossing (PZX) Histograms - The time series of each modal coordinate is analyzed to determine the number of zero crossings with positive slope which lie in ranges or bins of given amplitudes. These results are tabulated in the form of numerical histogram on a mile by mile basis, i.e. one modal coordinate PZX histogram per mile.

4. Probability Density Function - The data for each modal coordinate are divided into 200 equal amplitude increments to cover the range from minimum to maximum amplitudes observed. The percentage of occurrences within each amplitude increment is then calculated and plotted to form a probability density function for that coordinate over the test run. In addition, the standard deviation, 95% level, 99% level and rms of the coordinate are calculated and printed out.

5. RMS Time History - The time history of each modal coordinate is Fourier transformed and a PSD calculated for each 4 seconds of time. The PSD is divided into octaves with center frequencies at 2, 4, 8, and 16 Hz, and the rms value calculated for each octave as well as the band from 0 to 30 Hz. These five values are then plotted as a function of time for the test run. The averaging time can be varied in increments of 4 seconds up to 16 seconds. In

addition, speed is plotted on an adjacent graph for ease of analysis.

RESULTS AND CONCLUSIONS

The instrumentation and data analysis techniques developed for this program have proved highly successful in the evaluation of the dynamic performance of lightweight and conventional flatcars. In particular the use of modal coordinates provided clear, concise engineering results which correlate well with physical phenomena.

Shown in Figure 8 is the effect of speed on the bounce of fully loaded flatcars. This result is representative of the acceleration environment in general; however, it should be kept in mind that bounce is only one of ten modes needed to completely describe the vibrational response of the carbody. This and other plots like it show conclusively that the lightweight and the conventional flatcars are quite comparable in performance. Furthermore, Figure 9 shows the load bounce versus speed, and serves to illustrate that this conclusion can be extended to the loads. This is of primary importance in considering the economic performance of rail vehicles. Finally, the magnitudes of elastic deformation for both types of cars were generally equal which indicates that the lightweight flatcars are as structurally sound as their conventional counterparts.

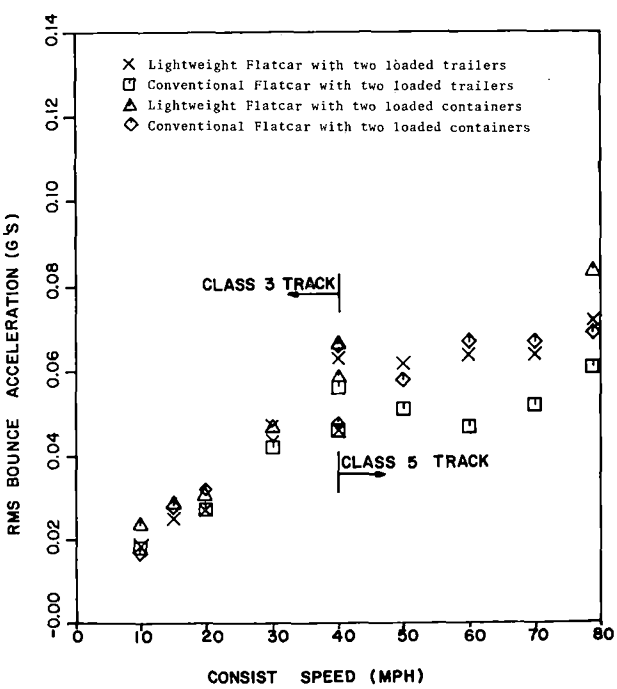

Figure 8. Fully Loaded Carbody Acceleration

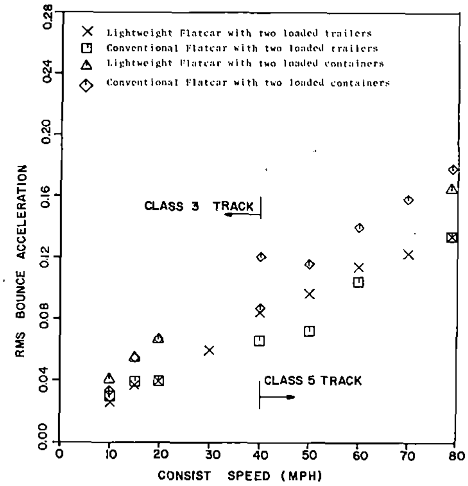

Figure 9. A-End Load Acceleration

Figure 10 shows the rms time history of the conventional carbody bounce mode. There are basically two points to note here. First, the first octave* with its center at 2 Hz is the largest contributor to the overall acceleration. This is anticipated based on considerations of carbody/load mass and the spring stiffness. Secondly, it is apparent that the acceleration level is speed dependent. In fact, Figures 8 and 9 show speed to be the most important single parameter influencing the acceleration level. This serves to emphasize the comparable performance of the lightweight and conventional flatcars.

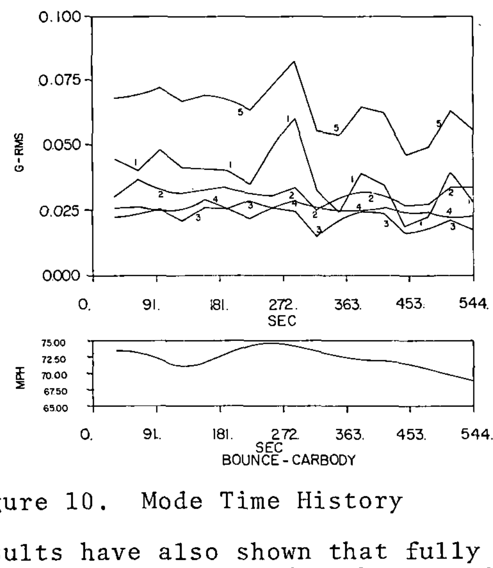

Figure 10. Mode Time History

Results have also shown that fully loaded flatcars provide a better ride performance than other load config-

* In the rms time history plots the curves of the octaves are numbered consecutively corresponding to center frequencies of 2, 4, 6 and 8 Hz. The fifth curve is the 0 to 30 Hz band rms value.

urations. In particular, it was found that the trucks of the conventional flatcar carrying a single load had a tendency to go into a lateral oscillation or hunting mode. This in turn had an extremely adverse effect on the acceleration levels in the loads.

Finally, data has been obtained which characterize the vibrational environment of containers and trailers during revenue operation. This data should prove beneficial to both the shippers and the railroads in evaluating the use of this mode of freight transportation.

Testing will continue during the upcoming year to provide similar data at specified points in the mileage accumulation of the test cars. This will allow a determination of the effects of component wear on the ride performance of these cars. This information is vital to engineers who will be designing and evaluating the future generation of intermodal rail cars.

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