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# Validation of FRATE, Freight Car Response Analysis and Test Evaluation

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# Validation of FRATE, Freight Car Response Analysis and Test Evaluation

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# ABSTRACT

A nonlinear computer program for Freight Car Response Analysis and Test Evaluation (FRATE) has been developed under the sponsorship of the Federal Railroad Administration. This computer program incorporates a model of a Trailer on Flatcar (TOFC) configuration. Validation of the FRATE/TOFC program is accomplished through comparison of analysis results to the results of vibration tests performed on a TOFC configuration at the Rail Dynamics Laboratory in Pueblo, Colorado. Validation criteria include resonant frequencies, deflection shape at resonance and amplification of input motions.

This paper presents a brief review of the FRATE/TOFC computer program, a summary of TOFC vibration test results and comparisons of test and analysis results. Procedures followed to achieve acceptable agreement between test and analysis are reviewed. Pre-and Post-validation versions of FRATE/TOFC are presented. Limitations of the validated program are reviewed.

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## 1.0 INTRODUCTION

In the development of any mathematical model it is advisable to demonstrate the degree of accuracy with which the model is able to duplicate its real life phenomenon. Validation of models is usually accomplished through a parallel effort of analysis and experiment with comparison of results to show the accuracy of the model. This paper presents the results of this kind of validation effort. The vehicle modeled is a railroad flatcar loaded with two highway trailers generally referred to as a trailer on flatcar or TOFC. Experimental data was acquired at the Federal Railroad Administration's Rail Dynamics Laboratory in Pueblo, Colorado.

The Federal Railroad Administration (FRA) and the Railroad Industry have been engaged in research and development activities related to the dynamics of trains and tracks. The main thrust of these activities have been to address those facets of "railroads" where changes as to dynamic characteristics would result in reduced maintenance costs, by virtue of reduced wear on track and railcar, and increased revenues by virtue of being able to safely carry more fragile commodities. The part of these activities which are of interest to us were a set of vibration tests performed on a TOFC configuration using the Vertical Shaker System at the Pueblo Rail Dynamics Laboratory. The objectives of this testing were to determine the vibration response characteristics of the TOFC configuration and to arrive at an optimum configuration, with respect to minimizing dynamic response, within the confines of practical parameter variations.

In conjunction with the TOFC vibration test the FRA contracted with Wyle Laboratories to develop a computer program for the dynamic analysis of freight cars. The program was to include a representation of the specific TOFC configuration and was to be used as a test aid. Subsequent to the testing the program was developed by MITRE into a general purpose computer program for freight cars and has been given the acronym FRATE, for Freight car Response Analysis and Test Evaluation. It uses a lumped parameter representation, includes certain non-linear properties and is solved in the time-domain by numerical approximation methods.

## 2.0 VALIDATION

### 2.1 Application of FRATE Defined

The use objective of the computer program FRATE is to predict the dynamic response of a freight car to track roughness where "dynamic response" includes loads, deflections and accelerations at representative points in and on the freight car. Where "track roughness" includes those periodic vertical and lateral track irregularities found with jointed rail and the impulse type irregularity encountered at crossings and switches. The freight car used is an empty or TOFC loaded flatcar but can be modified to other freight car configurations with truck component variations.

### 2.2 Direct Validation Impractical

Direct validation is defined here as demonstrating the accuracy of the dynamic response predictions through the performance of over-the-road tests and parallel analytical efforts and comparison of results. In order to be complete, comparisons should cover the full range of anticipated variables of both the track roughness and freight car configurations. Further, the program should be structured such that where test/analysis correlations are not acceptable the information acquired will provide clues to the changes needed to achieve improved correlation. Practical considerations, time and cost in particular, make sucn an approach impractical.

### 2.3 Validation Through Dynamic Characteristics Comparison

The accepted practice for validation of any dynamic system has been to show that certain dynamic characteristics are accurately reproduced by the analytical program, it being found that the general response prediction capabilities will be of equal accuracy. These dynamic characteristics are resonant frequencies, deflection shapes at resonance and the ratio of response to input motions at resonance.

Thus the validation effort consists of defining these dynamic characteristics experimentally and analytically and showing that the analysis is an accurate reproduction. A significant result of this procedure is that where results are different the differences supply clues to changes which would achieve improvement. For example, stiffness affects resonant frequency by the one-half power: consequently if a resonance frequency is off it is probable that some stiffness needs to be changed. If deflection shapes at resonance are not the same an examination of the differences in shape will provide clues as to which parts are too stiff or, too soft or are too heavy or too light. Finally, comparison of the ratio of response to input motions will indicate whether the damping values used are too high or too low.

The validation process used for FRATE is one in which resonant conditions are first defined from the test data - frequencies, amplitudes and deflection shapes of the entire physical structure - and then the analysis is made to reproduce these resonant conditions as closely as the project budget and schedule allow. The process used depends on the skill and engineering judgement of the analyst since he makes the comparisons between test and analysis. If in his opinion parameter values (e.g., inertias, spring rates or damping values) need adjusting to produce a closer match between test and analysis he makes them on a sequential and/or parallel computer run basis. If in his opinion the model's topology (i.e., the equations of motion) requires modification he will do so in a manner similar to the parameter adjustment process. Throughout, the physical behavior of the model is constantly being monitored by the analyst. He must repeatedly judge if the model is reasonable and consistent and thereby provides a very important governor on the validation process.

## 2.4 Estimation Procedures, Basis For Not Using

Prior to the start of the validation effort consideration was given to the possible use of some form of System Identification, Parameter Identification or Parameter Estimation Procedure. These are procedures where the math model is related to experimental results through some mathematical relationship or procedure. That is, the experimental results are used directly to define the model or to modify prescribed model parameters. These procedures have the advantage of taking the guesswork out of model changes. They have the disadvantage of becoming computationally impractical for large or nonlinear systems. There were three overriding considerations which ruled out the use of any of these procedures. First, the planned applications of FRATE are in the study of freight cars insofar as the effects of configurational and/or design changes on dynamic response characteristics. Consequently paramount importance was placed on maintaining clear visibility of physical/mathematical relationships of the simulation. Second, development of the FRATE computer program and performance of the TOFC vibration testing preceeded any formal validation planning. Validation consequently was a matter of using available data rather than tailoring the data to accommodate the validation effort. Third, automated identification/estimation procedures are invariably built around frequency domain solutions. It was felt that development of procedures for a time domain solution would require a major development effort with no assurance of success within reasonable time and effort limits.

## 2.5 Practical Consideration

A final thought on validation has to do with the degree of validation achievable in a practical effort and on the other hand the degree of accuracy needed in a valid analysis tool being validated. As with anything physical, there is no sharp definable line between good and no-good in model validation. It should be understood that there are variabilities within any given freight car configuration and in fact any given freight car will have variations in dynamic behavior from day to day: that the test data which is being used to validate a model will probably be incomplete and will certainly have inaccuracies; and that all other things being optimal the model and analysis method will have inherent inaccuracies. Consequently validation criteria should conform with the level of accuracy of the comparison standard (i.e., the test data). Further, the application of the validated model must also be in cognition of the accuracy of validation.

A validated analysis then will not provide us with exact answers, indeed in the applications context there is no such thing. The validated analysis will provide us with a valuable tool which can be used in the study of freight car dynamic behavior and in the study of optimizing design changes. Further no matter how accurate the analyses method may be, final experimental verification is mandatory. The value of the analyses is that it can be used to largely replace development testing and actually providing more information, both as to configurations studied and measurements, in less time and at less cost.

## 3.0 MODEL DESCRIPTION

The freight car model of this report is a TOFC (Trailer on Flatcar) configuration evolved by Wyle Laboratories from work done by Healy, Reference 1 and Ahlbeck, et al, Reference 2. It consists of the lumped mass simulation shown schematically in Figures 1 and 2. Figure 1 defines the notation used for masses, inertias and degrees of freedom. Figure 2 defines spring damper notation. The railcar is represented by three rigid body masses: two trucks and the carbody. Carbody flexibility is included through application of a component normal mode technique using the first seven free-free normal modes. Each of the two highway trailers are represented by two rigid masses: one for the trailer body and one for the wheel/axle assembly.

### 3.1 Truck Model

The simple, one mass, truck representation was justified by Healy on two basic counts: (1) the intended use of the model was for analysis of overall railcar dynamic response characteristics for which the one mass truck simulation is adequate, (comparisons to experimental results shown in Reference 1 support this statement), (2) the one mass truck simulation has computer cost advantages in that the size of the model is minimized and the higher frequencies and attendent small integration time step of a more complex truck model are avoided.

Spring/dampers 1, 2, 3 and 7, 8, 9 represent the vertical and lateral stiffness/damping of the track, wheels, bearings and side frames. Spring/dampers 4, 5, 6 and 10, 11, 12 represent the vertical and lateral stiffness and damping of the spring nests, bolster and centerplate.

Ahlbeck, in Reference 2 and shown here in Table I, notes that the lateral stiffness and damping of the track will vary, in orders of magnitude, in going from the flange-contact to no-flange contact conditions.

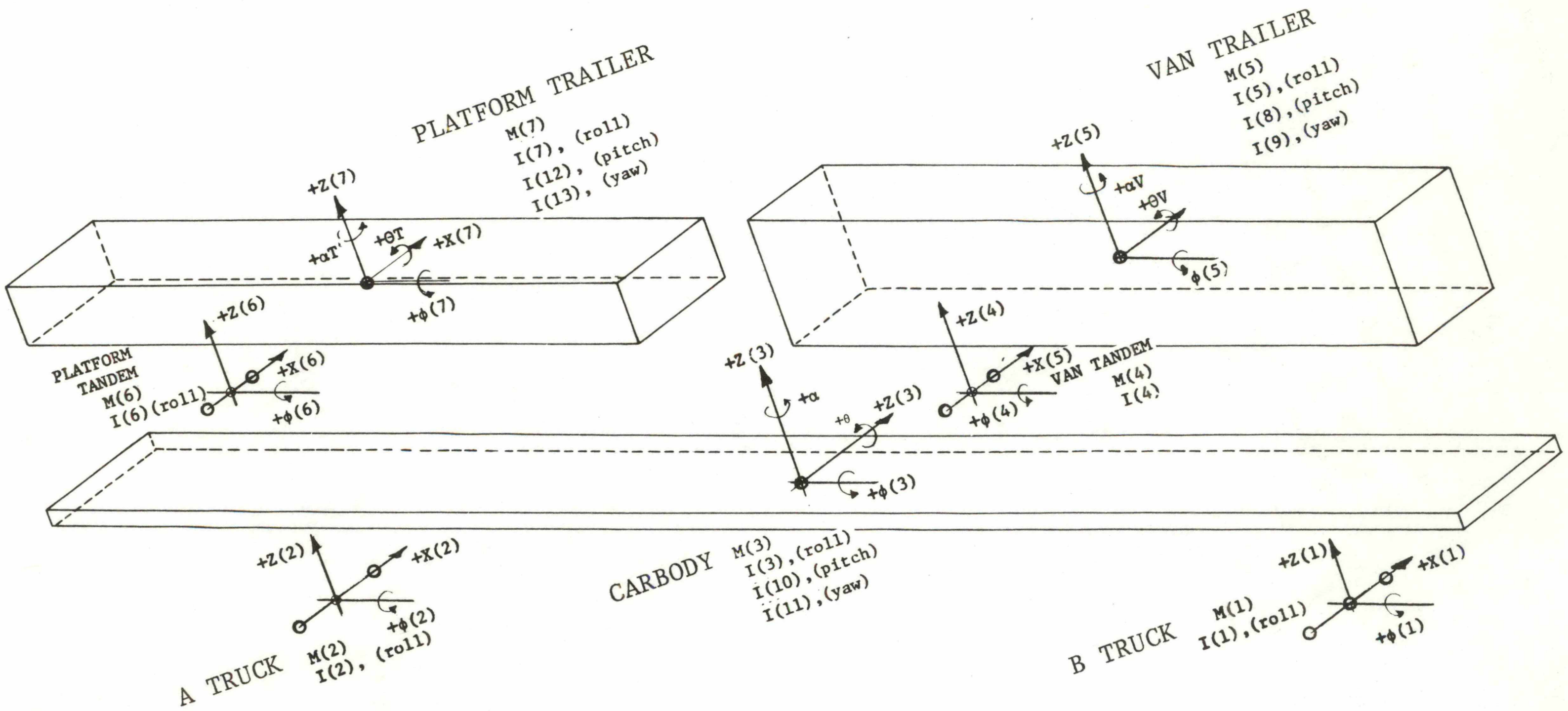
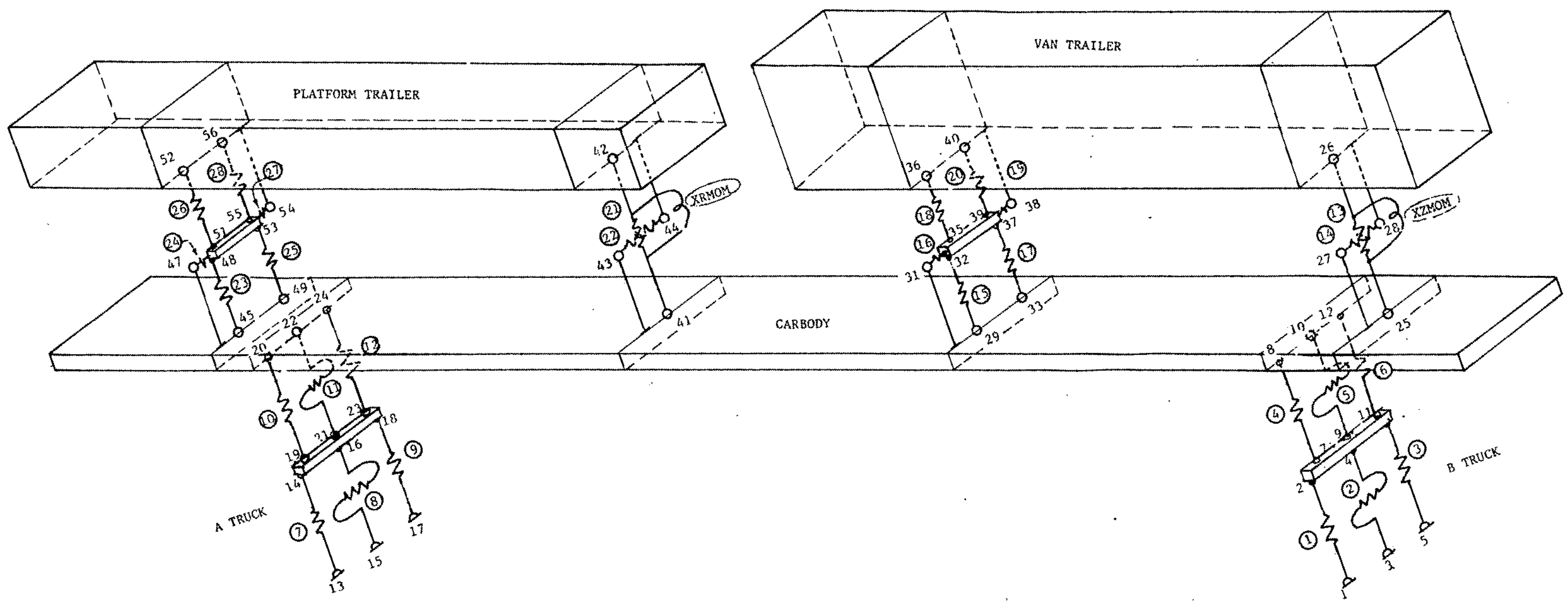



**FIGURE 1**
**FRATE GEOMETRY**




SPRING AND DAMPER NOTATION

LINEAL SPRING AND DAMPER

ANGULAR SPRING AND DAMPER

CIRCLED NUMBERS - SPRING AND DAMPER

UNCIRCLED NUMBERS - SPRING AND DAMPER ATTACHMENT POINTS

**FIGURE 2**
**FRATE SPRING/DAMPER NOTATION**

With flange contact the numbers represent the lateral stiffness and damping afforded by the rails and wheels. With no-flange contact the numbers are actually a representation of the hunting characteristics of the truck at sub-critical speeds.

TABLE I

TRACK LATERAL IMPEDANCE
(Reference 2)

| CONDITION | STIFFNESS (KL) lb/in | DAMPING (CL) lb sec/in |
|---|---|---|
| Flange-Contact | 200,000 | 500 |
| No-Flange-Contact | 2,228 | 12,570 |

The flange-contact values are used here because they are more representative of laboratory test conditions against which analysis results will be compared. That is, the lateral load transfer between wheel and track without wheel rolling should be the same as with flange contact. If a rolling condition is to be simulated the no-flange-contact spring/damper values should be used, or if large amplitudes of side-to-side motions are to be studied a bi-linear spring/damper can be used.

## 3.2 Trailer Models

Each trailer is modeled by two lumped masses, one for the trailer body and the other for the wheel and axle assembly. The forward support, which is the trailer hitch, is represented by three spring/dampers: vertical, lateral and angular in the roll direction. The aft support is made up of two vertical and one lateral spring/dampers representing the tandem tires. A second set of two vertical and one lateral spring/dampers represent the tandem springs.

A series of vibration tests were performed on a loaded van trailer at the Rail Dynamics Laboratory in Pueblo, Colorado. Results of this testing were analyzed and used as a basis for evaluating the Wyle trailer simulation. Conclusions reached were that the model was too stiff with respect to both the hitch springs and the tandem springs, and that the trailer body was too flexible to be assumed rigid.

### 3.3 Carbody Flexibility

The flexibility of the carbody is included in the problem solution through the use of normal mode methods and the use of a superposition technique referred to by Levy, Reference 3, as the Component Element method. Developments and applications of the methods can be found in the literature, for example, References 3, 4 and 5.

Normal modes of vibration of the carbody were obtained by Wyle Laboratories. The discussion here is paraphrased from an unpublished Wyle report in which the carbody structure and mass only, with free-free boundary conditions were analyzed. The highway trailers and the rail car trucks were not included.

The vibration analysis was performed using ANSYS, a commercially available structural analysis program using finite element methods. The geometry of the flatcar was defined by a number of node points which described the endpoints of the structural elements making up the flatcar body. The section properties of these structural elements were calculated from flatcar structural member counterparts.

The carbody model was then reduced to an eleven mass, 26 degree of freedom (DOF) simulation using the Guyan Reduction method. The undamped natural frequencies and deflection shapes for the first seven flexible modes were obtained with ANSYS. Deflection shapes were normalized to obtain unity generalized mass. With this information each "normal" mode was included in the time domain problem as a separate equation-of-motion and treated as another lumped mass.

### 3.4 Description of FRATE

The analysis method used, generally known as the Newtonian method, requires an equation-of-motion for each mass in each of its degrees of freedom. This includes the generalized modal masses discussed previously. A time domain solution is obtained through a numerical integration procedure.

FRATE consists essentially of a time do-loop with an input section before it and an output section after it. A flow diagram of the program at its most fundamental level is shown in Figure 3. The input section reads the input, calculates the 1 g static deflections and initializes constants. In the time do-loop the equations-of-motion are solved approximately using a numerical recursive algorithm based on the Runge-Kutta method of solving different equations. The output section stores the response data in the various forms specified.

The vertical and lateral displacements and velocities which are induced by rotational motions, included in the present version of FRATE, are listed in Table II. There are no pitch and yaw motions of the truck and trailer suspension masses, therefore there are no induced vertical or lateral displacements or velocities due to them. The L/2 coefficient in each term is included to illustrate the form of the term and is not necessarily the one in the program. Note that there are two terms associated with the vertical displacement of the flatcar due to roll motion. The second term - $(H/2)(1 - \cos \phi_3)$ - is usually, but not always, small with respect to the first term $L/2(\sin \phi)$. Users of FRATE are strongly advised to develop their own tables of induced motions, as shown here. The user can choose the amount of this motion coupling by inclusion or exclusion of such terms in the equations of motion.

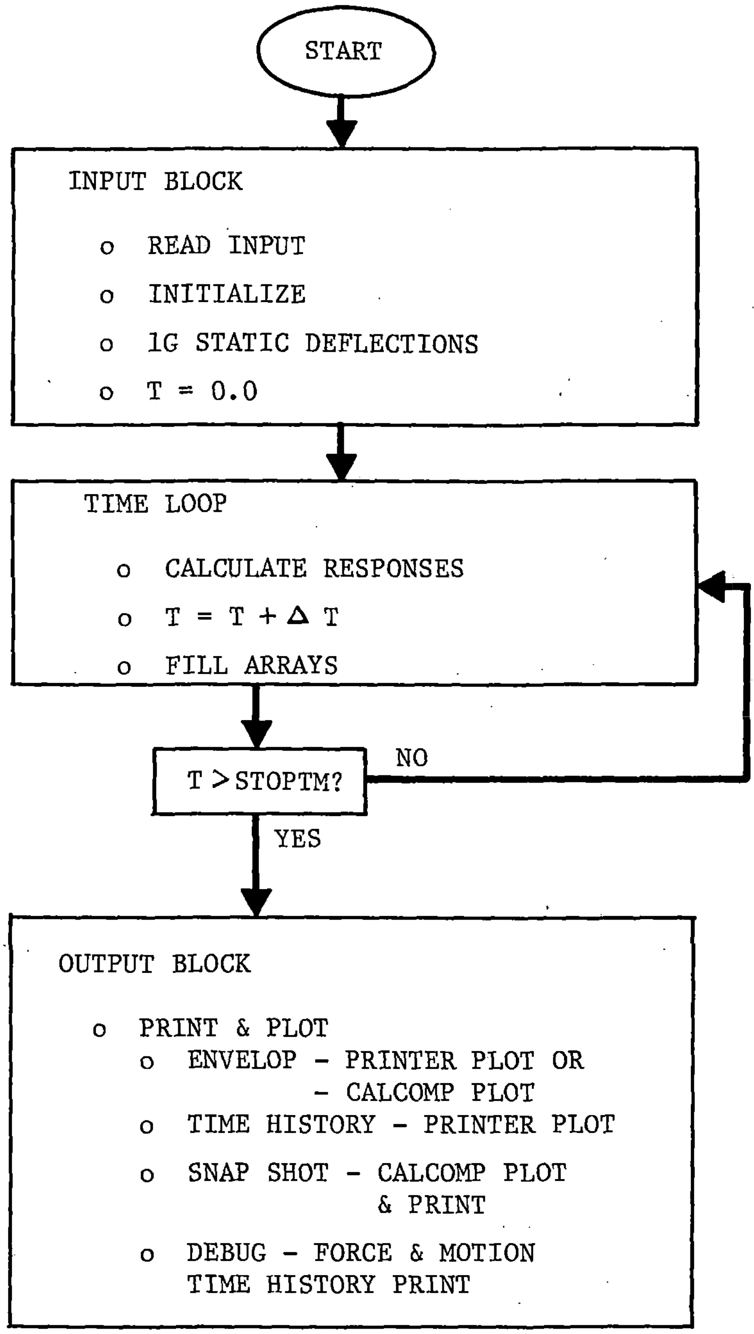



**FIGURE 3**
**FRATE PROGRAM - LOGIC FLOW**

TABLE II

VERTICAL AND LATERAL MOTIONS DUE TO ROTATION

| DESCRIPTION | TRUCKS AND TRAILER SUSPENSION MASSES | FLATCAR MASS | TRAILER MASSES |
|---|---|---|---|
| Vertical Displacement Due to Roll | $R/2\ \mathrm{SIN}\phi$ | $R/2\ \mathrm{SIN}\phi$, $H/2(1-\cos\phi)$ | $R/2\ \mathrm{SIN}\phi$, $H/2(1-\cos\phi)$ |
| Vertical Velocity Due to Roll | $R/2\ (\dot{\phi})\cos\phi$ | $R/2\ (\dot{\phi})\cos\phi$, $H/2(\dot{\phi})\mathrm{SIN}\phi$ | $R/2\ (\dot{\phi})\cos\phi$, $H/2(\dot{\phi})\mathrm{SIN}\phi$ |
| Vertical Displacement Due to Pitch | - | $L/2\ \mathrm{SIN}\theta$, $H/2(1-\cos\theta)$ | $L/2\ \mathrm{SIN}\theta$, $H/2(1-\cos\theta)$ |
| Vertical Velocity Due to Pitch | - | $L/2\ (\dot{\theta})\cos\theta$, $H/2(\dot{\theta})\mathrm{SIN}\theta$ | $L/2\ (\dot{\theta})\cos\theta$, $H/2(\dot{\theta})\mathrm{SIN}\theta$ |
| Lateral Displacement Due to Yaw | - | $L/2\ \mathrm{SIN}\alpha$ | $L/2\ \mathrm{SIN}\alpha$ |
| Lateral Velocity Due to Yaw | - | $L/2\ (\dot{\alpha})\cos\alpha$ | $L/2\ (\dot{\alpha})\cos\alpha$ |
| Lateral Displacement Due to Roll | - | $H/2\ \mathrm{SIN}\phi$ | $H/2\ \mathrm{SIN}\phi$ |
| Lateral Velocity Due to Roll | - | $H/2\ (\dot{\phi})\cos\phi$ | $H/2\ (\dot{\phi})\cos\phi$ |

The forcing function in FRATE is presently set up to impose sinusoidal motion at any of the six node points at the wheel-rail interface (node points 1, 3, 5, 13, 15 and 17 shown on Figure 2). The amplitude and phase angle of each input can be set at any relative value for a run. The reference amplitude of the sinusoidal input motion can be specified as a displacement, velocity or acceleration (in g's). All six input motions will be at the same frequency. The input frequency can be held constant, varied linearly or logarithically or be imposed for a limited number of cycles in order to study the decay characteristics.

## 3.5 FRATE Output Options

The major contribution made by MITRE to the program FRATE has been in the development of output options. Acknowledgement is made of the work of John F. Caskey of MITRE in the programming of these output options.

The direct output of FRATE is a listing of the forces in the 28 spring/dampers and the accelerations, velocities and displacements of the 27 degrees of freedom.

However, the most useful output are response motions of the 30 locations shown in Figure 4. The X and Z direction responses in acceleration and displacement are determined for each point and can be output in three different forms: (1) envelope, (2) time history and (3) deflection shape, or SNAPSHOT.

The envelope option is used where a frequency sweep is performed to show the variation of the peak values with frequency variation. The program identifies positive peak values and stores for a selected interval of time steps. The data is plotted as a function of frequency causing the time domain analysis results to appear as frequency domain

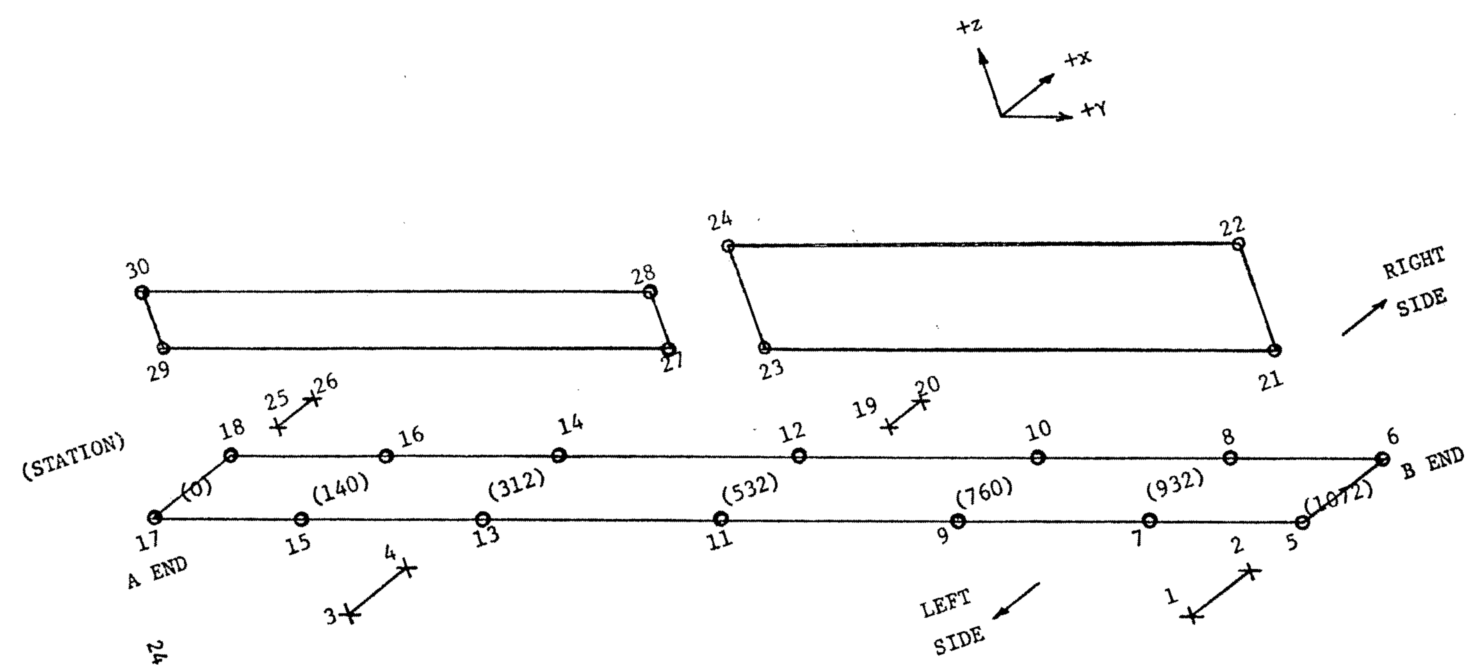



FIGURE 4
FRATE/TOFC OUTPUT DATA NUMBERING SYSTEM

transfer functions. The data can be obtained in plotted form either on printer plots or calcomp plots. A sample CALCOMP plot is shown in Figure 5. The intended application of envelope plots is to identify resonant frequencies and to obtain an estimate of effective damping at resonance.

Time history plots can be obtained on printer plots for a selected number of response points. The plots are superimposed in groups of three and four to facilitate comparison of responses and to minimize plotting costs. An example time history output is shown in Figure 6. The intended application of time history plots is to obtain a better estimate on resonant frequencies primarily through the analysis of response phase shifting. Figure 6 shows a decay time history following an 8.40 Hertz roll excitation.

The deflection shape, or SNAPSHOT, plots define the deflection shapes. With SNAPSHOT it is possible to obtain several pictures of the deflection shape in the period of one cycle of oscillation. This feature is especially useful because with the non-linear and highly damped aspects of rail cars, responses are usually found to be in the complex mode as having the character of a "traveling wave" across the structure (as opposed to the usual "standing wave" produced by a normal mode). Experience with FRATE/TOFC to date has shown the deflection shapes to have varying degrees of this traveling wave characteristic. Series of SNAPSHOTS for several resonances are presented later in this section which illustrate this effect.

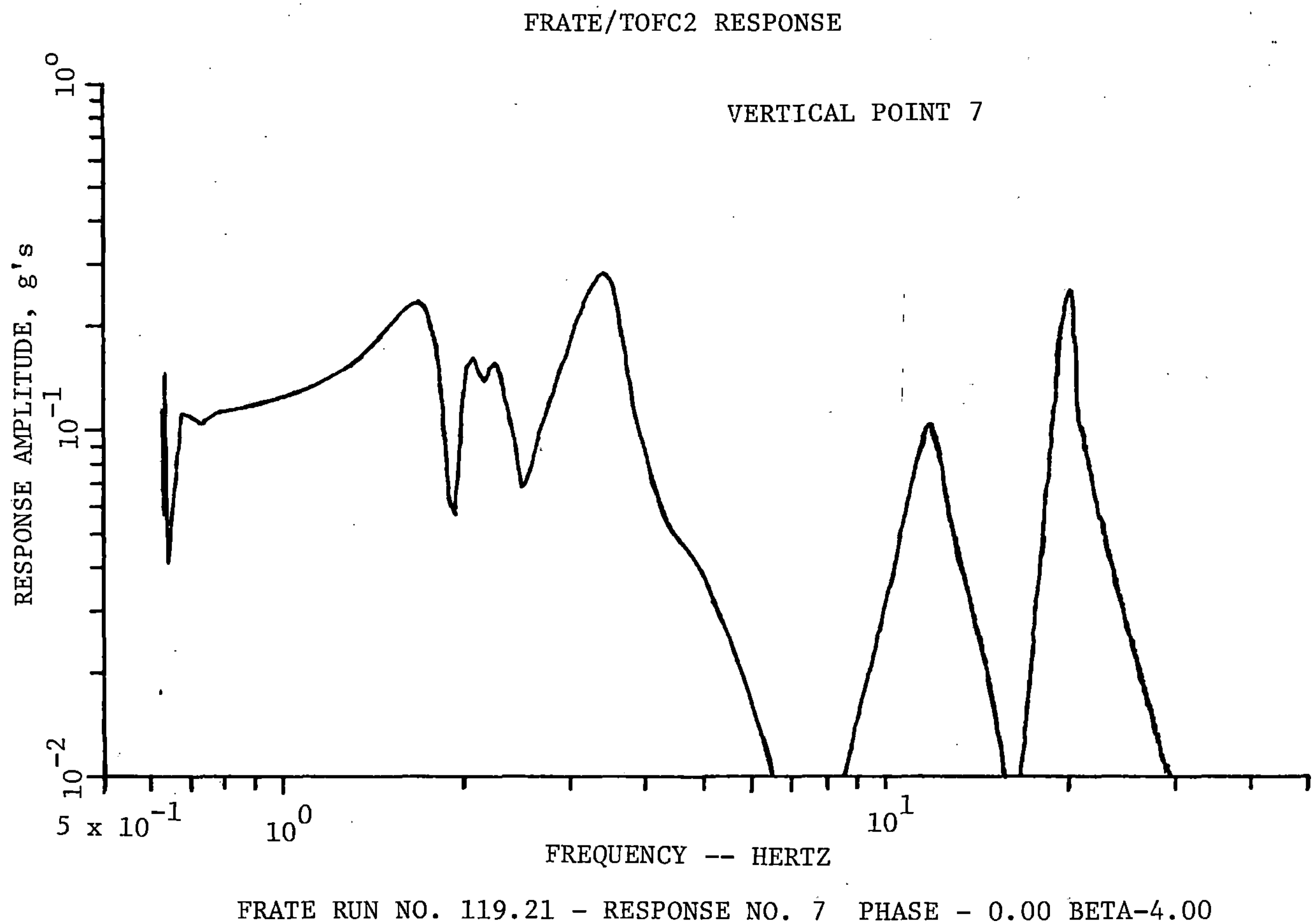



**FIGURE 5**
**SAMPLE CALCOMP PLOT OF ENVELOPE DATA**

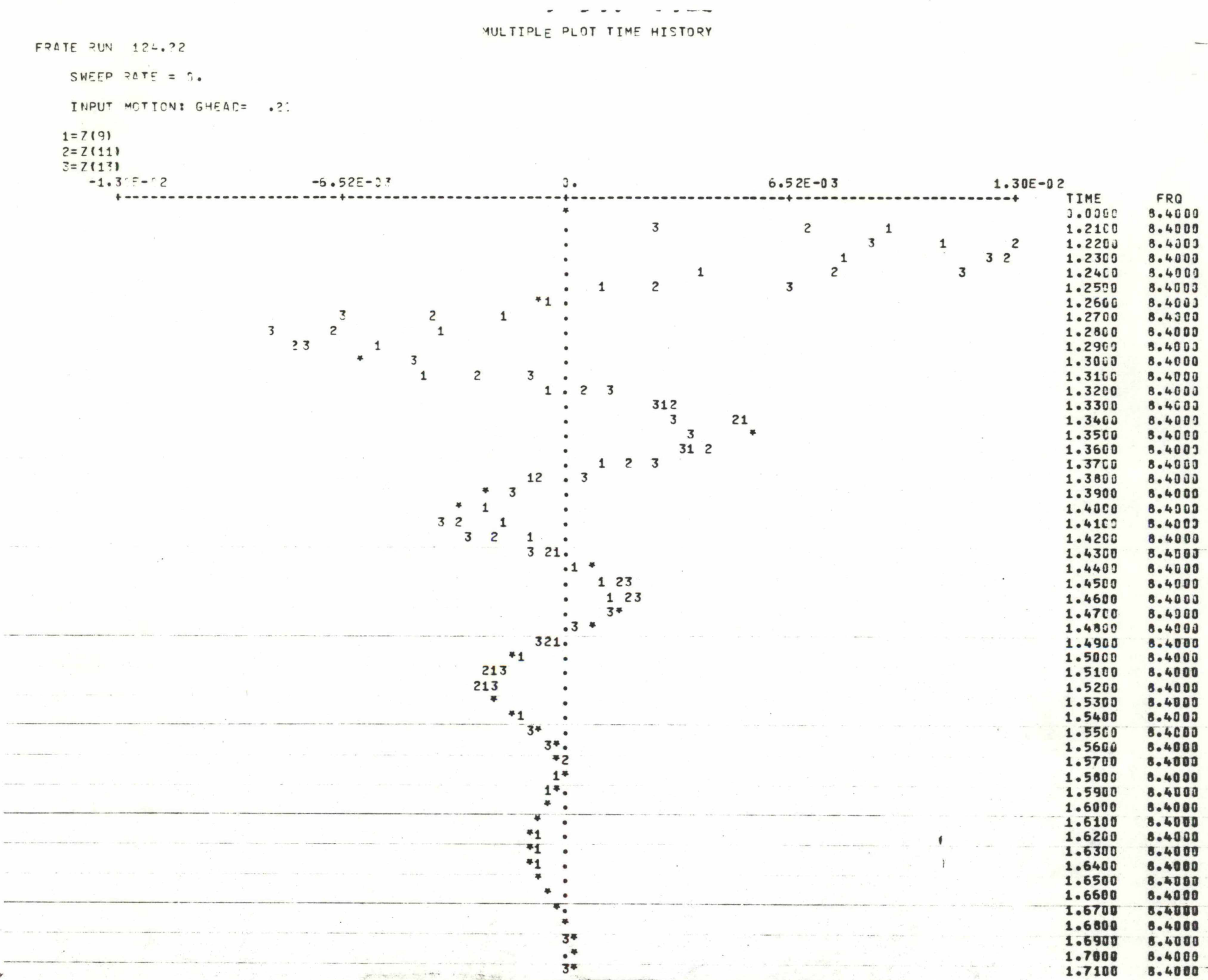



**FIGURE 6**
**SAMPLE TIME HISTORY PRINTER PLOT, SHOWING DECAYING OSCILLATIONS FOLLOWING CUT OFF OF 8.4 HERTZ FORCE INPUT**

## 4.0 FRATE ANALYSIS

Analyses which have been performed first on FRATE were with the TOFC model as generated by Wyle Laboratories, Reference 2, and modified by MITRE. The purposes for performing these analyses were first to verify the working conditions of FRATE as modified by MITRE and second to define a baseline TOFC model. The baseline TOFC model is to provide initial comparisons to test results before modifying the model to improve comparisons to the point of achieving validation.

The analyses performed were intended to duplicate the tests performed on a TOFC configuration at the Rail Dynamic Laboratory (RDL) Vertical Shaker System (VSS). The facility and test techniques are discussed in References 6 and 7. In these tests excitation was sinusoidal, applied at the B truck with four thrusters, one under each wheel. The thruster forces were either all in phase with each other, or the thrusters on one side were at $\pi$ phase with the thrusters on the other side. The in-phase motions were intended to excite vertical and pitch resonances. The $\pi$-phase motions were intended to excite roll and carbody torsion resonances. The general test procedure was to perform sinusoidal sweep tests, varying the frequency of excitation at a prescribed rate and through a prescribed frequency range, sweeping both with increasing frequencies (up) and decreasing frequencies (down). Resonant frequencies and amplification factors were determined. Attempts were made to obtain more accurate measurements through the performance of fixed frequency (dwell) and transient (decay) tests. These were not always successful because non-linear resonance characteristics vary with fixed and changing frequencies. Modifications were then made to the TOFC configuration to find an "optimum"; i.e., one where the response amplitudes were the smallest.

There were, in the course of the test, many configurations tested. For the purpose of validating the FRATE/TOFC model there were two configurations of primary interest. The empty flatcar with no snubbers and the fully loaded TOFC configuration with no snubbers. Snubbers are friction damping devices that are an integral part of the standard truck suspension system. There are four snubbers per truck. The most reasonable approach to the validation task was to show correlations to the simplest case first and then step up in complexity until all desired configurations have been validated. The undamped empty flatcar is the obvious starting configuration and the loaded TOFC with no snubbers the second.

## 4.1 Analysis Description

The analysis procedure followed was essentially the same as the test procedure. Excitation was applied to the B trucks; up and down sinusoidal frequency sweeps were performed to identify resonances; narrow band sweeps, dwells and decays were performed to characterize each resonance.

Resonance identification was made on the bases of maximum response and quadrature ($\pi/2$) phase relationship between response and input motions. In some cases resonance identification was relatively clear. There were also complex mode resonances where the response vs. frequency curve was broad and muted and the phase angles were not at quadrature. In these cases resonance identification was based largely on the response curves and could not be done with the same degree of confidence.

## 4.2 Analysis Results

The resonant frequencies determined by FRATE analysis of the original empty flatcar and TOFC models are listed in Tables III and IV. Deflection shapes for the TOFC resonances are shown in Figures 7 through 15.

The deflection shapes are presented as a sequence of shapes through approximately one cycle of motion. This was necessary because the resonances generally did not have fixed nodal patterns, associated with normal modes. They had, to greater or lesser degree, the moving nodal patterns generally associated with complex modes. Figure 6 is a good illustration of this. If the node point of the carbody deflection is followed in each time sequence, it is seen to start at the right hand end of the picture and progress across to the left in about a half a cycle. This characteristic is attributed to two factors: the excitation being at one end of a long body, and the relatively large amount of damping in the system.

The first three vertical resonances are seen to be varying combinations of carbody pitch, carbody bending and trailer pitch. The fourth and fifth vertical resonance are obviously second and third carbody bending.

The first three roll resonances are all carbody rigid body roll. In the first the vehicle is rolling about a center of rotation below the carbody center of gravity. In the second resonance the center of rotation is above the c.g. of the carbody. In the third resonance the carbody and trailers are in roll motions in opposition to each other. The fourth resonance is carbody torsion.

The empty carbody deflection shapes are discussed later.

TABLE III

RESONANT FREQUENCIES FRATE ANALYSIS
BEFORE VALIDATION

Vertical Mode Frequencies

TRAILERS ON FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| COUPLED PITCH BOUNCE AND CARBODY BENDING | 1.77 |
| SAME PLUS PLATFORM TRAILER PITCHING | 2.2 |
| SAME PLUS VAN TRAILER PITCHING | 3.3 |
| CARBODY 2ND BENDING | 12.0 |
| CARBODY 3RD BENDING | 19.65 |

EMPTY FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| VERTICAL BOUNCE AND BENDING | 3.6 |
| PITCH | 6.7 |
| FIRST CARBODY BENDING | 8.1 |
| SECOND CARBODY BENDING | 13.0 |

## TABLE IV

### RESONANT FREQUENCIES FRATE ANALYSIS BEFORE VALIDATION

### Roll Mode Frequencies

TRAILERS ON FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| CARBODY ROLL - LOW CENTER OF ROTATION | .87 |
| CARBODY ROLL - HIGH CENTER OF ROTATION | 3.5 |
| COUPLED ROLL, TRANSLATION AND CARBODY TORSION | 8.7 |
| CARBODY 1ST TORSION | 14.3 |

EMPTY FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| CARBODY ROLL - LOW CENTER OF ROTATION | 3.80 |
| CARBODY ROLL - HIGH CENTER OF ROTATION | 5.32 |
| FIRST TORSION | 13.3 |

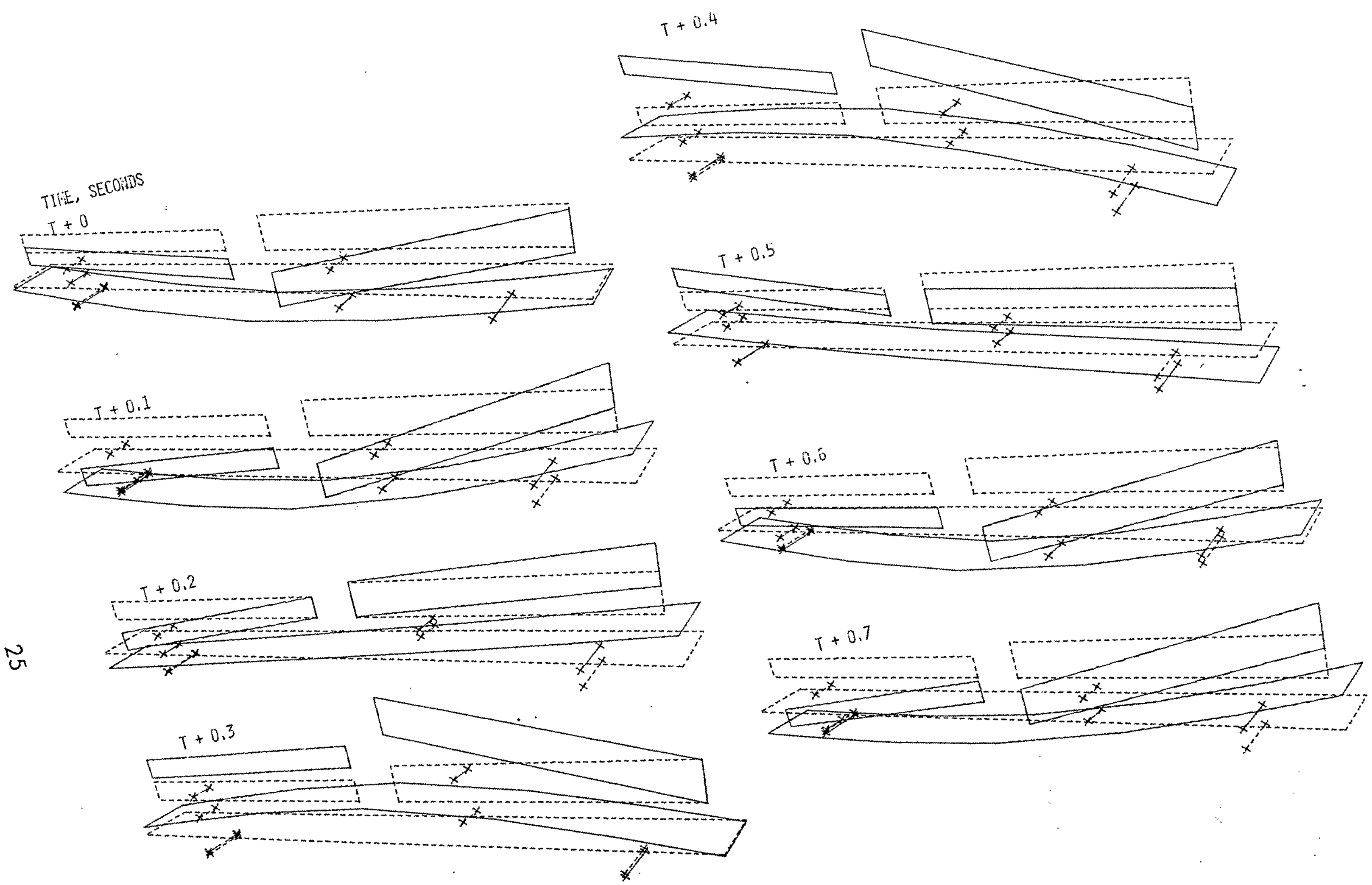



**FIGURE 7**
**FRATE/TOFC SNAPSHOT 1.77 HERTZ VERTICAL RESONANCE**

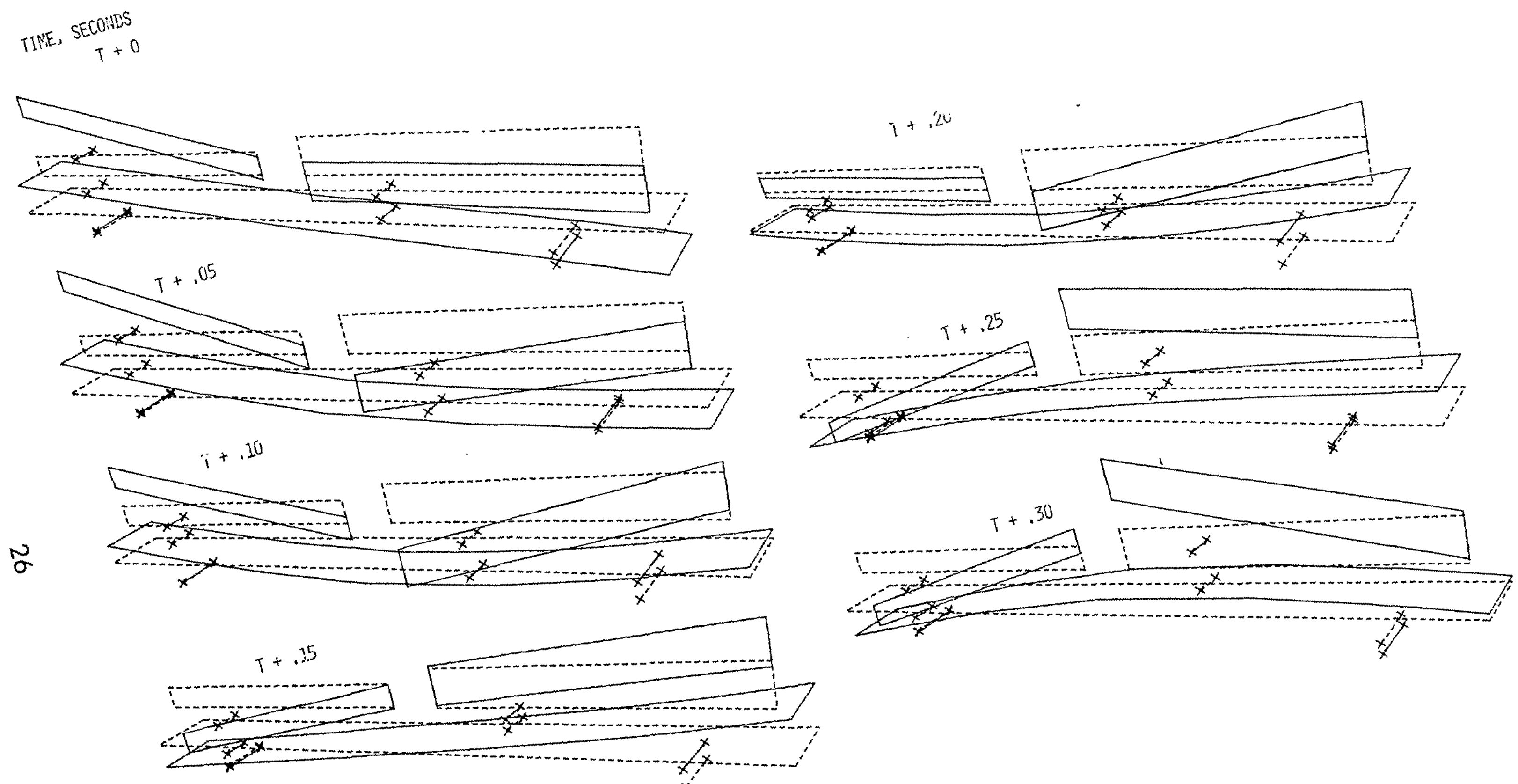



FIGURE 8
FRATE/TOFC SNAPSHOT 2.22 HERTZ VERTICAL RESONANCE

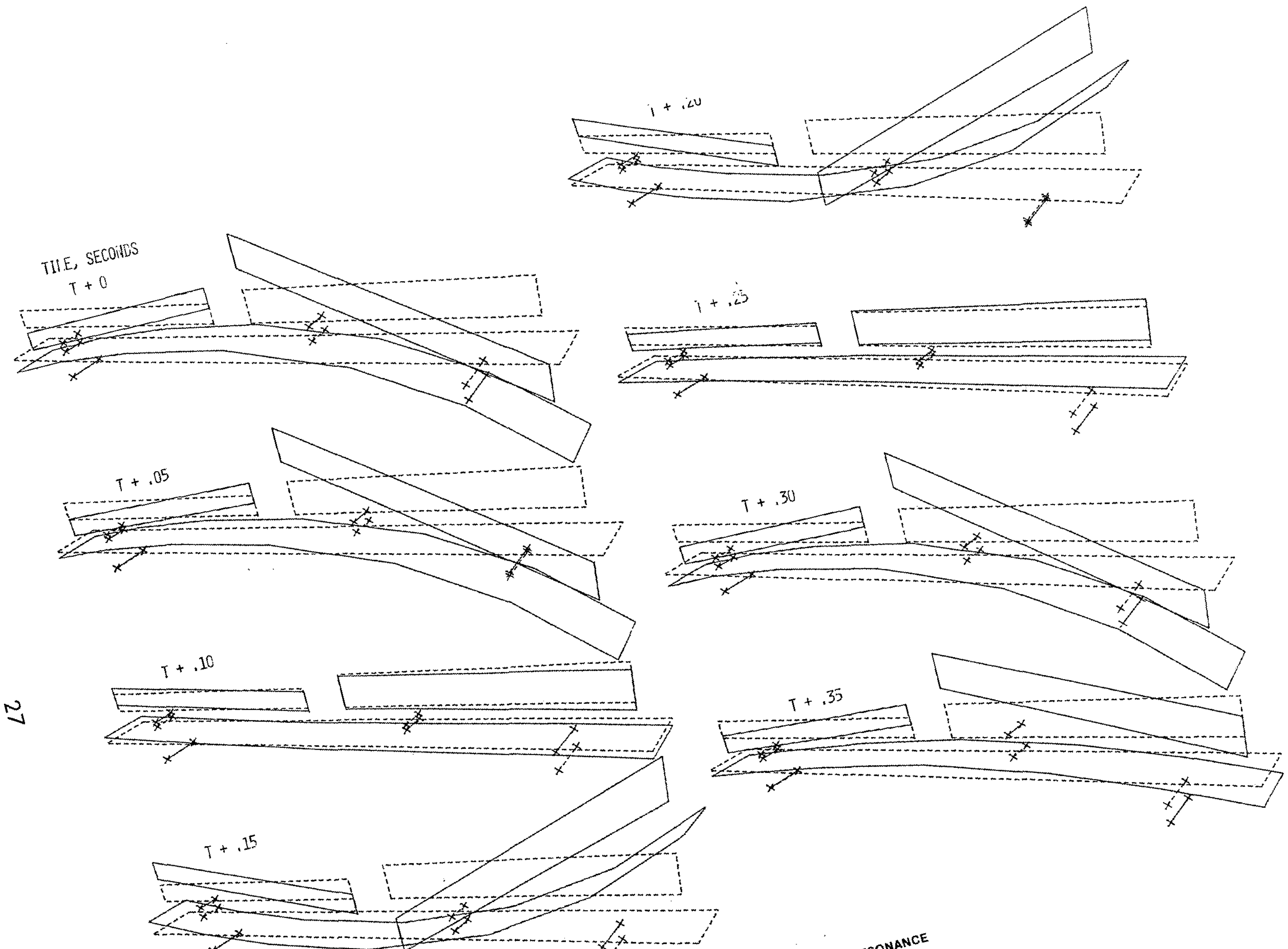



FIGURE 9
FRATE/TOFC SNAPSHOT 3.33 HERTZ VERTICAL RESONANCE

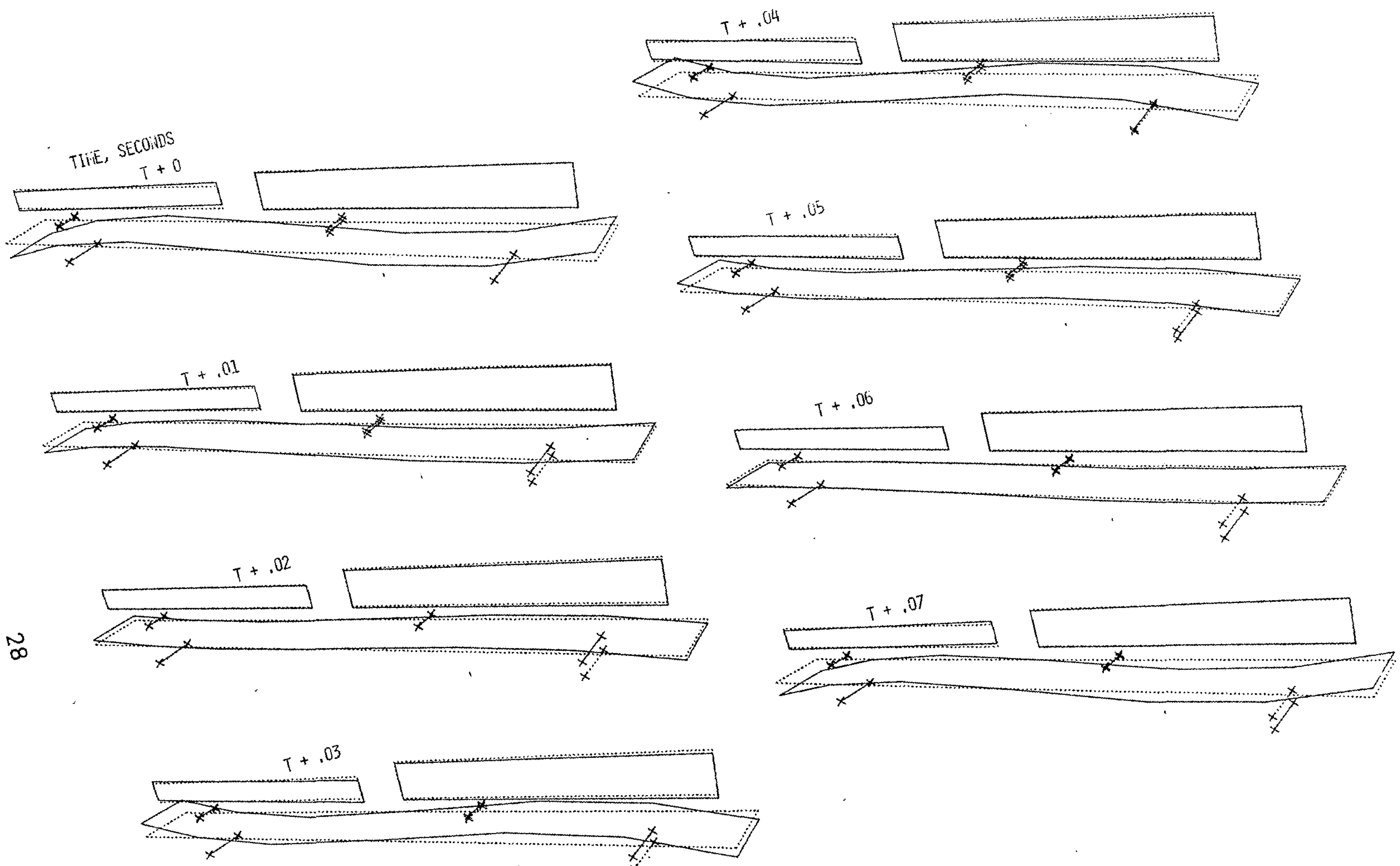



FIGURE 10
FRATE/TOFC SNAPSHOT 12.0 HERTZ VERTICAL RESONANCE

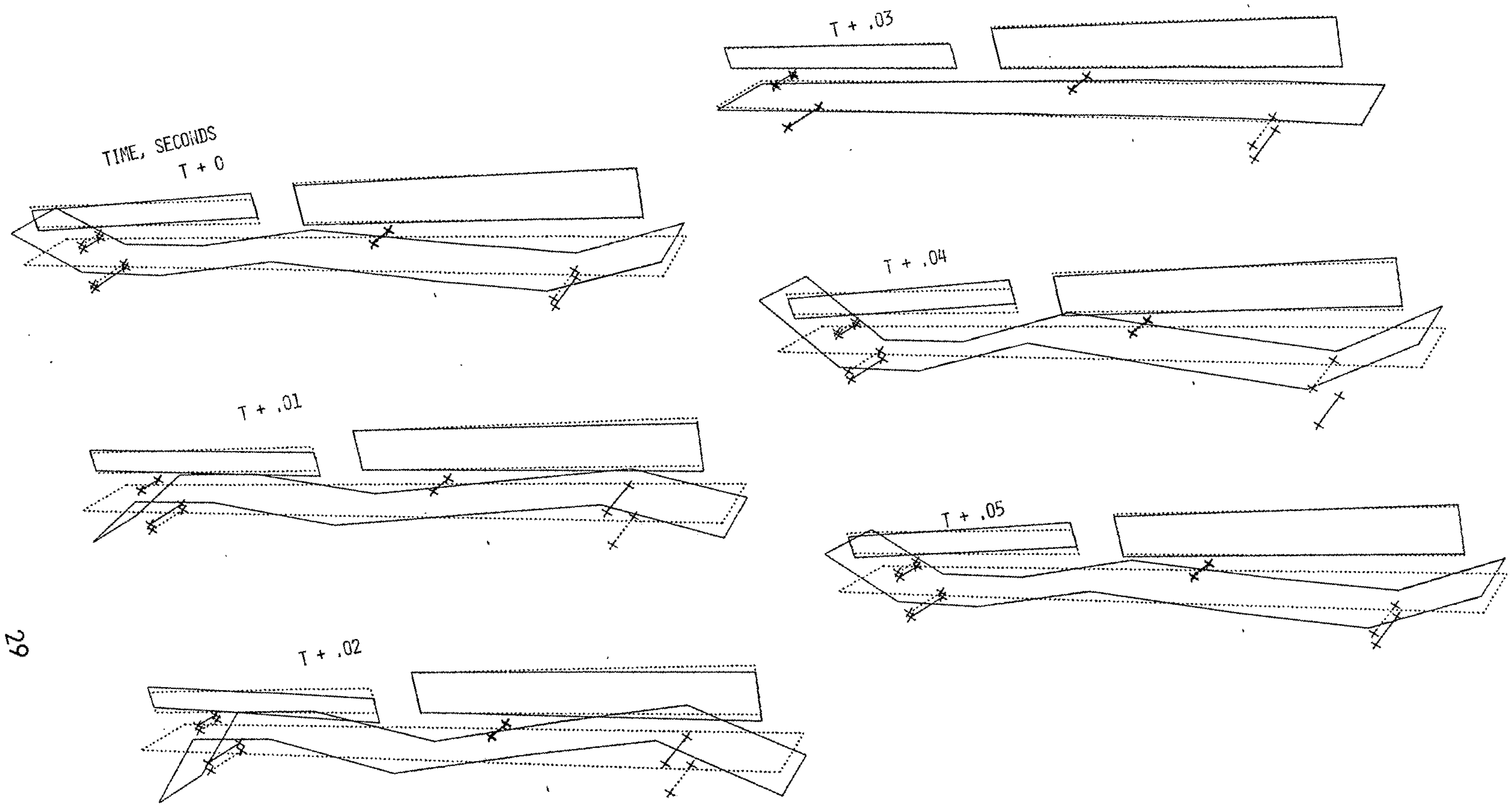



FIGURE 11
FRATE/TOFC SNAPSHOT 19.65 HERTZ VERTICAL RESONANCE

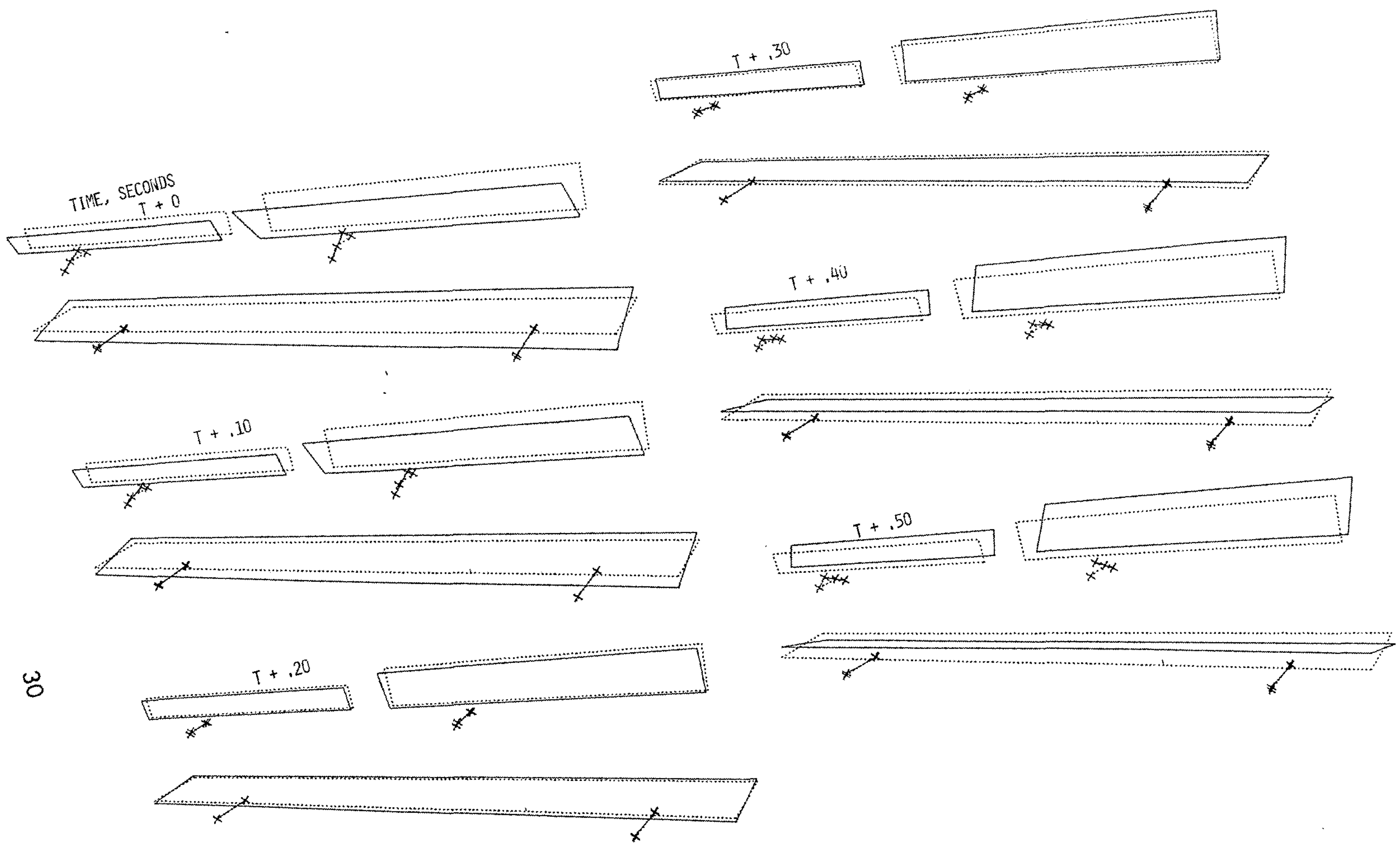



FIGURE 12
FRATE/TOFC SNAPSHOT 0.87 HERTZ ROLL RESONANCE

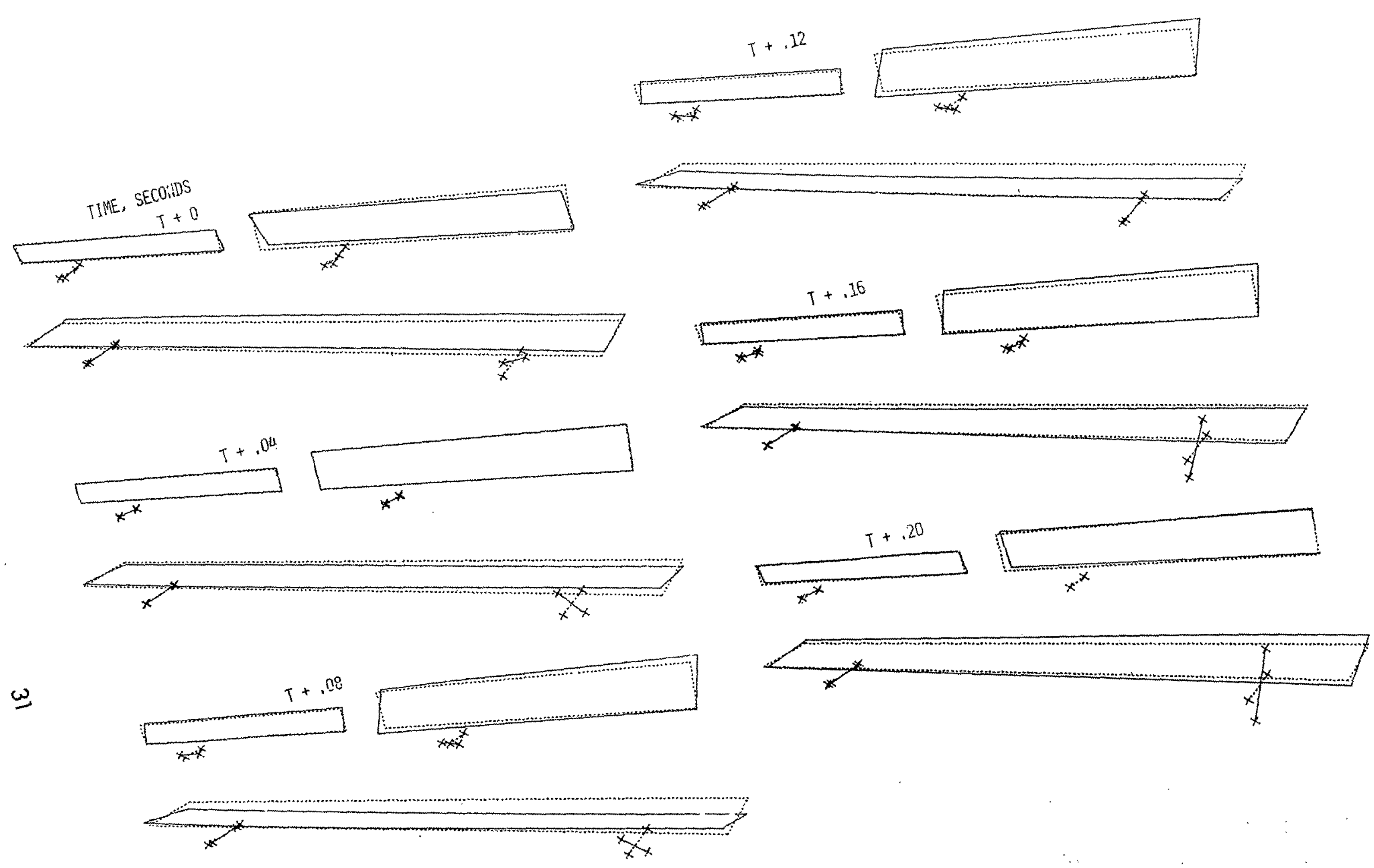



FIGURE 13
FRATE/TOFC SNAPSHOT 3.5 HERTZ ROLL RESONANCE

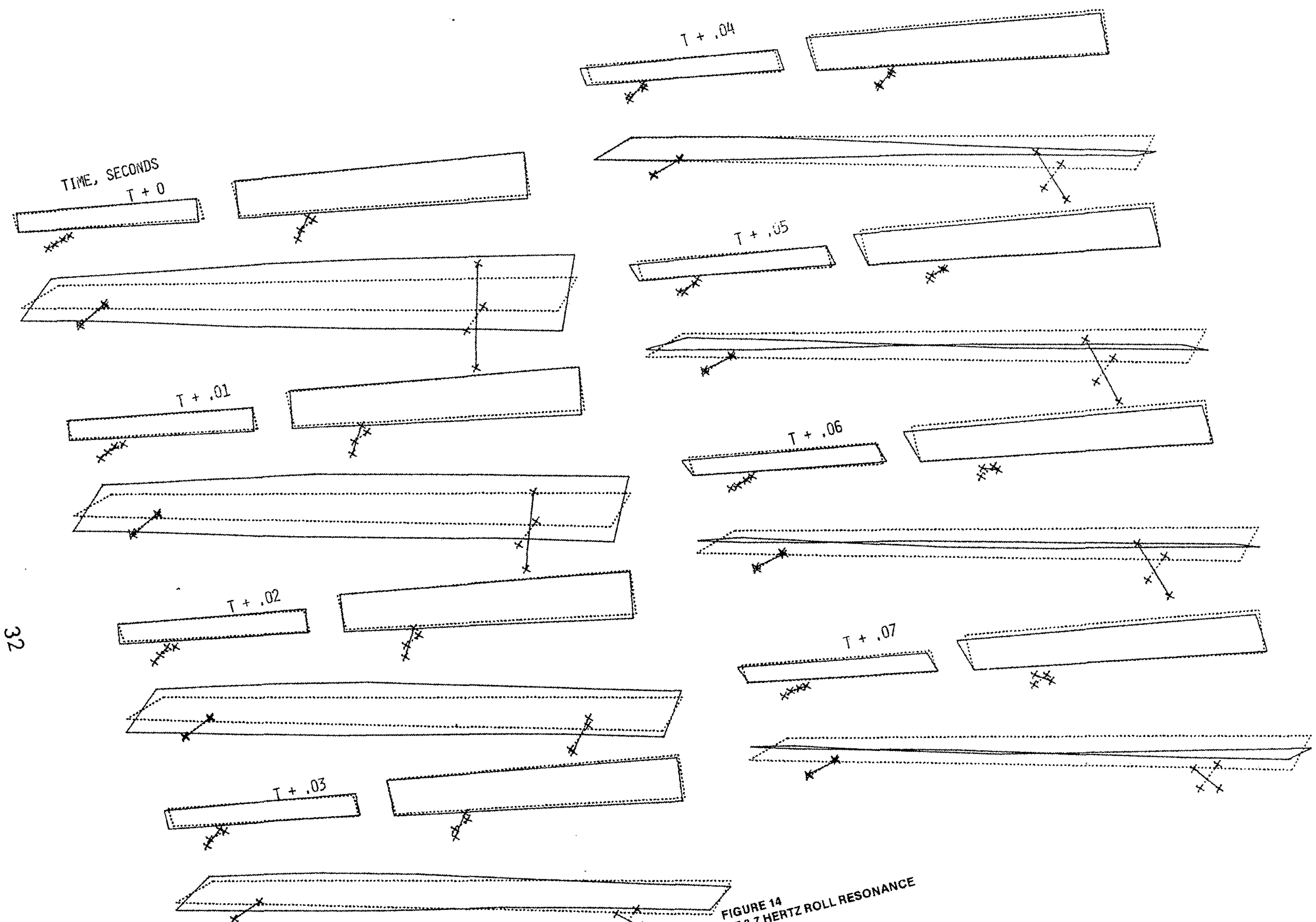



FIGURE 14
FRATE/TOFC SNAPSHOT 8.7 HERTZ ROLL RESONANCE

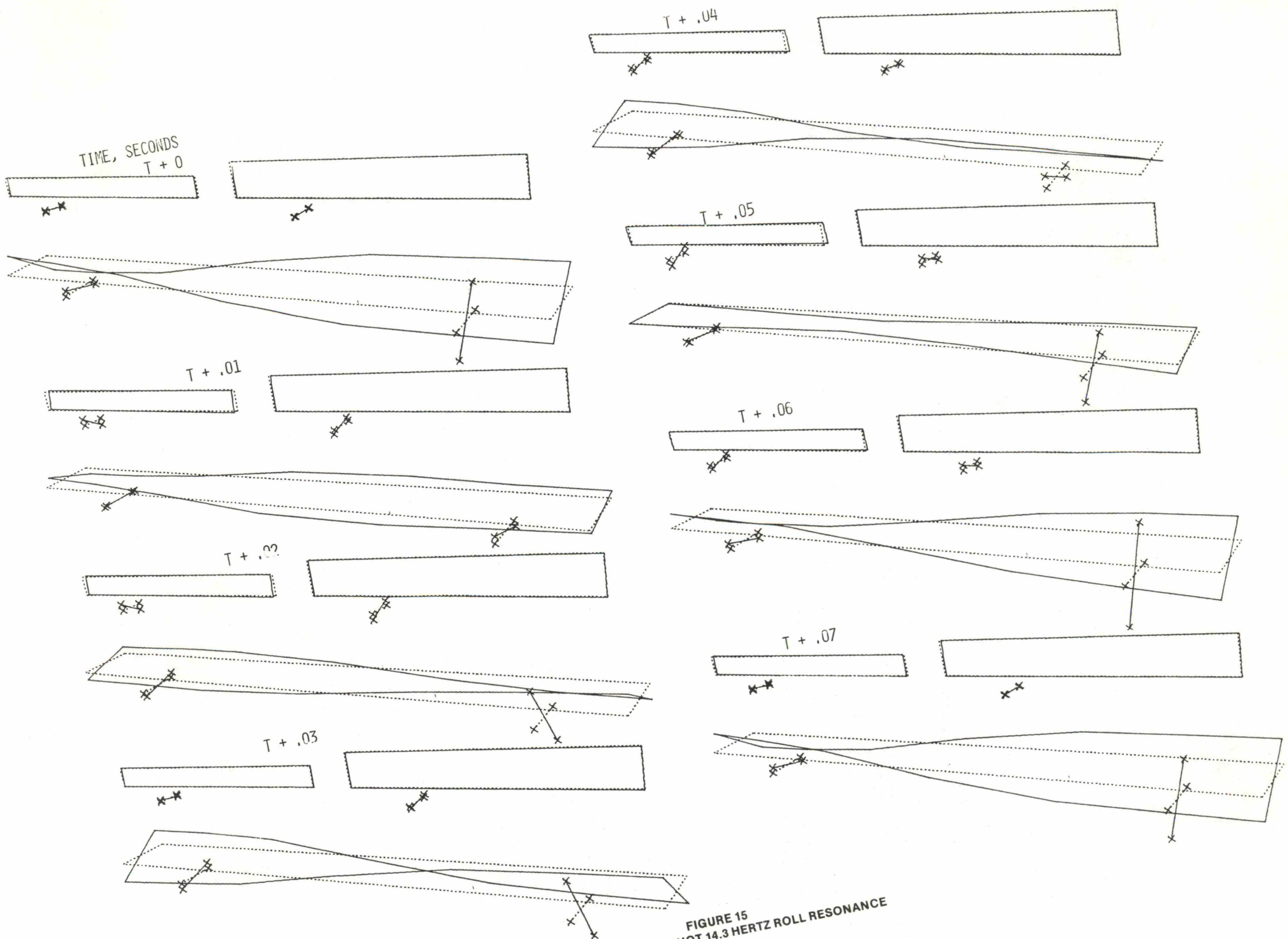



**FIGURE 15**
**FRATE/TOFC SNAPSHOT 14.3 HERTZ ROLL RESONANCE**

## 5.0 DESCRIPTION OF TOFC VIBRATION TEST

Trailer on a Flatcar (TOFC) optimization tests were performed for the Federal Railroad Administration (FRA) using the Vertical Shaker System at the Rail Dynamics Laboratory (RDL), Pueblo, Colorado. The purpose of these tests was to determine the effect of TOFC system component variations on lading acceleration levels. Selected configurations of full-scale loaded and unloaded TOFC systems were tested by exciting all four wheels of one truck of a TTAX flatcar using the Vertical Shaker System (VSS). Accelerometers and displacement transducers were mounted on the shakers, the flatcar, the two trailers located on the flatcar, and lading in the trailers. Analog-conditioned signals from these sensors were input to the VSS computer control system for digitization, recording and additional processing as needed.

The primary goal of this effort was to determine, under controlled laboratory conditions, the effect of TOFC configuration changes on lading acceleration levels. Results from this series of tests could then be used to provide guidelines for future field testing.

There were two basic sources of information used in the work reported in this section. The ENSCO test report, Reference 7, provided detailed information on descriptions of the test set up, test procedure and methods of data conditioning and recording. Reference 7 also provided a summarization of results. The second source of information was copies of processed data requested by Metrek through FRA and supplied by Wyle Laboratories of Colorado Springs, Colorado. This data consisted of sweep test results in the form of response amplitude ratio and phase angle relative to shaker input. The data was provided in both tabular and plotted form.

### 5.1 Test Facility

The Vertical Shaker System (VSS) used for TOFC testing consisted of four 40,000 force-pound computer-controlled hydraulic shakers. As shown in Figure 16, the shakers were positioned under one end of the

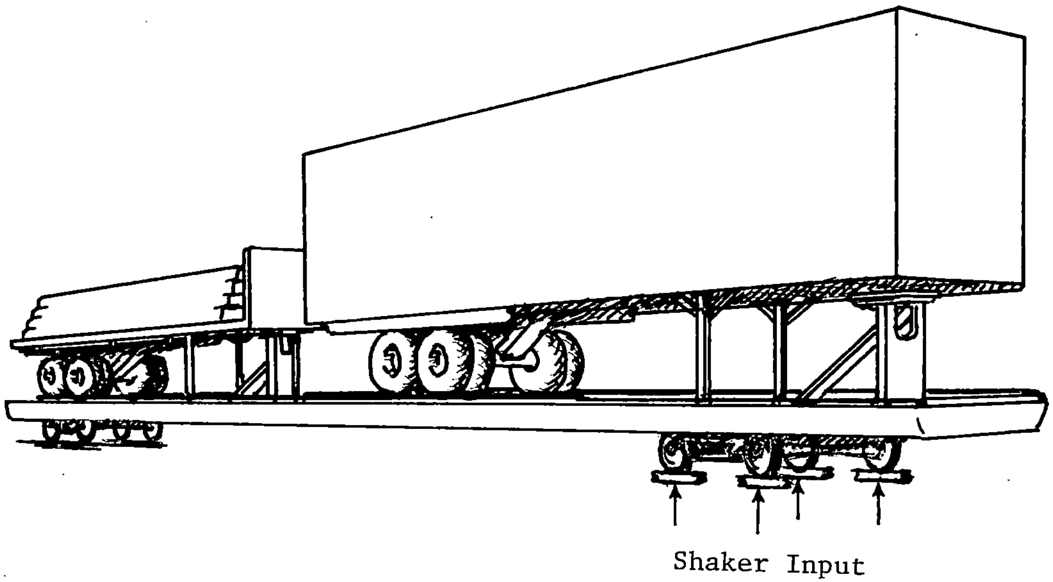



**FIGURE 16**
**TOFC TEST CONFIGURATION**

TTAX flatcar and operated to simulate pitch (all shakers in the phase) and roll (left pair of shakers at $\pi$ phase with the right pair). Instrumentation was located throughout the TOFC vehicle to permit determination of: the frequencies and deflections at vibration resonance of the flatcar and trailers; and the resulting effect on the lading in the trailers.

VSS computer programs were used to drive the shakers either in an incrementally swept sinusoidal frequency mode or in a constant sinusoidal frequency mode over a maximum-frequency range of 0.2 Hz to 30.0 Hz. Initial TOFC pitch and roll sweep tests were performed using constant amplitude and constant acceleration excitation profiles; however, a number of these tests resulted in sufficiently large dynamic response to automatically terminate the test due to VSS safety limit checks. This situation resulted in the establishment of sweep excitation profiles consisting of: a low-frequency constant displacement profile; a mid-frequency constant velocity profile; and a high-frequency constant acceleration profile.

Constant frequency dwell tests were performed after observing TOFC peak response modes of interest from sweep test data. This type of shaker operation permitted close examination of TOFC system dynamics at predominant resonant frequencies. In addition, abrupt termination of the dwell excitation (decay testing) permitted the measurement of TOFC system damping characteristics at the predominant frequencies. Three different dwell and decay VSS excitation level tests were normally performed at each predominant frequency to examine:

- o The response before the friction suspension group was activated.
- o The response while the friction suspension group was operating in the normal mode.
- o The maximum response consistent with safe operating conditions.

The VSS utilized a data acquisition system consisting of a bank of 128 analog signal conditioning and filtering units, a multiplexer and a 12-bit high-speed analog-to-digital converter. Multiplexed digital data is formatted by the VSS computer and recorded on a digital data tape. The recorded signals include vertical shaker control signals, accelerometer signals, displacement transducer signals, and roll angle signals.

Two computer-controlled data collection programs were utilized during TOFC optimization testing. The first was the sweep program which was used to incrementally step the shakers through a pre-selected frequency range, while simultaneously recording 128 channels of data for two and one-half cycles during each frequency step. Digitization rates and frequency step increments were operator selected.

The second data collection program was the DWELL/DECAY program which permitted operator selection of the amplitude, excitation frequency, data sample rate, recording time, and real-time analysis of up to 15 operator-specified channels of test data. The VSS was capable of simultaneously recording 64 channels of data when in the DWELL/DECAY mode. A second dwell test run was required to record all 128 data channels. Decay tests were run at the completion of each dwell test by operator selection of the desired phase angle for abruptly stopping the vertical shakers. Data was then recorded in a manner similar to dwell tests.

## 5.2 Test Configurations

A total of twenty TOFC configurations were tested using the VSS. The TTAX railcar, Serial #973295, utilized a suspension configuration consisting of ASF Ride Control trucks with D-5 spring groups, friction snubbing and standard side bearings. The trailers loaded on the TTAX railcars were a #A11A Trailmobile van trailer, serial number L32820 and a #P41T Trailmobile flatbed trailer, serial number L32821. Additional suspension components tested on the TTAX flatcar included Stucki SV-6B horizontal stabilizers and Stucki-resilient side bearings. Limited testing was also performed with Barber S-2 trucks.

Were these tiedowns the normal way of securing loads?

The van trailer lading consisted of a 50,000 pound palletized load of canned goods and the flatbed trailer lading a 50,000 pound load of rail. Isle blocking and rear area cribbing were utilized to minimize movement of the palletized load in the van and strapping was used to contain the flatbed load.

## 5.3 Summary of Test Results

The quantity of TOFC vibration test data generated and available was far beyond practical limits of analysis. There were 116 sweep tests and 186 dwell and decay tests performed. There were between 80 and 100 response measurements made in each test or some 27000 data plots potentially available for analysis. To obtain and use all of this data was of course out of the question.

Our primary objective was to validate the FRATE/TOFC model. Consequently the analysis of TOFC vibration data was planned to provide information needed for validation; specifically resonant frequencies, deflection shapes at resonance and damping estimates. Dwell and decay tests were dropped in deference to sweep test data. In doing this there was some sacrifice of information on the nonlinear characteristics of the vehicle. The plan limited the analysis of sweep test data to the four configurations of:

a. empty undamped flatcar
b. loaded undamped flatcar
c. loaded, damped flatcar
d. loaded, damped flatcar in an optimized configuration.

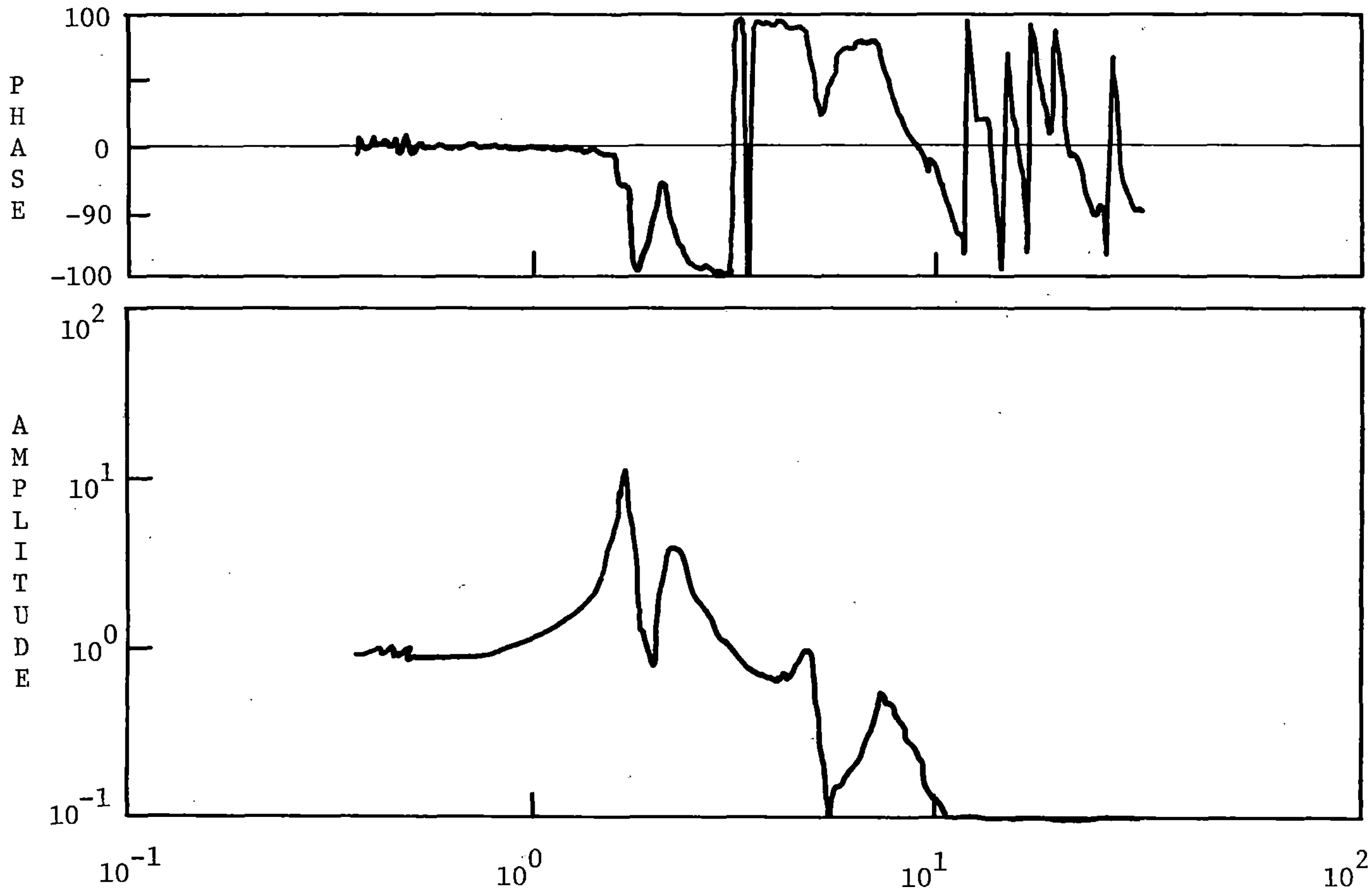
Configuration (a) would permit validation of the flatcar in its simplest and most tractable condition. Configuration (b) would permit a validation of the model including the trailers. Conditions (c) and (d) are the final objectives of representative TOFC configurations.

The data analysis procedure followed is outlined below. The sweep test data was processed into plots of response amplitude ratio and phase angle relative to input motion as a function of frequency. This data was also obtained in computer printout of excitation frequency, response amplitude and response phase angle. Typical plots are shown in Figures 17 and 18.

The phase angle data were used as an aid to identifying resonances since in a well behaved, linear, undamped system the $90^{o}$ phase angle crossing can be used as a resonance identifier. However, because of the nature of our vehicle it was necessary to use the response amplitude ratio plots as the main tool for resonance identification. An outline of the data analysis procedure is shown in Figure 19. The data was first categorized five ways; measurements on the flatcar carbody structure, on the flatcar trucks, on the van trailer trailer structure, on the lading within the van trailer and finally the platform trailer structure. The data was further divided into directions of measurement; vertical, lateral and longitudinal.

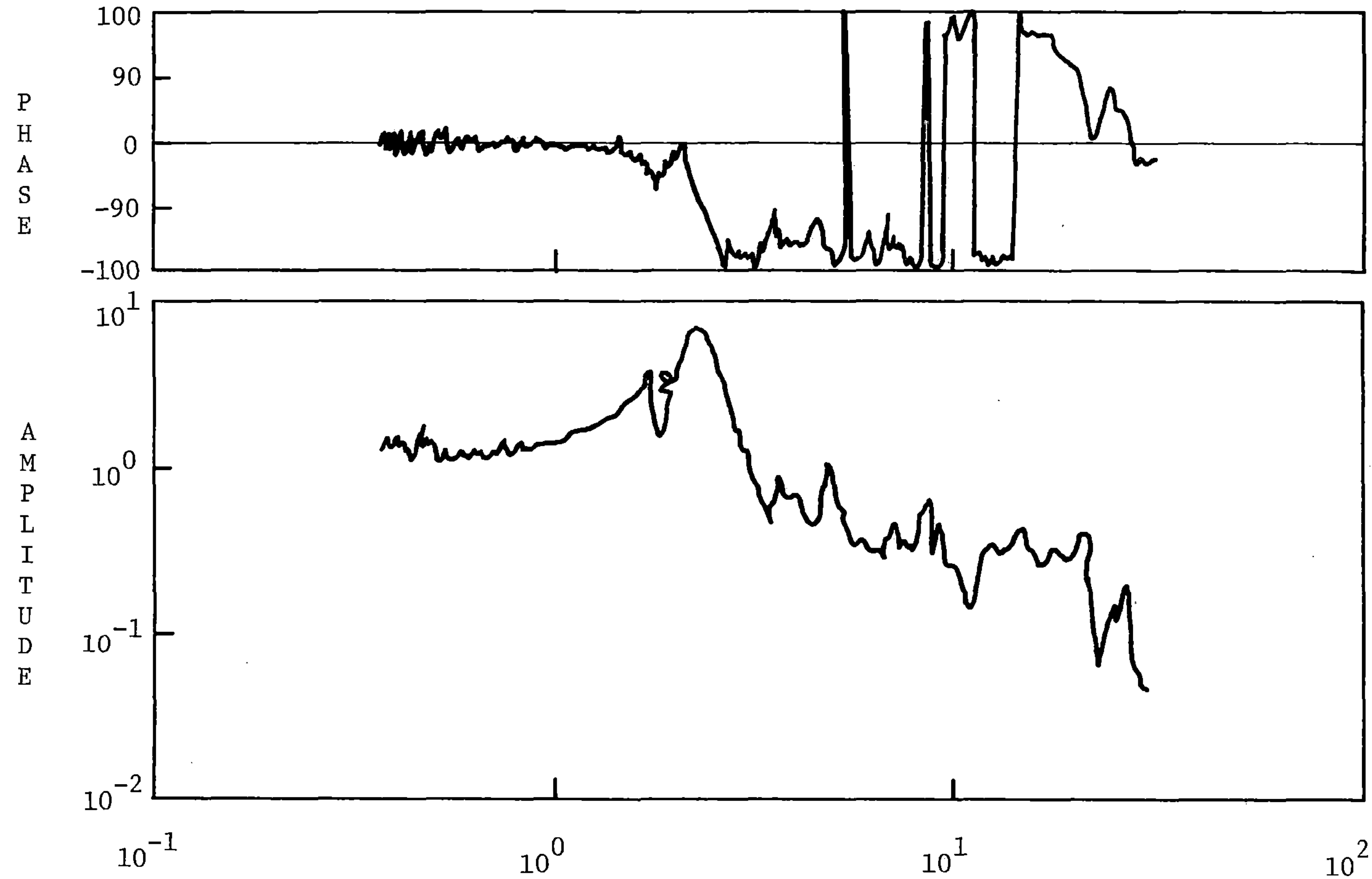
With the data thus divided a superposition tracing was made of all accelerometer measurements on 15 plots grouped three plots on each of five pages. A sample set of these plots are shown in Figures 20 through 24. A study of these superimposed plots, with referral to the original plots as necessary, enabled the identification of resonant frequencies. The primary contributors of each resonance, the direction of predominant motions and a scaling of the relative importance or strength of each resonance were noted.

The next step after resonance identification was to obtain a tabulation of amplitude ratio and phase angle for each resonance identified using the computer tabulation. During this process a more




PLOT #169
43 T57A , 54,E1,/,
44 T57B , 54,E2,/,

**FIGURE 17**
**EXAMPLE TEST DATA PLOT TOFC OPTIMIZATION TEST**
**(PLOT #169)**




PLOT #132
19 T57A , 66,E1,/,
20 T57B , 66,E2,/,

**FIGURE 18**
**EXAMPLE TEST DATA PLOT TOFC OPTIMIZATION TEST**
**(PLOT #132)**

1. REQUEST DATA FROM DISC STORAGE

   - FORMAT: PLOTS - PRINT
   - WHICH RUNS SPECIFIED
   - WHICH MEASUREMENTS SPECIFIED

2. GENERATE SUPERIMPOSED SUMMARY PLOTS

3. IDENTIFY RESONANT FREQUENCIES

   - SUMMARY PLOTS
   - SINGLE PLOTS
   - PRINT DATA

4. TABULATE AMPLITUDE AND PHASE ANGLE FROM PRINT DATA FOR EACH RESONANCE IDENTIFIED

5. IDENTIFY PREDOMINANT PHASE ANGLE

6. STORE DATA IN COMPUTER

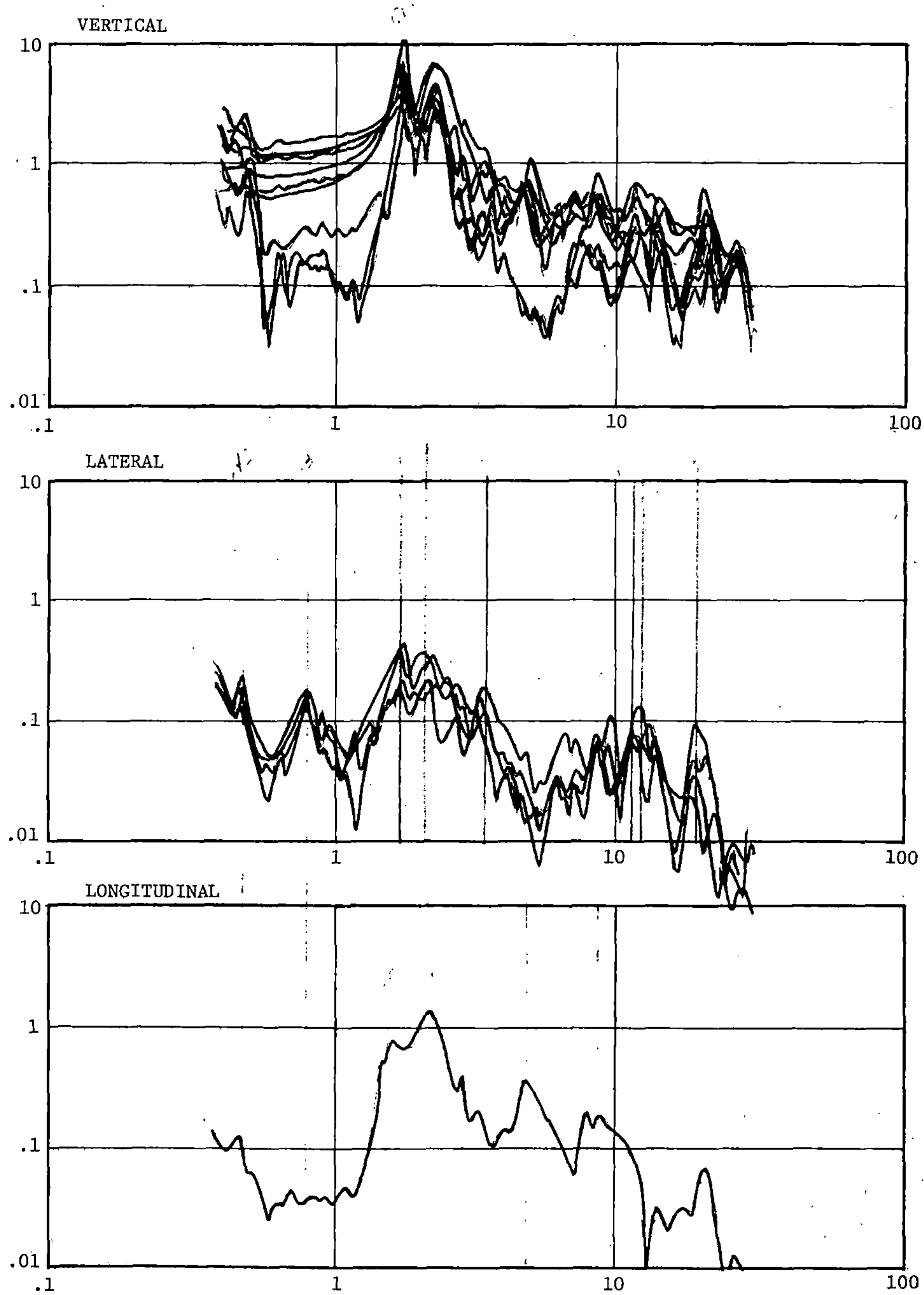
7. OBTAIN COMPUTER PLOTTED DEFLECTION MEASUREMENTS FOR SEVERAL PHASE ANGLES

8. DRAW DEFLECTION SHAPES

**FIGURE 19**
**OUTLINE OF TEST DATA ANALYSIS PROCESS**

axis labels?

amplitude?




frequency?

FIGURE 20

**SUPERIMPOSED TEST DATA,CONFIGURATION 7, RUNS 57A AND 57B CARBODY MEASUREMENTS**

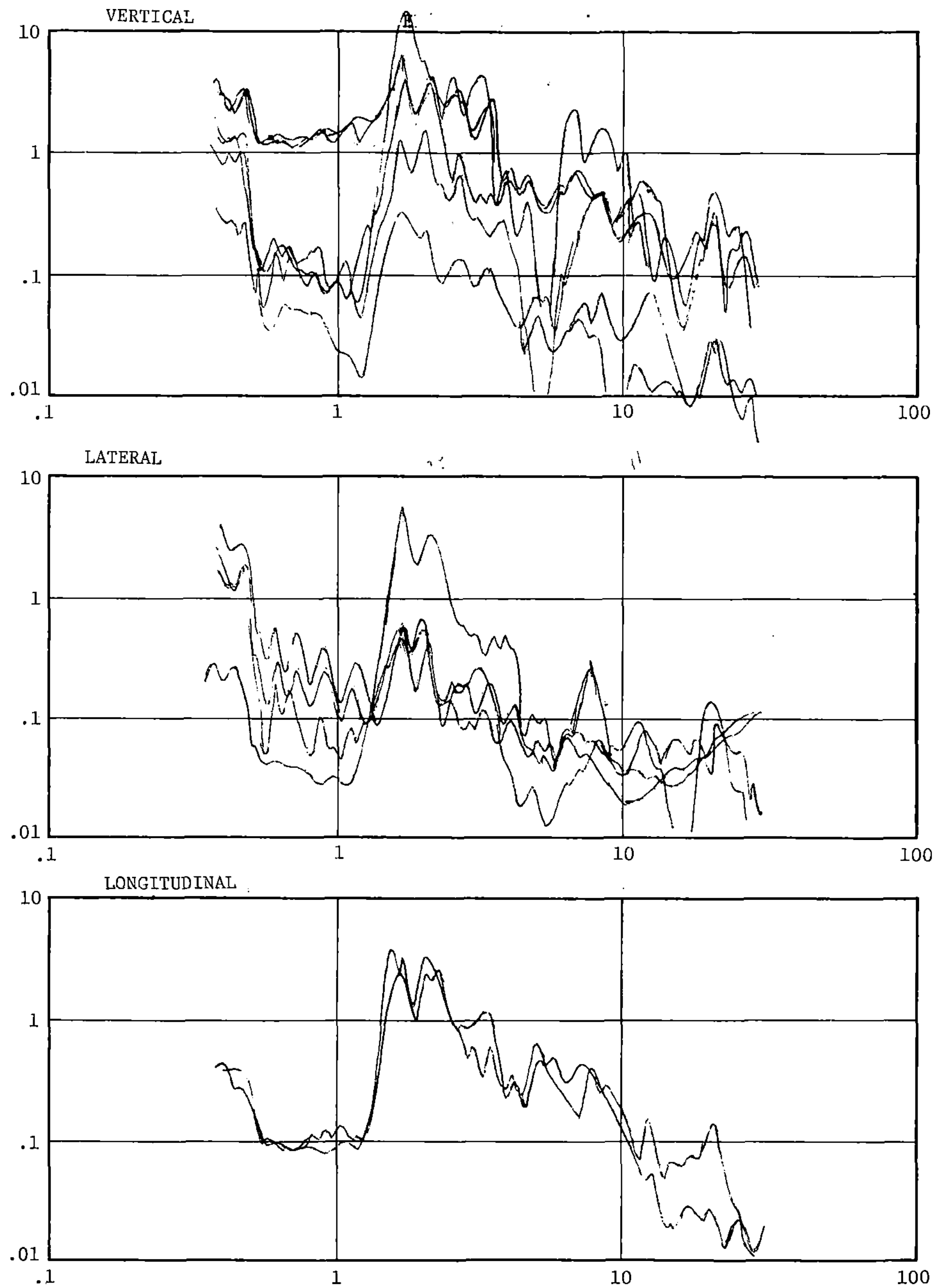



**FIGURE 21**
**SUPERIMPOSED TEST DATA, CONFIGURATION 7, RUNS 57A AND 57B**
**TRUCK MEASUREMENTS**

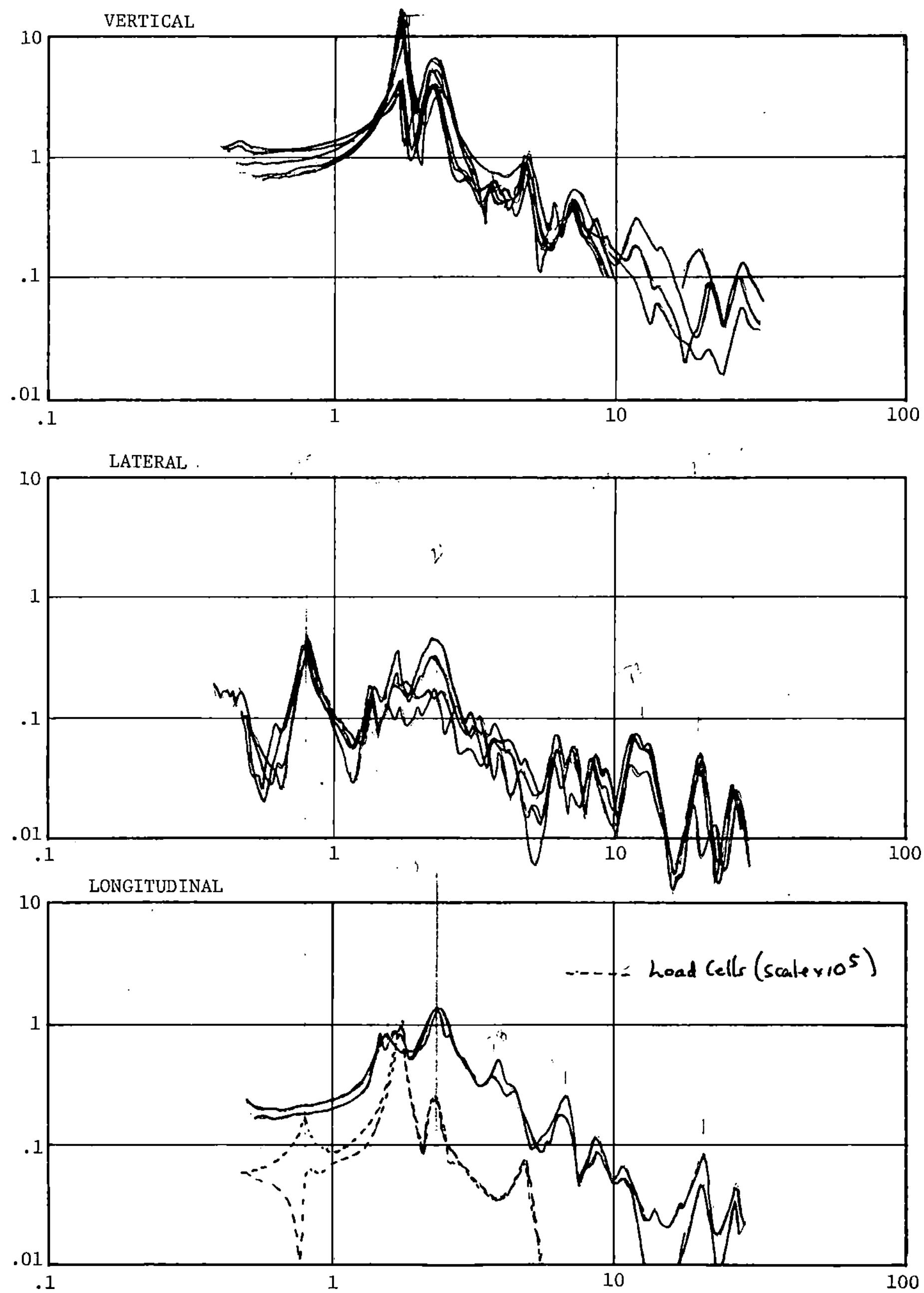



**FIGURE 22**
**SUPERIMPOSED TEST DATA,CONFIGURATION 7, RUNS 57A AND 57B**
**VAN TRAILER MEASUREMENTS**

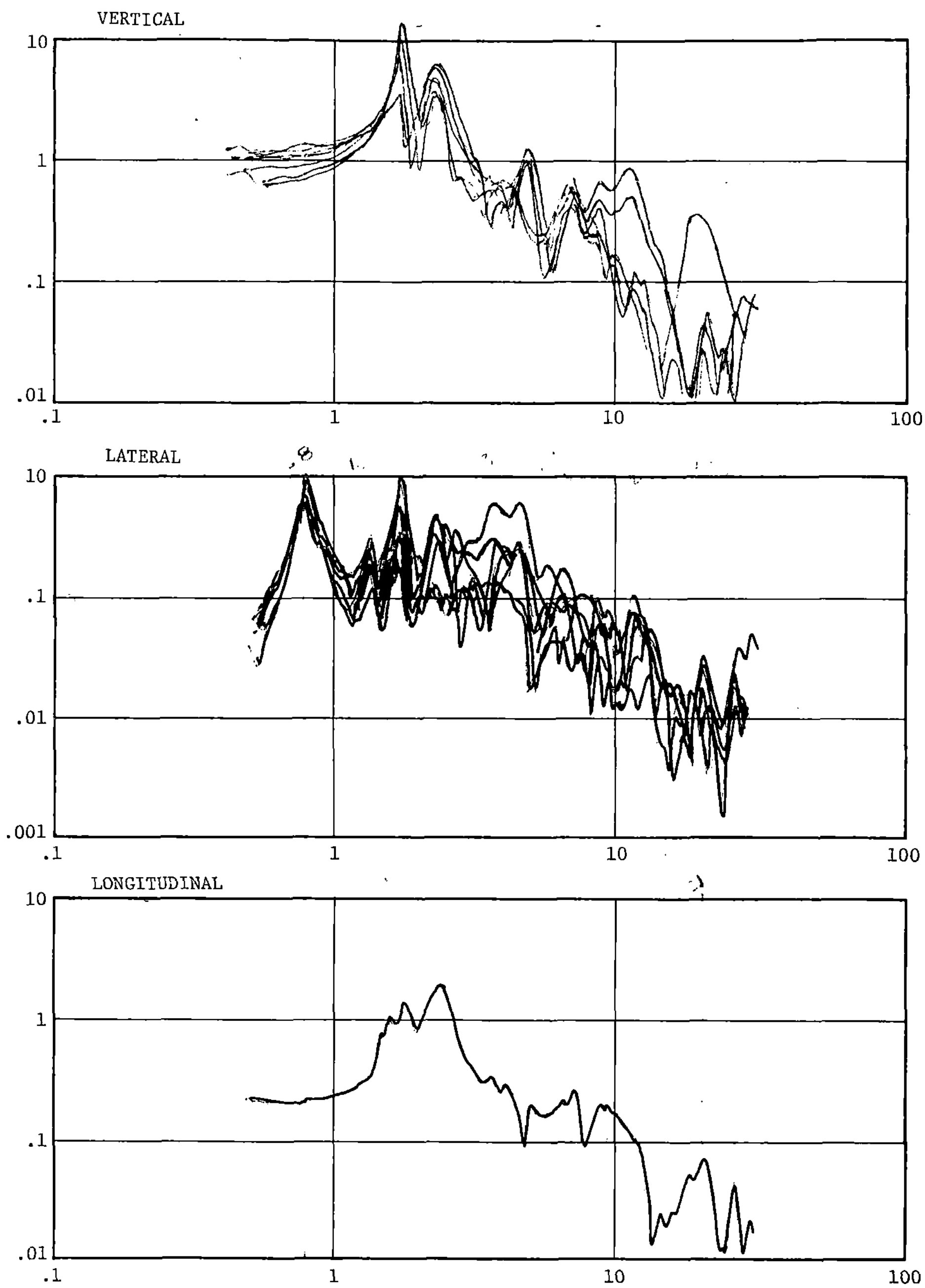



**FIGURE 23**
**SUPERIMPOSED TEST DATA,CONFIGURATION 7, RUNS 57A AND 57B**
**LADING MEASUREMENTS**

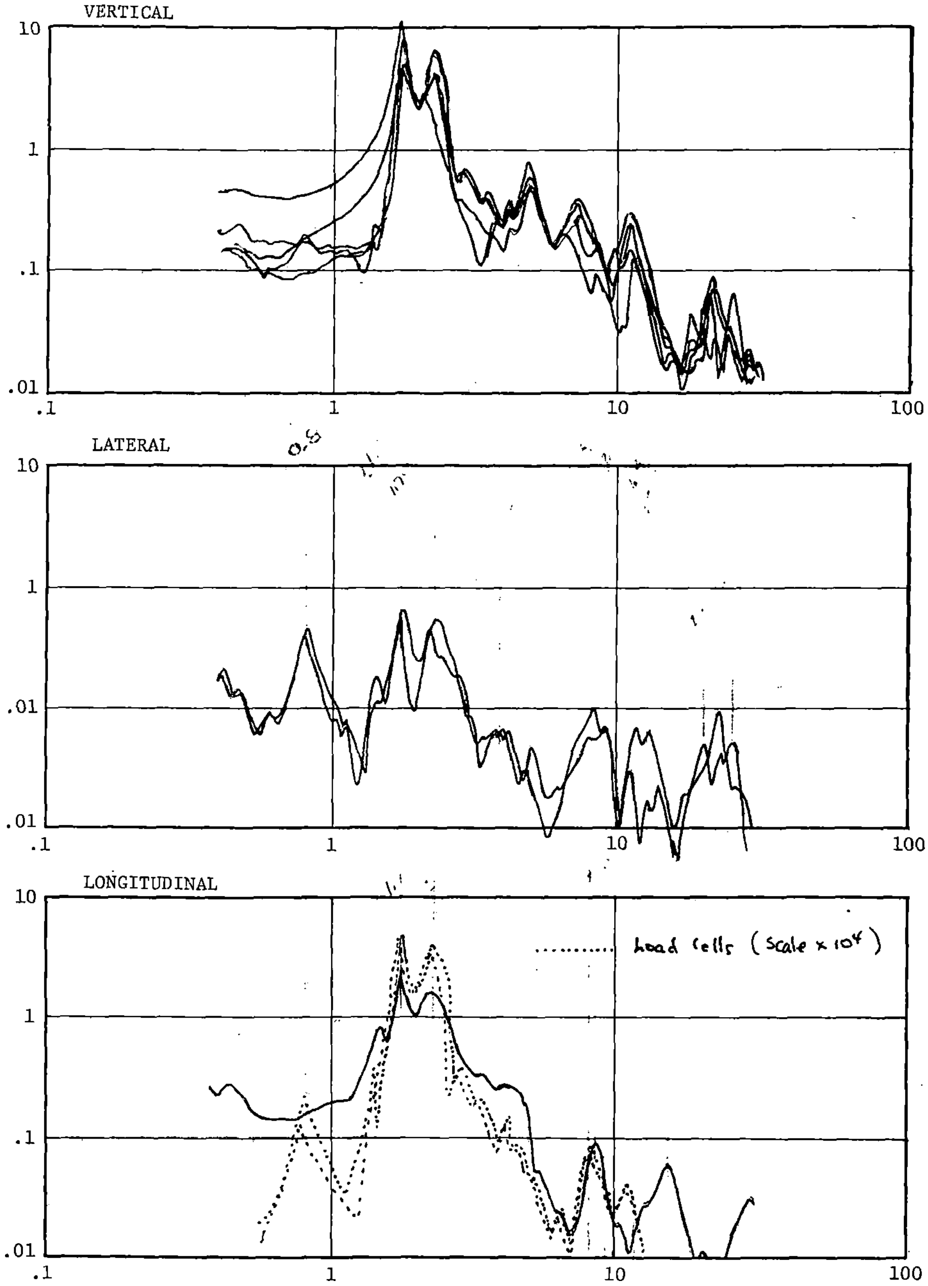



**FIGURE 24**
**SUPERIMPOSED TEST DATA,CONFIGURATION 7, RUNS 57A AND 57B**
**PLATFORM TRAILER MEASUREMENTS**

precise frequency identification was obtained through study of the computer printout looking for maximum responses and $90^{o}$ phase angle crossings.

In confirmation of analysis results discussed earlier it became apparent that we would not be able to obtain a deflection shape in its general sense (normal mode). There was an absence, in the phase angle data, of the in-phase or out-of-phase relationship associated with a normal mode deflection shape. It was necessary to study each deflection shape tabulation to find a predominant phase angle. This phase angle, call it $\phi_o$, was used to adjust each amplitude by the following relationship:

$$A_i = A_i \sin (\phi_i + 90 - \phi_o)$$

In order to facilitate the plotting of the test deflection shapes the plot subroutine SNAPSHOT which had been developed in conjunction with FRATE was modified to accept the test data. The test data from the fully loaded TOFC with roll excitation did not result in intelligible deflection shapes using the computerized plot program. Description of the resonances for this case were based on a study of the tabular data and guidance from the analytical results.

Resonant frequencies identified are listed in Tables V and VI.

Deflection shape plots are shown in Figures 25 through 28.

TABLE V

RESONANT FREQUENCIES FROM TEST
VERTICAL MODE EXCITATION

EMPTY FLATCAR

| DESCRIPTION | HERTZ |
| --- | --- |
| VERTICAL TRANSLATION WITH BENDING | 3.444 |
| PITCH | 4.213 |
| CARBODY BENDING | 5.185 |
| CARBODY SECOND BENDING | 10.805 |
| CARBODY THIRD BENDING | 17.640 |

TRAILER ON FLATCAR

| DESCRIPTION | HERTZ |
| --- | --- |
| TRANSVERSE RESONANCE | .799 |
| COUPLED CARBODY PITCH & BENDING, PLATFORM TRAILER BENDING AND VAN TRAILER PITCH | 1.711 |
| CARBODY PITCH/BENDING WITH TRAILER BENDING | 2.238 |
| CARBODY PITCH/BENDING WITH TRAILER BENDING | 4.803 |
| TRAILER SECOND BENDING | 7.33 |
| COUPLED CARBODY BENDING & VAN TRAILER BENDING | 8.844 |
| CARBODY SECOND BENDING | 11.506 |

TABLE VI

RESONANT FREQUENCIES FROM TEST
ROLL MODE EXCITATION

EMPTY FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| ROLL WITH LOW CENTER | 1.184 |
| ROLL WITH HIGH CENTER | 2.000 |
| CARBODY TORSION | 10.114 |
| CARBODY SECOND TORSION | 16.209 |

TRAILER ON FLATCAR

| DESCRIPTION | HERTZ |
|---|---|
| CARBODY ROLL WITH LOW CENTER | .539 |
| CARBODY YAW | 1.131 |
| CARBODY ROLL WITH HIGH CENTER | 2.330 |
| PLATFORM TRAILER ROLL | 2.681 |
| COUPLED TRAILER & CARBODY ROLL | 3.747 |
| COUPLED TRAILER & CARBODY ROLL | 6.222 |
| COUPLED TRAILER & CARBODY ROLL | 8.425 |
| CARBODY TORSION | 12.936 |

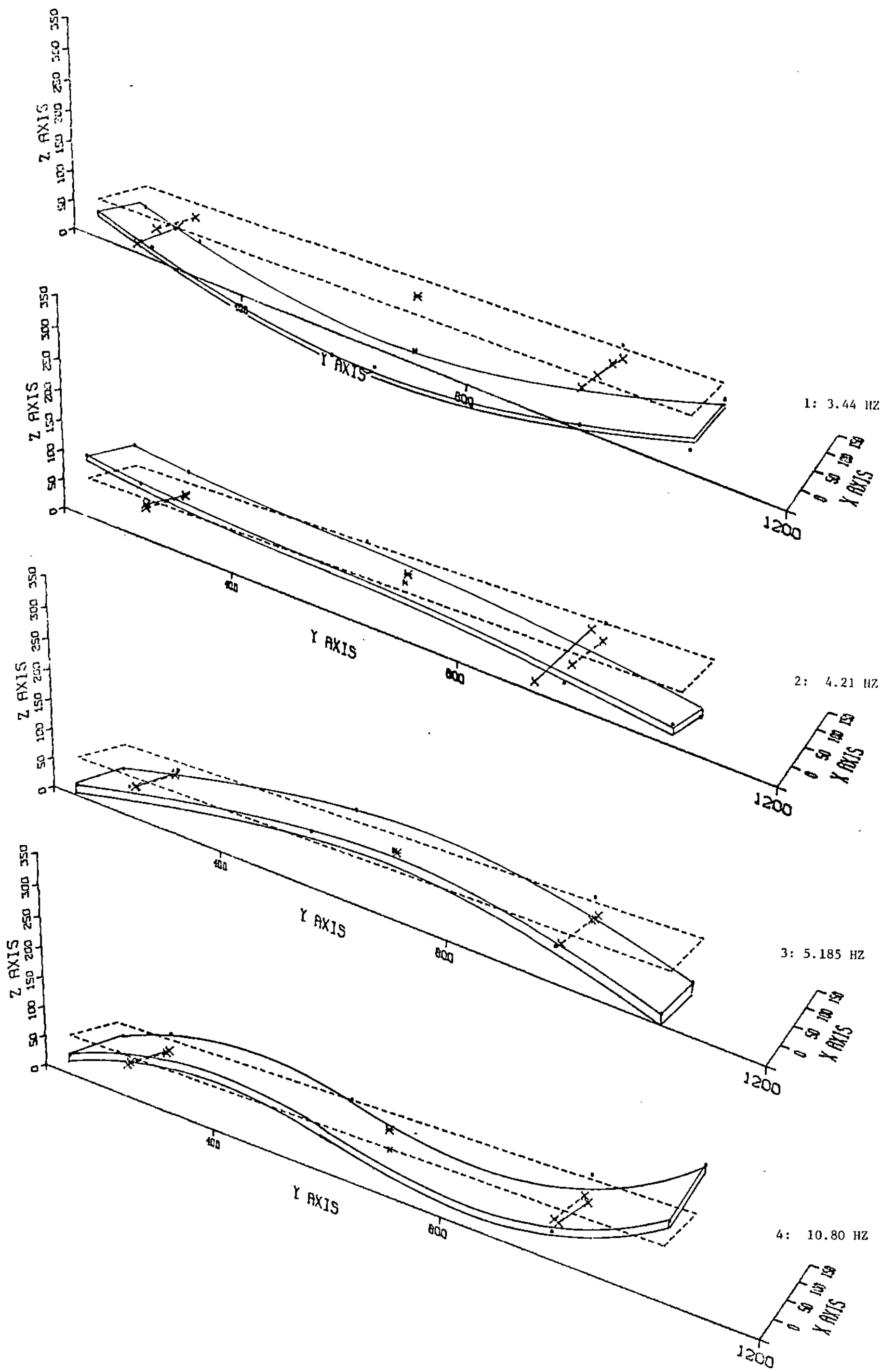



FIGURE 25
EMPTY FLATCAR TEST DEFLECTION SHAPES VERTICAL EXCITATION

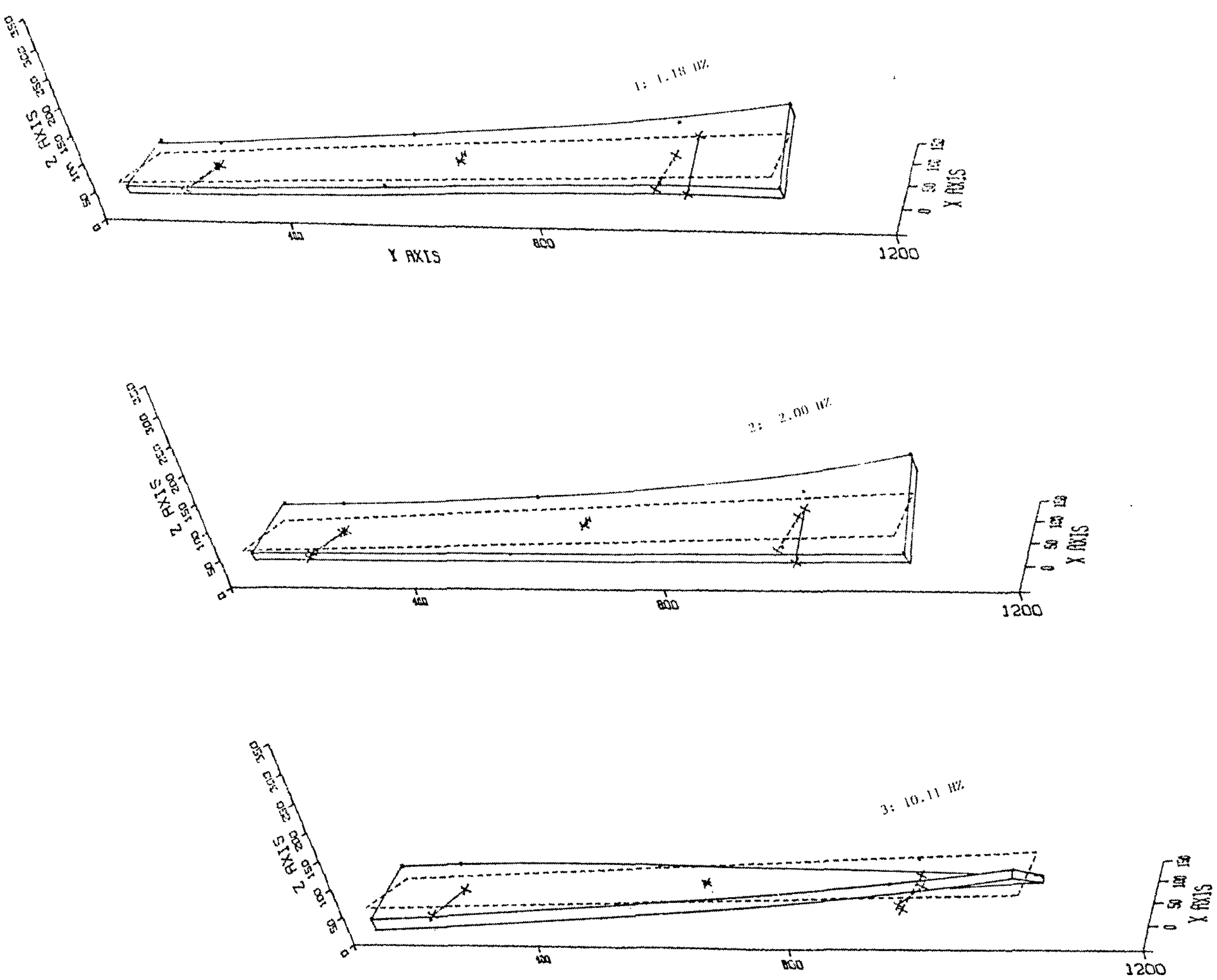



FIGURE 26
EMPTY FLATCAR TEST DEFLECTION SHAPES ROLL EXCITATION

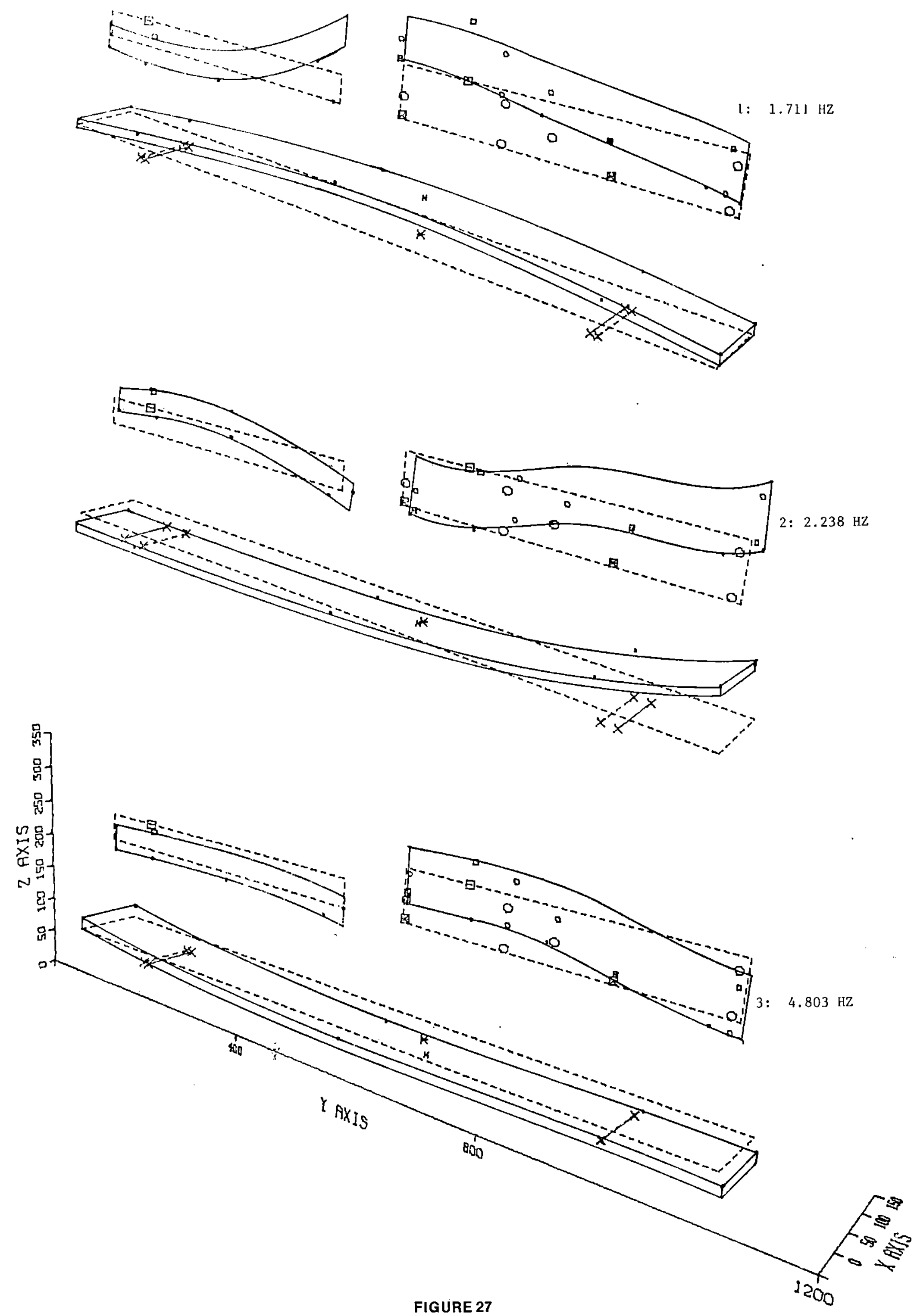



FIGURE 27
TOFC TEST DEFLECTION SHAPES VERTICAL EXCITATION

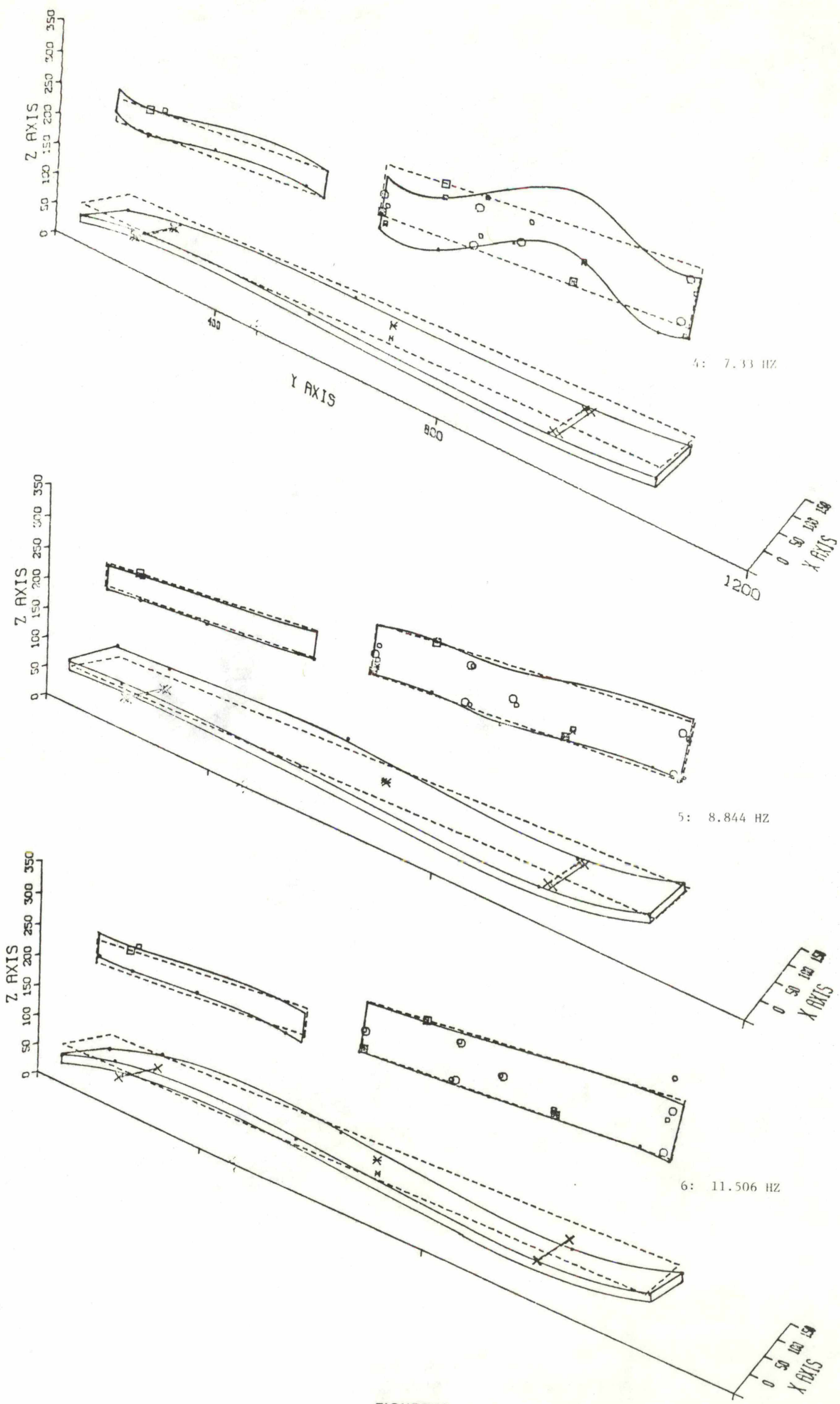



**FIGURE 28**
**TOFC TEST DEFLECTION SHAPES VERTICAL EXCITATION**

## 6.0 COMPAIRSON BETWEEN TEST AND ANALYSIS AND MODIFICATIONS TO MODEL TO IMPROVE AGREEMENT

The validation plan was to proceed sequentially from the empty, no snubber configuration to the fully loaded, 100% snubber. The implied assumption in this plan was that differences between the empty and full configurations could be accounted for by mass and inertia differences. However, after a study of the resonant frequencies, summarized in Table VII, it was concluded that some spring rates in the rail car suspension system changed with the change in loading conditions. Consequently our approach was modified to comparing full and empty conditions simulataneously so that model changes could be picked which would result in improvements for both conditions.

### 6.1 Observations from Test to Analysis Comparison

For example, referring to Table VII, the FRATE analysis frequencies are close to test frequencies for the fully loaded condition but high for the empty condition. The comparison is especially poor for the empty condition, first roll resonance where the analysis frequency is 3.2 times test. Model change objectives were consequently set to decrease stiffness for the empty condition and to find a change which would result in a change in roll stiffness but not in vertical stiffness.

A number of other observations were made from the comparisons of test and analysis results. These observations are summarized below.

*Missing Resonances* in the FRATE results for the loaded conditions were noted and were hypothesized to be due to inadequate trailer models. This hypothesis was reinforced by the results of trailer vibration tests as reported in Reference 8. Reference 8 compared test results to the trailer model used in FRATE with the conclusions that trailer suspension should be softened and that trailer flexibility should be included.

TABLE VII

SUMMARY COMPARISON OF TEST & ANALYSIS RESONANT FREQUENCIES USING PRE VALIDATION FRATE MODEL

A. VERTICAL RESONANT FREQUENCIES (HERTZ)

A.1 EMPTY FLATCAR

| DESCRIPTION | TEST | ANALYSIS |
|---|---|---|
| VERTICAL TRANSLATION WITH BENDING | 3.444 | 3.6 |
| PITCH | 4.213 | 6.7 |
| CARBODY BENDING | 5.185 | 8.1 |
| CARBODY SECOND BENDING | 10.805 | 13.0 |
| CARBODY THIRD BENDING | 17.640 | - |

A.2 LOADED FLATCAR (TOFC)

| DESCRIPTION | TEST | ANALYSIS |
|---|---|---|
| COUPLED TRAILER/ CARBODY PITCH/ BENDING | 1.711 | 1.77 |
| COUPLED TRAILER/ CARBODY PITCH/ BENDING | 2.238 | 2.2 |
| COUPLED TRAILER/ CARBODY PITCH/ BENDING | 3.65 | 3.3 |
| COUPLED TRAILER/ CARBODY PITCH/ BENDING | 4.803 | - |
| TRAILER SECOND BENDING | 7.33 | - |
| COUPLED CARBODY BENDING & VAN TRAILER BENDING | 8.844 | - |
| CARBODY SECOND BENDING | 11.506 | 12.0 |

B. ROLL RESONANT FREQUENCIES (HERTZ)

B.1 EMPTY FLATCAR

| DESCRIPTION | TEST | ANALYSIS |
|---|---|---|
| ROLL WITH LOW CENTER | 1.184 | 3.8 |
| ROLL WITH HIGH CENTER | 2.000 | 5.5 |
| CARBODY TORSION | 10.114 | 13.3 |
| CARBODY SECOND TORSION | 16.209 | - |

B.2 LOADED FLATCAR (TOFC)

| DESCRIPTION | TEST | ANALYSIS |
|---|---|---|
| CARBODY ROLL WITH LOW CENTER | .50-.80 | .87 |
| CARBODY YAW | 1.131 | - |
| CARBODY ROLL WITH HIGH CENTER | 2.330 | - |
| PLATFORM TRAILER ROLL | 2.681 | - |
| COUPLED TRAILER & CARBODY ROLL | 3.747 | 3.5 |
| COUPLED TRAILER & CARBODY ROLL | 6.222 | - |
| COUPLED TRAILER & CARBODY ROLL | 8.425 | 8.7 |
| CARBODY TORSION | 12.936 | 14.3 |

*Resonances Identified with Carbody Flexibility* were noted to be high for the empty conditions. The comparisons appeared to be better for the full conditions but the missing resonances and the effects of the inadequate trailer models made it difficult to reach a clear conclusion.

*Nonlinear Effects* were most noticeable in the first roll resonance in the fully loaded condition. The apparent resonant frequency was sensitive to level of excitation and was found to vary between 0.5 and 0.8 Hertz.

The difference in response between up sweep and down sweep, which was consistantly as shown in Figure 29 is typical of a nonlinear softening spring. (Sweep rates in this frequency range were equivalent to about 10 minutes per octave which is slow enough to have no effect on response differences.)

*Deflection Shapes and Effective System Damping* were also compared. The comparison provided directions for changes in relative stiffness in the truck model springs to improve relative displacement comparisons and directions for changes to damping in general to improve response amplitude comparisons.

## 6.2 Modifications to FRATE and the TOFC Simulation

There were many changes made to FRATE to achieve closer agreement between experimental and analytical results. A greater part of the changes were variations of spring and damper values. There were three mass term changes. There were changes to model topography requiring changes to certain of the basic equations of motion. The changes to the trailer models recommended in Reference 8 were incorporated with the exception that the trailers remained rigid bodies. Finally a coulomb damping representation was included.

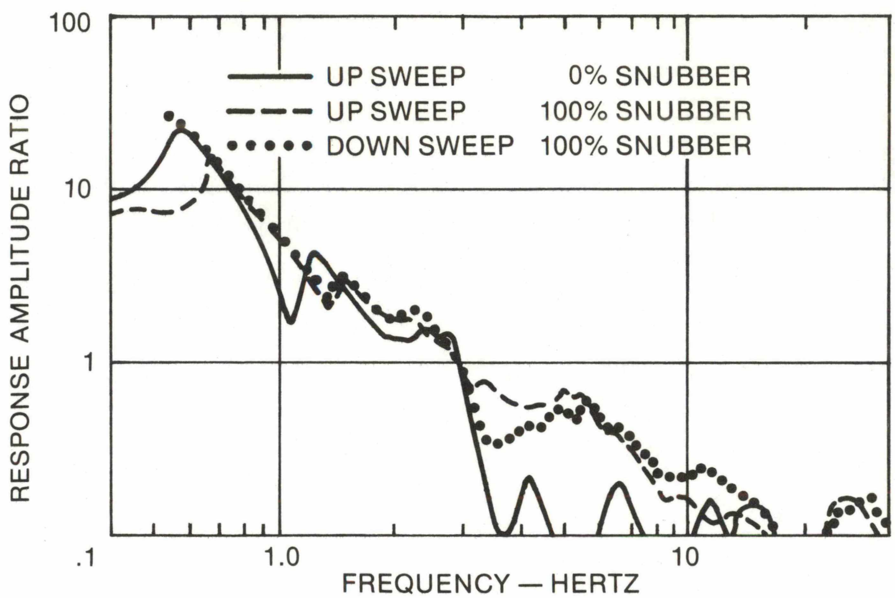



**FIGURE 29**
**COMPARISON OF RESPONSE WITH UP AND DOWN SWEEP, TOFC OPTIMIZATION RESPONSE LOCATION NO. 48 (LADING, TOP LAYER, CENTER OF TRAILER, LATERAL)**

The total modification procedure was too lengthy and involved to be described here in toto. However certain changes made to the rail-car trucks are felt to be significant and are discussed. These changes have to do with a simulation of center plate and side bearing effects, a nonlinear transverse spring rate and a friction snubber. Several other changes are also briefly reviewed.

### 6.2.1 Center Plate and Side Bearing Effects

With an examination of the truck configuration, sketched in Figure 30, one would conclude that the roll spring rate will have three different values; i.e., for when the center plate is seated, for when the center plate is rocking and for when the side bearing is in contact. That is, for small amplitudes of roll motion where the center plate remains flat on its seat the major flexibility in roll is in the truck springs. At some amplitude of roll motion the transverse "g" forces will cause a moment at the center plate greater than the hold down moment afforded by the vertical "g" load of the carbody. At this point the center plate will start its rocking motion on its seat. As the roll amplitude increases the restoring moment on the carbody will decrease until side bearing contact is made. After side bearing contact the restoring moment is again dependent on the truck spring properties.

The center plate rocking and side bearing contact were concluded to be the probable major cause of the nonlinear characteristics displayed in the test data; e.g., Figure 29. It was consequently decided that these effects should be included in the FRATE simulation.

Tse and Martin, Reference 9, have modeled the center plate and side bearings as sketched in Figure 31. Ke1 and Ke2 are relatively stiff springs representing side bearing local structure with a gap representing side bearing clearance. Kp1 and Kp2 are relatively stiff compression springs representing center plate local structure. As

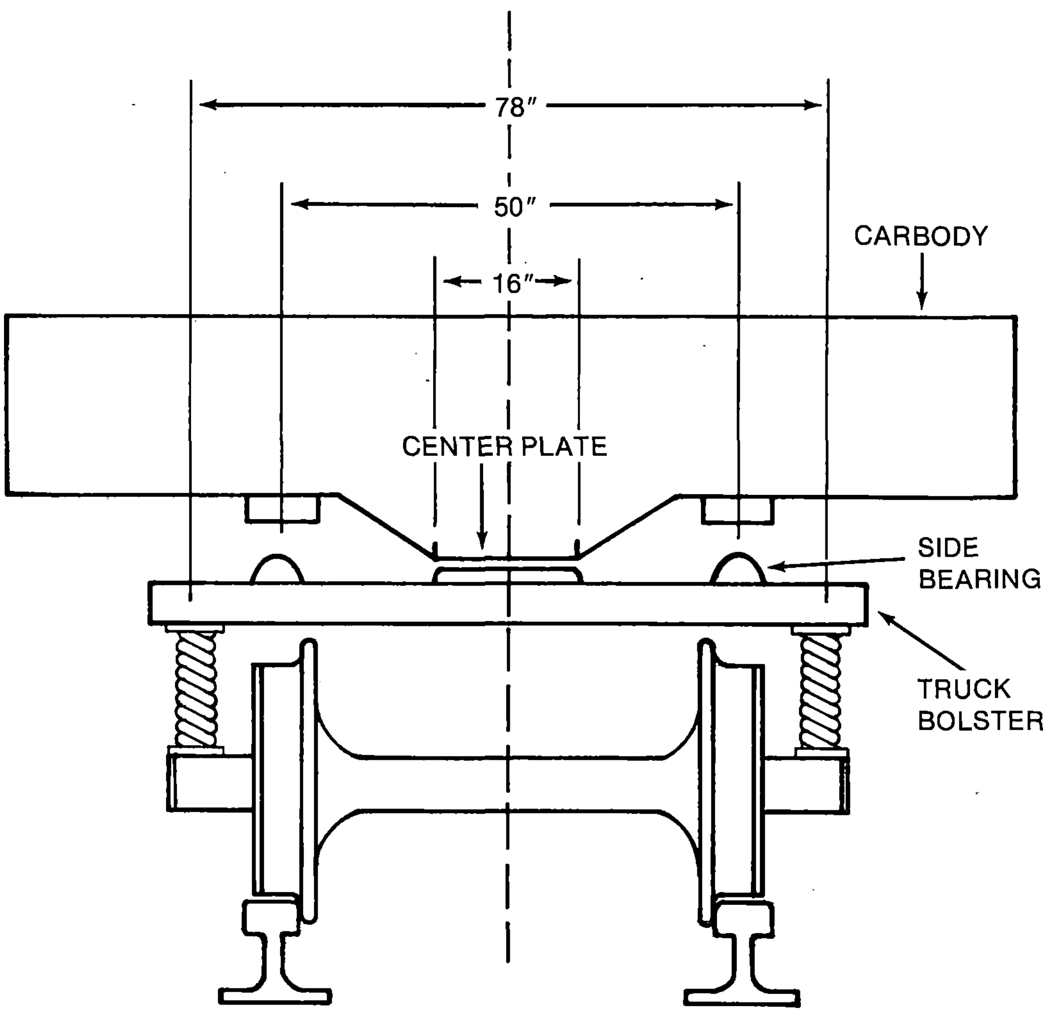



**FIGURE 30**
**SKETCH OF FREIGHT CAR END VIEW SHOWING CENTER PLATE AND SIDE BEARING**

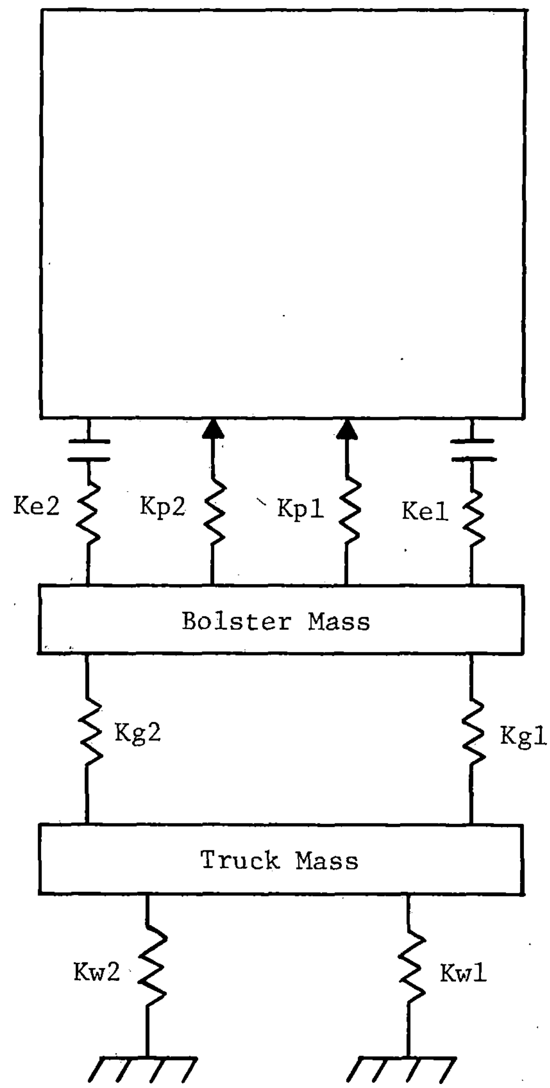


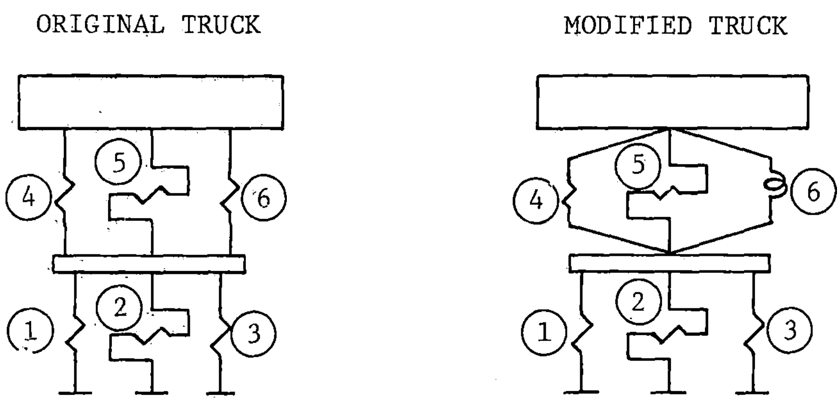

**FIGURE 31**
**SCHEMATIC OF TRUCK MODEL USED BY TSE & MARTIN, REFERENCE 9**

the carbody rolls one or the other of the center plate springs, depending on the direction of roll, will act as a pivot point about which the carbody rotates until the side bearing gap is closed. This is considered to be an accurate representation of the complex rocking motion of the carbody on the center plate. It, however, has disadvantages in that the bolster mass adds two degrees of freedom per truck to the model and, more important, it introduces a high frequency element requiring a smaller integration time step.

It was decided to keep the one mass truck model in FRATE and thereby avoid the disadvantages of the two mass simulation. It was also desired to have a minimum modification to the existing equations in the FRATE program. The FRATE truck model was consequently modified as sketched in Figure 32. The new K(4) replaces the vertical properties of the old combined K(4) and K(6). The new K(6) replaces the roll properties of the old combined K(4) and K(6). The corresponding viscous dampers were also modified in the same manner. The next step was to program K(6), the truck roll spring, as a trilinear spring to simulate the seated center plate, the rocking center plate and side bearing contact.

However, we were unsuccessful in our attempt to simulate the kinematics of the rocking center plate with the one mass truck model. As a compromise the truck roll spring was modeled as bilinear, the first section being made to represent roll motion before side bearing contact and the second section to represent roll motion after side bearing contact.

With the bilinear truck roll spring it was recognized that the nonlinear, softening spring characteristics would not be achieved. However it was concluded that the model could be considered valid if correlation of resonant frequencies was achieved and worst care (i.e., down sweep) resonance response was matched.




OPTIONS AVAILABLE WITH MODIFIED TRUCK

- Bilinear Roll Spring
- Side Bearing Gap
- Center Plate Rocking
- Coulomb Damping

**FIGURE 32**
**B TRUCK SPRING DAMPER MODIFICATION FOR CENTER PLATE ROLL**

### 6.2.2 Coulomb Damping

Figure 33 is a truck side view which shows the snubber location. The snubbers are a pair of spring loaded friction shoes that move with the bolster and bear against friction plates on the side frame. Until the static friction forces in the snubbers are overcome they provide a direct structural path from the bolster to the side frame, effectively locking out the spring nest. Once the static friction has been overcome the bolster moves relative to the side frame, working against the spring nest and the snubber sliding friction.

Simulation of the snubber damper was incorporated into FRATE with the sliding point concept used in References 11 and 12. Figure 34 shows this change schematically. Again vertical motions and rolling motions are kept separate. KS4 and KS6 are springs which represent the relatively stiff structure of the bolster and side frame. The program logic was rewritten with the spring forces in K(4) and KS4 to be additive, with the force in KS4 limited to the value MFS4 at which point the end of KS4 "slides". When the motion reverses, KS4 will unload and load to -MFS4 at which point sliding again occurs. A load deflection path is pictured in Figure 35.

The roll counterparts K(6), KS6 and MFS6 were set up in a slightly different manner because of the center plate/bilinear spring configuration. The truck and local structure springs do not start to load up until the side bearing gap has been reached. A load deflections path for the roll springs is shown in Figure 36.

### 6.2.3 Nonlinear Transverse Truck Spring

The springs K(5) and K(11) represent the transverse stiffness of trucks. Extensive testing has been performed and reported in Reference 10 on 70 ton truck components. The report shows the transverse spring rates for the spring nest of our configuration to be very nonlinear.

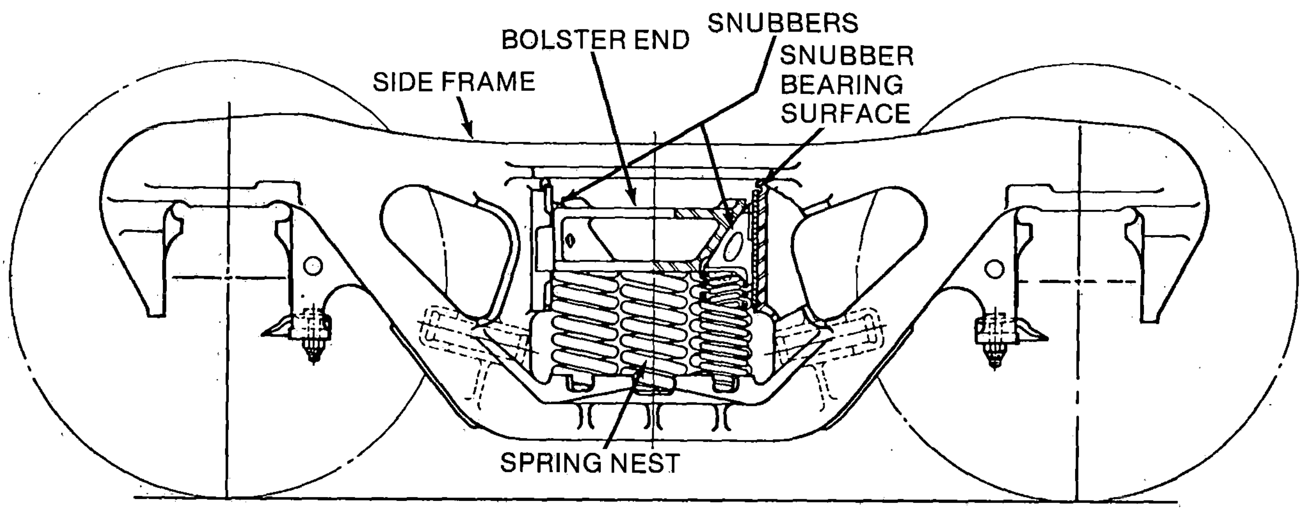
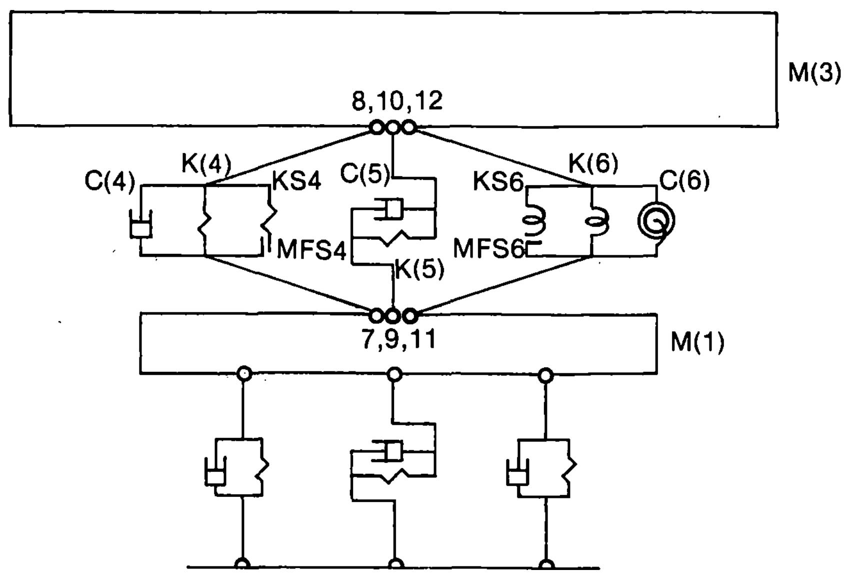



**FIGURE 33**
**SIDE VIEW OF TRUCK SHOWING SNUBBER LOCATION**




| | | |
|---|---|---|
| Legend: | K(4) | = total vertical spring rate for B truck |
| | C(4) | = viscous damper in parallel with K(4) |
| | MFS4 | = vertical friction damping force |
| | KS4 | = spring in series with MFS4 |
| | K(6) | = total, bilinear, roll moment spring rate for B truck |
| | C(6) | = viscous damper in parallel with K(6) |
| | MFS6 | = angular friction damping moment |
| | KS6 | = roll moment spring constant in series with MFS6 |

**FIGURE 34**
**SLIDING COULOMB DAMPER REPRESENTATION OF TRUCK SNUBBER**

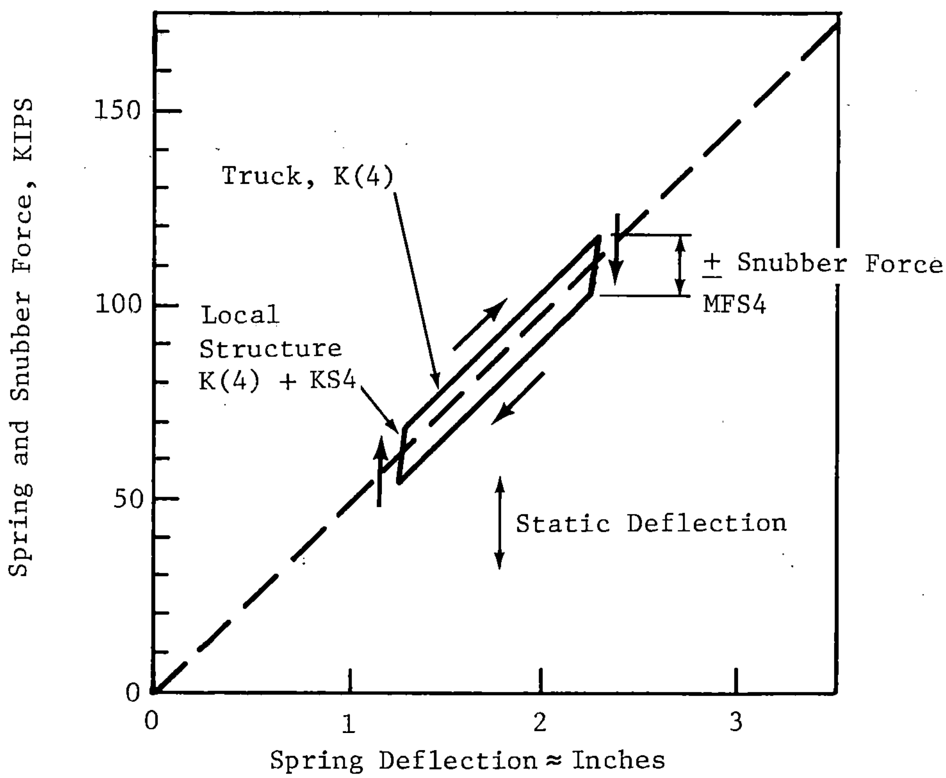



**FIGURE 35**
**FORCE DEFLECTION DIAGRAM FOR TRUCK VERTICAL SPRING WITH SLIDING FRICTION FORCE**

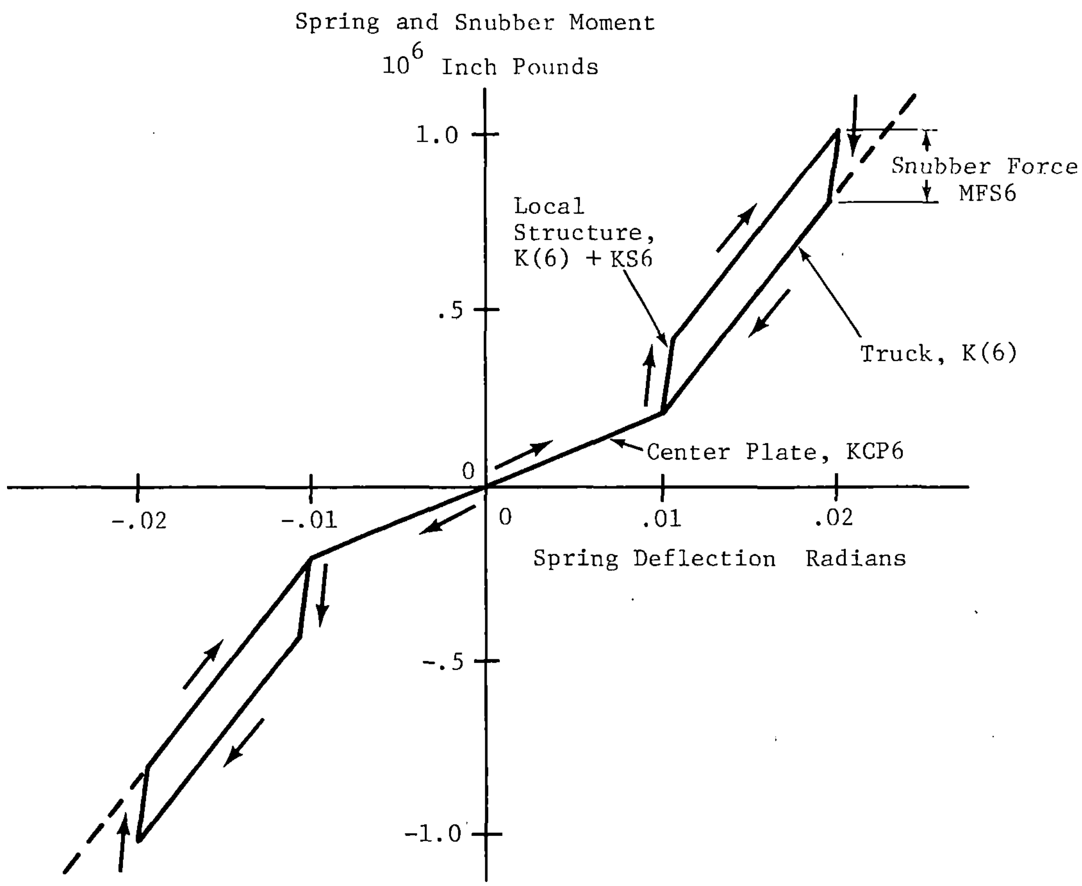



**FIGURE 36**
**MOMENT DEFLECTION CURVE FOR TRUCK AND CENTER PLATE ROLL SPRING WITH SLIDING FRICTION FORCE**

For very light loads, corresponding to the empty condition , the rate was measured to be about 4000 lb./in/ per spring nest, and 9000 lb./in. for the fully loaded condition.

Based on this information the truck transverse springs were varied separately for the empty and fully loaded cars to obtain agreement with test frequencies. The final values obtained were 1800 lb./in. empty and 8000 lb./in. full, per spring nest or 3600 and 1600 lb./in. per truck. 16,000 ?

6.2.4 Other Modifications

There were a number of additional changes made to achieve better correlation for the vertical resonances. First from test results it appeared that there was more deflection in the wheels and bearings than had been modeled for. This was probably because of compliance in the wheel bearings. The wheel spring and damping rates were accordingly lowered and the truck spring rates raised. It was desired to lower the second and third resonance without affecting the first resonance. To accomplish this, three changes were made through a series of exploratory runs. The carbody stiffness was reduced in order to lower the third resonance. This also tended to reduce both the second and first resonance. The rigid body pitch inertia was increased, this causing the second resonance to shift down without affecting either the first or third. The rigid body mass of the carbody was decreased to raise the first resonance, to counter-balance the effect of carbody bending stiffness reduction.

Finally, the number of carbody normal modes was reduced from seven to four. The carbody modes dropped were the second torsion, the second lateral bending and the third vertical bending modes all of which are in the region of 20 Hertz.

### 6.3 Correlation with Modified Model

The following pages presents and discusses comparison between test and analysis of resonant frequencies and associated deflection shapes.

#### 6.3.1 Empty Flatcar Vertical Resonances

Section A.1 of Table VIII lists a comparison of resonant frequencies for this case. The modifications made to FRATE have resulted in improvement in the 2nd, 3rd and 4th resonance while correlation worsened for the 1st resonance. As one would expect, it was not possible to change one resonance without having some effect on each of the others, so that the final configuration represents a compromise on frequency correlation. The only clue as to how to obtain further improvement lies in the two facts that (1) there was coupling between vertical and longitudinal motions in the test which was accentuated by the applied motion being at one end and (2) FRATE does not consider longitudinal motion.

A comparison of the deflection shapes of the first four resonances are presented in Figures 37 through 40. This comparison shows a good, though qualitative, comparison for all four deflection shapes.

#### 6.3.2 Loaded TOFC, Vertical Resonances

Section A.2 of Table VIII lists a comparison between test and analysis for this case. In the pre-validation comparison the third resonance, 3.3Hz, had been aligned with the fourth experimental resonance, 4.803Hz. The third resonance was a subdued resonance and was not detected in the test results until comparison of the 3.3Hz analysis and 4.803Hz test resonance showed poor correlation both as to frequency and deflection shape. With a review of the test data the 3.65Hz resonance was found, however, the amplitude of response were too small to be able to extract a deflection shape.

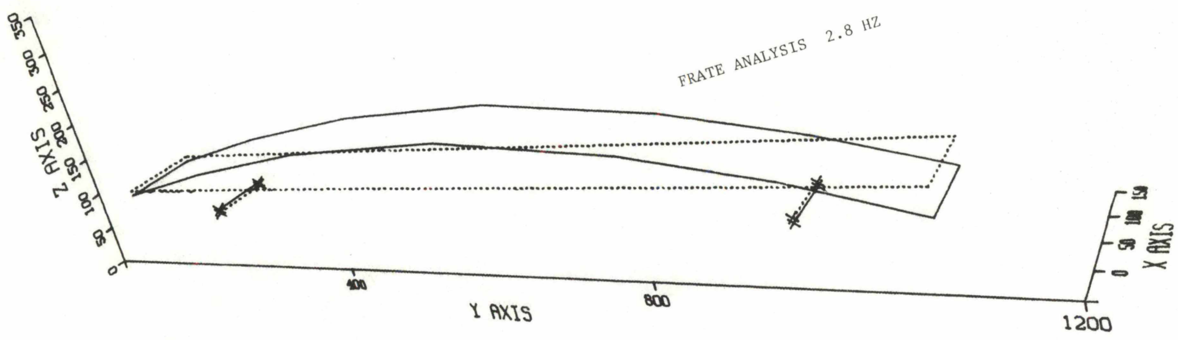
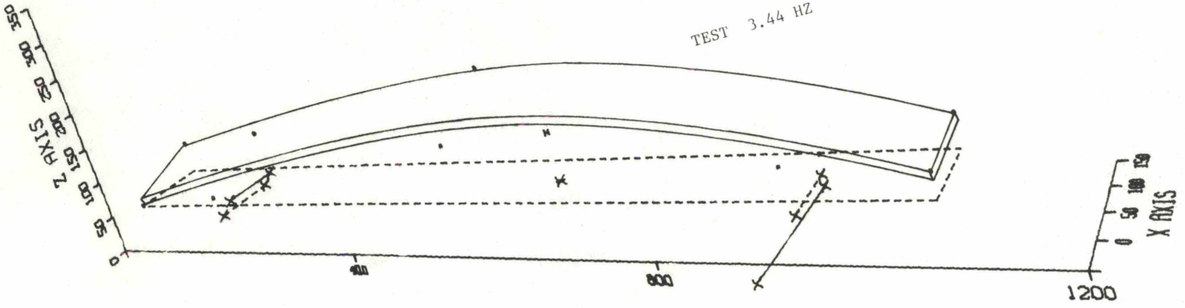
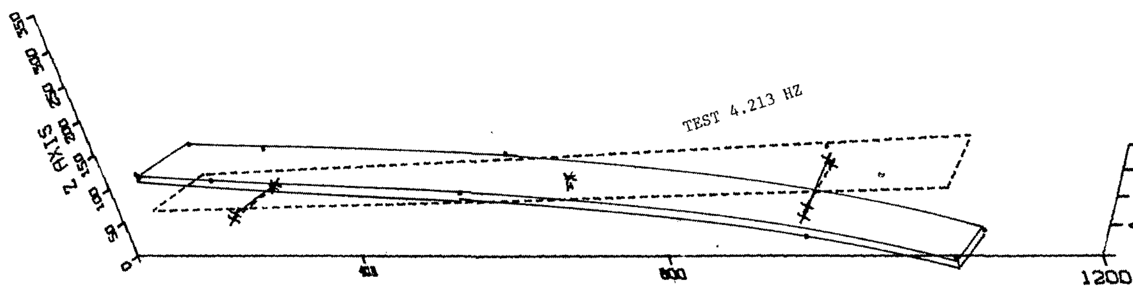



FIGURE 37
POST VALIDATION ANALYSIS TO TEST COMPARISON
EMPTY FLATCAR
FIRST VERTICAL RESONANCE


Z AXIS
0
50
100
150
200
250
300
350
400
800
1200
0
50
100
150
X AXIS

TEST 3.44 HZ



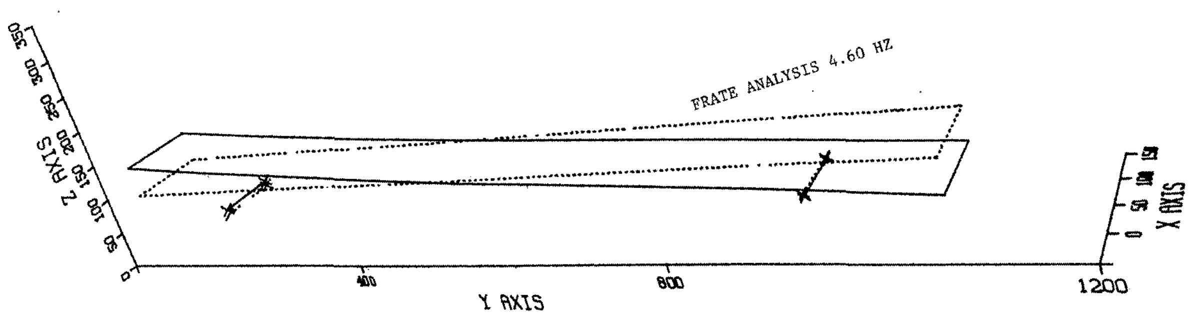
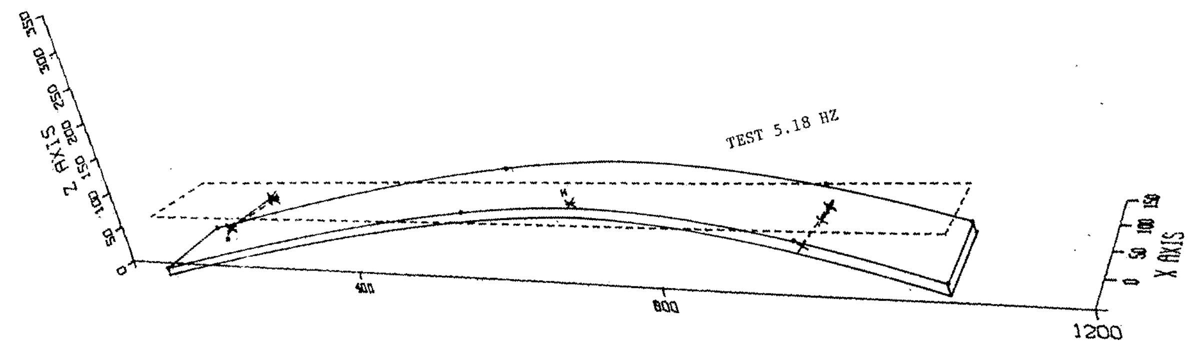




FIGURE 38
POST VALIDATION ANALYSIS TO TEST COMPARISON
EMPTY FLATCAR
SECOND VERTICAL RESONANCE



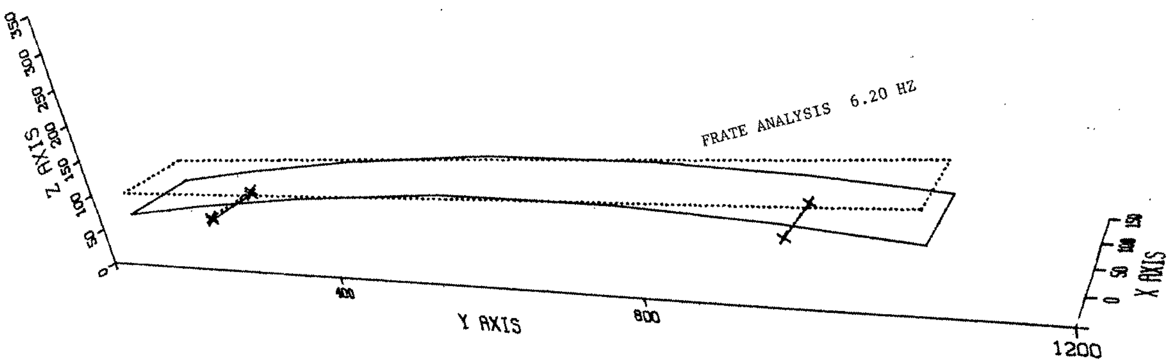
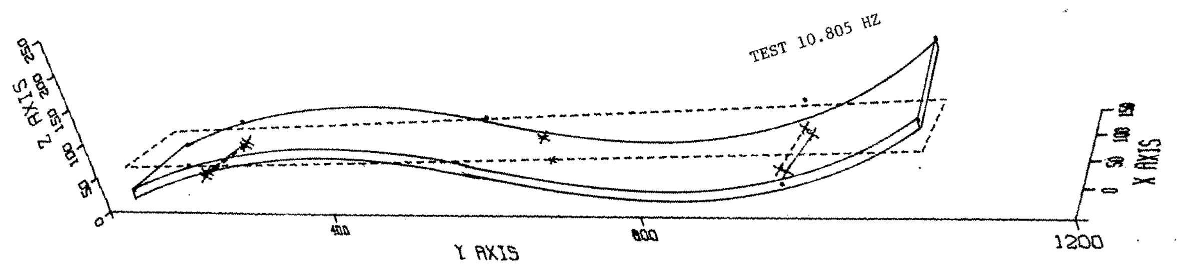




FIGURE 39
POST VALIDATION ANALYSIS TO TEST COMPARISON
EMPTY FLATCAR
THIRD VERTICAL RESONANCE



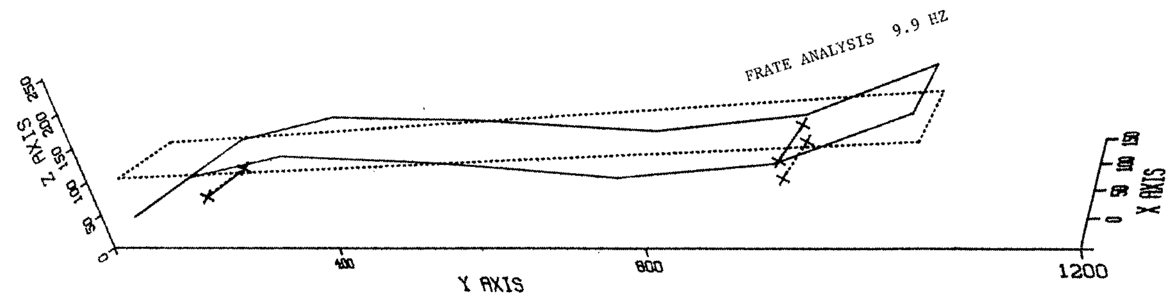




FIGURE 40
POST VALIDATION ANALYSIS TO TEST COMPARISON
EMPTY FLATCAR
FOURTH VERTICAL RESONANCE

TABLE VIII

COMPARISON OF TEST RESONANT FREQUENCIES TO PRE AND POST VALIDATED FRATE RESULTS

VERTICAL RESONANT FREQUENCIES (HERTZ)

A.1 EMPTY FLATCAR

| DESCRIPTION | TEST | ANALYSIS | |
|---|---|---|---|
| | | PRE | POST |
| VERTICAL TRANSLATION WITH BENDING | 3.444 | 3.6 | 2.8 |
| PITCH | 4.213 | 6.7 | 4.6 |
| CARBODY BENDING | 5.185 | 8.1 | 6.2 |
| CARBODY SECOND BENDING | 10.805 | 13.0 | 9.9 |
| CARBODY THIRD BENDING | 17.640 | - | - |

A.2 LOADED FLATCAR (TOFC)

| | | | |
|---|---|---|---|
| COUPLED TRAILER/CARBODY PITCH/BENDING | 1.711 | 1.77 | 1.65 |
| COUPLED TRAILER/CARBODY PITCH/BENDING | 2.238 | 2.2 | 2.3 |
| COUPLED TRAILER/CARBODY PITCH/BENDING | 3.65 | 3.3 | 2.9 |
| COUPLED TRAILER/CARBODY PITCH/BENDING | 4.803 | - | 4.5 |
| TRAILER SECOND BENDING | 7.33 | - | - |
| COUPLED CARBODY BENDING & VAN TRAILER BENDING | 8.844 | - | - |
| CARBODY SECOND BENDING | 11.506 | 12.0 | 9.0 |

ROLL RESONANT FREQUENCIES (HERTZ)

B.1 EMPTY FLATCAR

| DESCRIPTION | TEST | ANALYSIS | |
|---|---|---|---|
| | | PRE | POST |
| ROLL WITH LOW CENTER | 1.184 | 3.8 | 1.16 |
| ROLL WITH HIGH CENTER | 2.000 | 5.5 | 2.22 |
| CARBODY TORSION | 10.114 | 13.3 | 10.0 |
| CARBODY SECOND TORSION | 16.209 | - | - |

B.2 LOADED FLATCAR (TOFC)

| | | | |
|---|---|---|---|
| CARBODY ROLL WITH LOW CENTER | .50-.80 | .87 | .48-.7 |
| CARBODY YAW | 1.131 | - | 1.09 |
| CARBODY ROLL WITH HIGH CENTER | 2.330 | - | 1.9 |
| PLATFORM TRAILER ROLL | 2.681 | - | 2.1 |
| COUPLED TRAILER & CARBODY ROLL | 3.747 | 3.5 | 2.9 |
| COUPLED TRAILER & CARBODY ROLL | 6.222 | - | 6.1 |
| COUPLED TRAILER & CARBODY ROLL | 8.425 | 8.7 | - |
| CARBODY TORSION | 12.936 | 14.3 | 10.4 |

In this case the pre validation FRATE model shows a better comparison to test results than does the post validation model for four resonances. The post validation model is improved in that an additional resonance exists corresponding to the fourth test resonance. The two resonances still missing in the FRATE analysis results are undoubtedly related to trailer bending resonances which would not appear in the analysis since rigid trailer bodies were assumed. The poor correlation in the carbody second bending resonance, analysis at 77% of test, was also attributed to the effect of the rigid trailer body assumption in the analysis.

Deflection shape comparison could not be made on the 3rd resonance due to the lack of test data nor on the 5th and 6th resonance because analysis results were missing. However, for the shapes where comparison could be made, correlation is again seen to be qualatively very good. These are shown in Figures 41 through 44.

### 6.3.3 Empty Flatcar Roll Resonances

The first and second roll resonance of a freight car are characterized by a low center of rotation (i.e., near or below the wheel-rail interface) for the first resonance and a high center of rotation (near the carbody center of gravity) for the second resonance. With the empty flatcar the carbody motions with the first roll resonance are more accurately defined as lateral translation coupled with roll.

It was necessary to reduce the stiffness of the truck lateral springs (K(5) and K(11)) to obtain correlations in the first resonance and to reduce the center plate roll springs (KA6 and K(12)) for the second roll resonance. Variations of these spring rates with car loading condition, (i.e., empty or full) was found to be reasonable from other experimental data. See discussions in Section 6.2 and

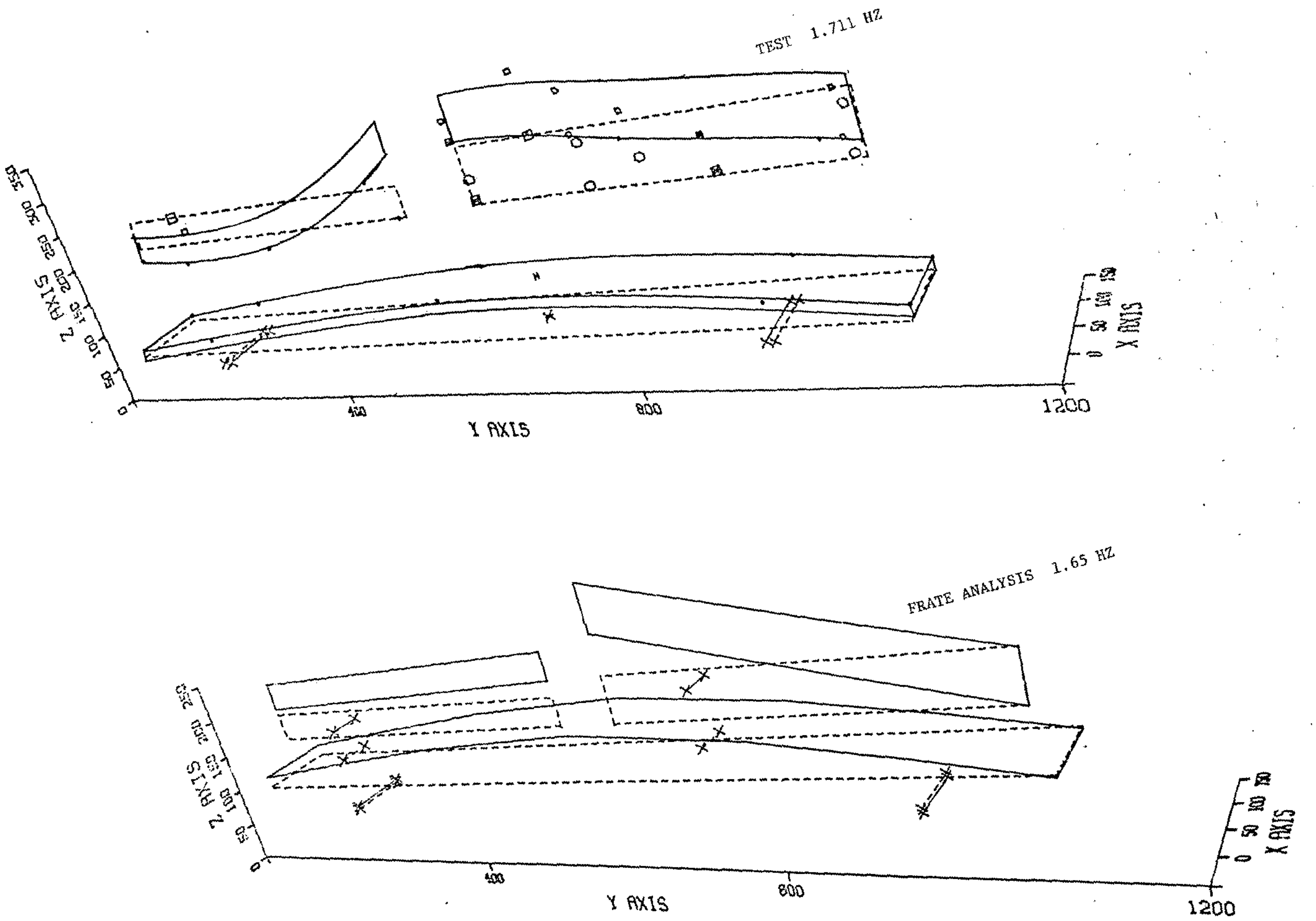



FIGURE 41
POST VALIDATION ANALYSIS TO TEST COMPARISON
TOFC CONFIGURATION
FIRST VERTICAL RESONANCE

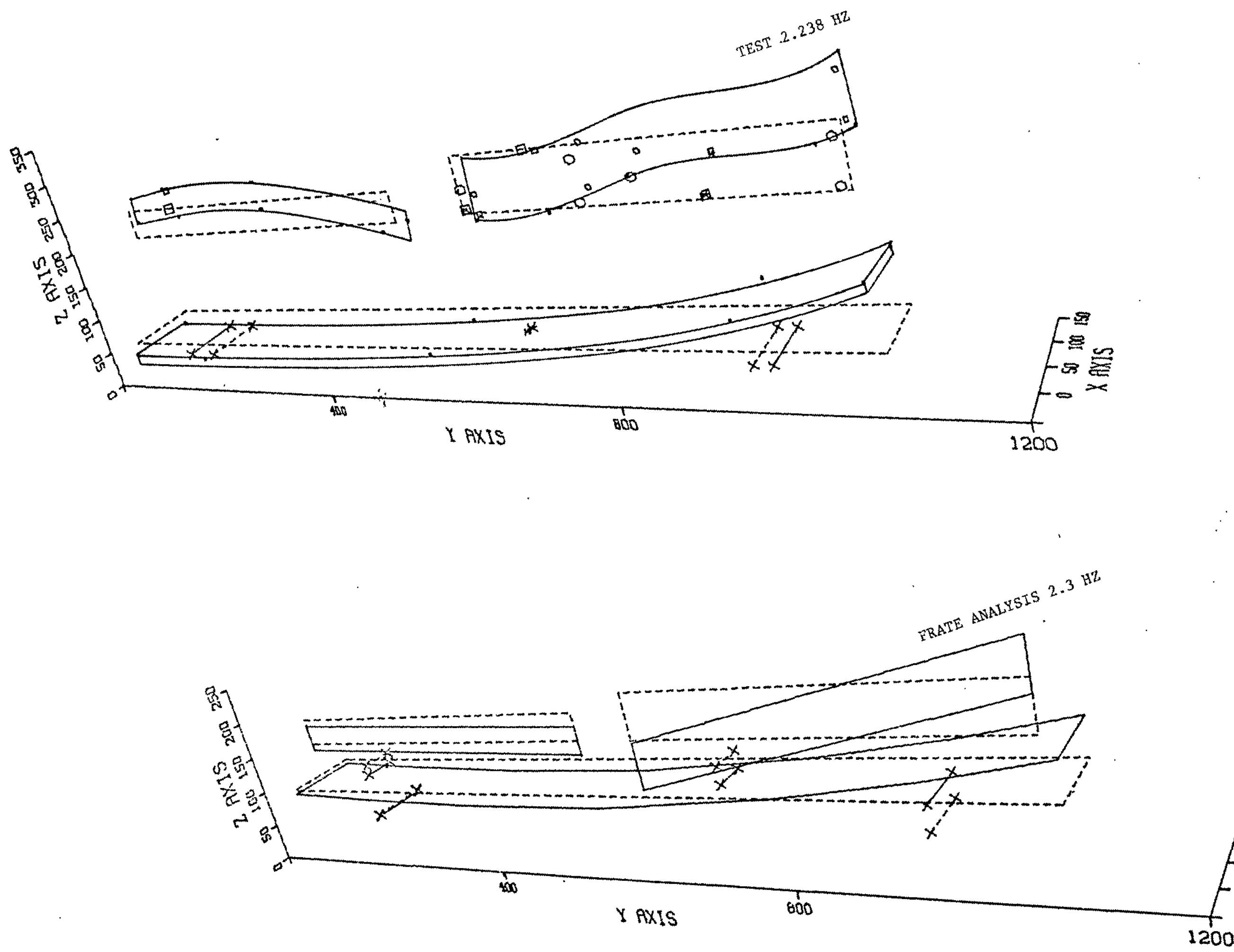



FIGURE 42
POST VALIDATION ANALYSIS TO TEST COMPARISON
TOFC CONFIGURATION
SECOND VERTICAL RESONANCE

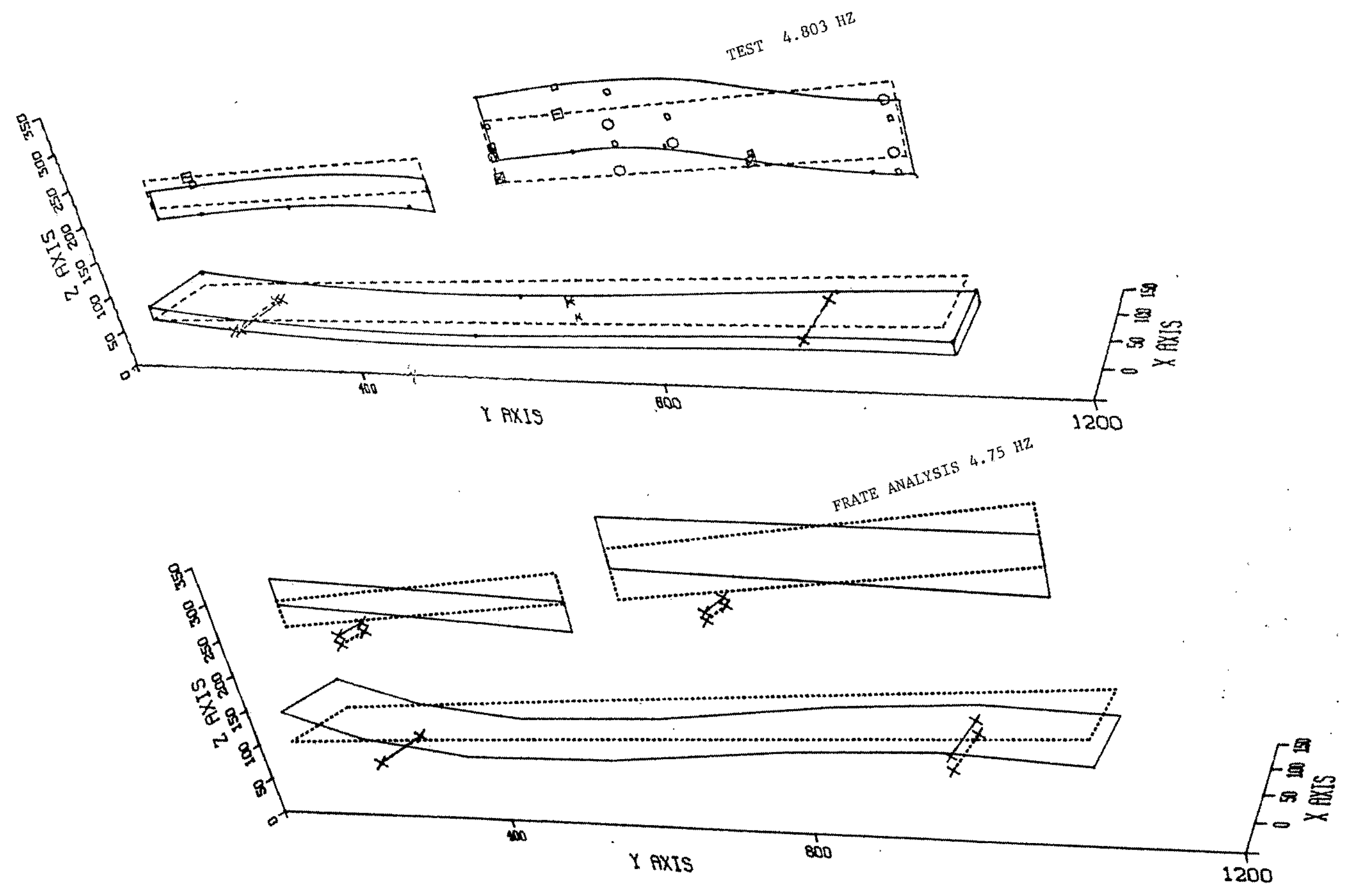
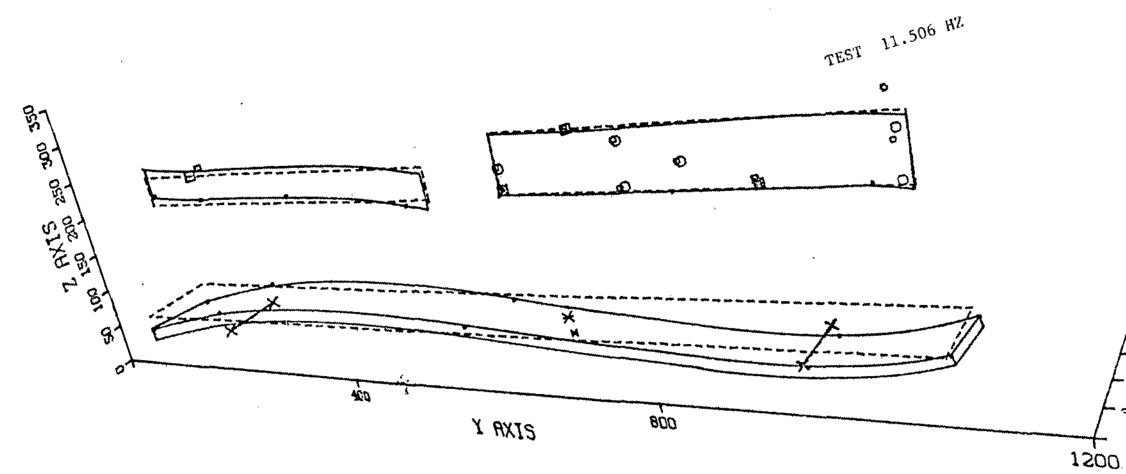



FIGURE 43
POST VALIDATION ANALYSIS TO TEST COMPARISON
TOFC CONFIGURATION
FOURTH VERTICAL RESONANCE



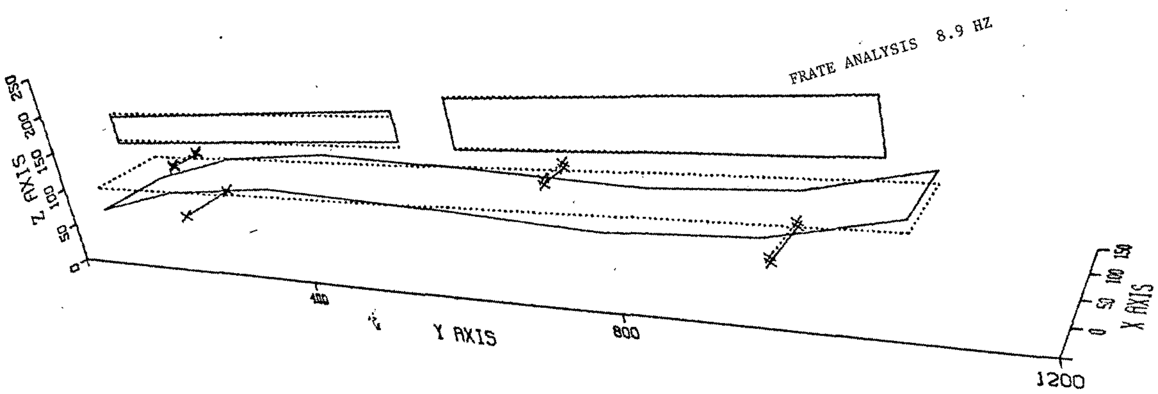




FIGURE 44
POST VALIDATION ANALYSIS TO TEST COMPARISON
TOFC CONFIGURATION
SECOND CARBODY BENDING RESONANCE

Reference 10. The resulting frequency correlation as shown in Section B.1 of Table VIII is seen to be very good, being -2%, +11% and -1% for the first three resonances. The third resonance is carbody torsion and the excellent frequency correlation indicate a good simulation of carbody flexibility.

### 6.3.4 Loaded TOFC Roll Resonance

The loaded TOFC roll resonances were probably the most difficult to identify in both test and analysis. In the test, since excitation was limited to vertical, out of phase excitation at the truck, there were some resonances which were difficult to excite. For example, the carbody yaw resonance at 1.131 Hz. Further for some unfathomed reason the test data did not plot intelligible shapes on the computer. Shape description of the test resonance was based on general motions discerned from studying the test data and relating to deflection shapes obtained by the FRATE analysis.

The final frequency comparison is shown in Section B.2 of Table VIII. The analysis resonance frequencies are seen to be between 77% and 98% if test resonances. Both the vertical and roll resonances in the fully loaded TOFC conditions indicate that an increase in stiffness is needed. The empty flatcar results show that the carbody flexibility is about right. Consequently some increase in truck stiffness would probably effect further improvement in frequency correlation.

## 7.0 PARAMETER VALUES FOR THE FRATE/TOFC SIMULATION

Tables IX, X, XI and XII contain values for all parameters used in the FRATE/TOFC simulation. Where a value has been changed, both pre-validated and post validated values are tabulated.

TABLE IX

TOFC MASS PROPERTIES

| SYMBOL | DESCRIPTION | FINAL VALUE | ORIGINAL IF CHANGED |
|---|---|---|---|
| M(1) | B Truck Mass | 22.33 | |
| M(2) | A Truck Mass | 22.33 | |
| M(3) | Carbody Mass | 125.00 | 136.2 |
| M(4) | Tandem Mass, Van Trailer | 8.179 | |
| M(5) | Van Trailer Mass | 152.70 | |
| M(6) | Tandem Mass, Platform Trailer | 7.013 | |
| M(7) | Platform Trailer | 148.9 | |
| | | | |
| I(1) | B Truck Roll Inertia | .2208E5 | |
| I(2) | A Truck Roll Inertia | .2208E5 | |
| I(3) | Carbody Roll Inertia | .1085E6 | |
| I(4) | Van Tandem Roll Inertia | .2000E5 | |
| I(5) | Van Trailer Roll Inertia | .2550E6 | |
| I(6) | Platform Tandem Roll Inertia | .1715E5 | |
| I(7) | Platform Trailer Roll Inertia | .1020E6 | |
| I(8) | Van Trailer Pitch Inertia | .2940E7 | |
| I(9) | Van Trailer Van Inertia | .3190E7 | |
| I(10) | Carbody Pitch Inertia | .1500E8 | .105E8 |
| I(11) | Carbody Van Inertia | .1500E8 | .105E8 |
| I(12) | Platform Trailer Pitch Inertia | .1310E7 | |
| I(13) | Platform Trailer Van Inertia | .1310E7 | |

Units: $M(I) = \text{lb. sec.}^2/\text{in.}$

$I(I) = \text{lb. in. sec.}^2$

TABLE X

TOFC SPRING AND DAMPER VALUES, FLATCAR TRUCKS

| NUMBER | SPRING CONSTANT K lb/in | VISCOUS DAMPING CONSTANT C lb/sec/in | DAMPING RATIO $C/C_c$ (3) | REPRESENTING |
|---|---|---|---|---|
| 1,3,7,9 | .91E5<br>(.182E6)† | 300.<br>(250) | .020 | Side frame, wheels and track, vertical, 2 per truck |
| 2,8 | .95E5 | 333. | .022 | Side frame, wheels and track, lateral, 1 per truck |
| 4,10<br>(was 6, 12) | .48E5<br>(.225E5) | 1400. (1)<br>140. (2)<br>(1400.) | .183<br>.018 | Truck spring vertical, |
| A6, A12* | .21E8 (4)<br>.114E8 (5) | .20E6<br>.10E6 | .060<br>.055 | To represent center plate rocking |
| B6, B12*<br>(Replaces K(6), K(12)) | .6185E8 | .20E7 (1)<br>.78E6 (2) | .203<br>.079 | To represent truck roll with side bearings in contact |
| 5,11 | .16E5 (4)<br>.36E4 (5)<br>.10E6 (6)<br>(.10E6) | 200.<br>45.<br>1250.<br>(1400) | .079 | Truck lateral, 1 per truck |

*Angular spring and dampers, units are inch pounds per radian and inch pound seconds per radian.

(1) 100% snubbers in truck springs.
(2) 0% snubbers in truck springs.
(3) $C/C_c$ is obtained by using $C_c = k/\pi f$ and, assuming $f = 2.0$ Hertz.
(4) Fully loaded TOFC.
(5) Empty Flatcar.
(6) Gib contact.
† Values in original model

TABLE XI

TOFC SPRING AND DAMPER VALUES, HIGHWAY TRAILERS

| NUMBER | SPRING CONSTANT K lb/in | VISCOUS DAMPING CONSTANT C lb/sec/in | DAMPING RATIO $C/C_c$ (3) | REPRESENTING |
|---|---|---|---|---|
| 13,21 | .225E6 (.10E7)† | 1000. | .028 | Trailer hitch vertical |
| 14,22 | .150E5 (.135E6) | 200. (500) | .084 | Trailer hitch lateral |
| 15,17,23,25 | .225E5 (.24E5) | 300. (1000) | .092 | Tandem tires, vertical, 2 per tandem |
| 16,24 | .180E5 (.10E6) | 200. (1000) | .070 | Tandem tires, lateral 1 per tandem |
| 18,20,26,28 | .5276E5 (.264E5) | 775. (500) | .092 | Tandem springs vertical, 2 per tandem |
| 19,27 | .180E5 (.10E6) | 200. (1000) | .070 | Tandem springs lateral, 1 per tandem |
| XZMOM* XRMOM | .30E8 (.113E9) | .10E6 (.6E5) | .021 | Trailer hitch, roll moment |

*Angular spring and dampers, units are inch pounds per radian and inch pound seconds per radian.

†Values in original model.

TABLE XII

TOFC DIMENSION VALUES

(Units are inches except as noted)

R(1) = R(3) = 58.00

R(2) = R(4) = 79.00

R(5) = R(7) = 62.25

R(6) = R(8) = 43.50

| | |
|---|---|
| L = 792.0 | |
| VL1 = 469.0 | VL1R = -89.0 |
| VL2 = 148.0 | VL2R = -413.0 |
| VL3 = 131.4 | VL3R = 115.4 |
| VL4 = 189.7 | VL4R = 208.6 |

| | |
|---|---|
| H = 16.0 | |
| VH = 60.4 | VHR = 37.8 |
| VH1 = 47.0 | VH1R = 47.0 |

OR(1) = 536.0

OR(2) =-536.0

OR(3) = -39.0

OR(6) = 224.0

OR(7) = 226.0

OR(8) = 254.0

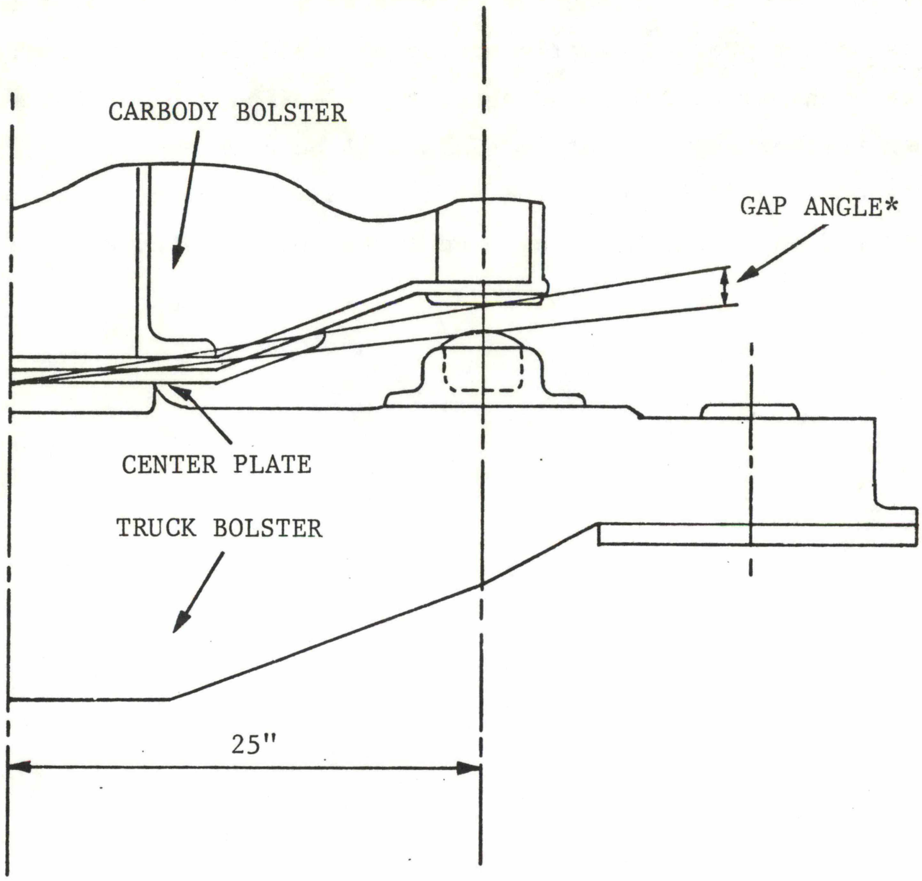
OR(9) = 245.0

OR(10)= 235.0

GAPB = .01

GAPA = .01

Note: GAP angle is based on side bearing clearance at 25 inches from carbody centerline and assumes equal gap on each side. (.01 radians = .25 inch gap.) See Figure 45.




*Roll motion is assumed to be about a longitudinal axis at the center of the carbody.

**FIGURE 45**
**SIDE BEARING GAP ANGLE**

## 8.0 CONCLUSION

The validation effort for the FRATE computer program containing flatcar and TOFC simulations has culminated in showing that the analysis does reproduce experimentally measured dynamic properties along the general guidelines presented in Section 2 of this report. Although further improvement in FRATE will be continued to be sought, the program with TOFC simulation as revised in this report is presented as validated in so far as being used in the design studies of freight cars.

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