This is to certify that the

thesis entitled

DYNAMIC ANALYSIS OF A LESS
THAN TRUCKLOAD SHIPMENT

presented by

Jorge Aparecido Marcondes

has been accepted towards fulfillment
of the requirements for

M. S. degree in Packaging

S. Paul Singh
Major professor

S. Paul Singh, PhD.

Date May 12, 1988

DYNAMIC ANALYSIS OF A LESS THAN TRUCKLOAD SHIPMENT

By

Jorge Aparecido Marcondes

A THESIS

Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

School of Packaging

1988

ABSTRACT

DYNAMIC ANALYSIS OF A LESS THAN TRUCKLOAD SHIPMENT

By

Jorge Aparecido Marcondes

The effect of shock and vibration forces in the distribution environment has been studied extensively by engineers. This study investigated the dynamics of three different package systems in a Less Than Truckload (LTL) shipment.

A truck was positioned on vibration actuators and a time domain input simulating road data was used as the drive signal. Front and rear truckbed RMS accelerations were measured. Three different resonant frequency (low, medium and high) packages were positioned at the top of front and rear stacks and their RMS acceleration were recorded.

The results show that the rear of the truck is not always the worst position for acceleration magnification in LTL shipments. Accelerations as high as 10 g´s were encountered during vibration. The front axle constributes about 20% and the fifth wheel about 30% of the vibration input to the rear truckbed.

This thesis is dedicated to

my parents	Benedicto and Mercedes
my wife	Patricia

ACKNOWLEDGMENTS

I wish to express my sincere gratitude to my major professor Dr. S. Paul Singh for his guidance and friendship throughout this study. I am also very grateful to the members of my committee, Dr. Gary J. Burgess and Dr. George E. Mase.

I also extend my appreciation to Fruehauf Corp. and Yellow Freight Systems, Inc., for providing laboratory facilities and collecting data during the experiments. I appreciate the support of Dr. Harold A. Hughes for the use of facilities at the School of Packaging. Thanks to Dr. James Goff, Dr. Diana Twede and Mr. Jeff Waldeck for providing materials used in the experiments.

Grateful acknowledgment is extended to Ms. Akemi Suzuki and Mr. Mituru Mori, Embalagens Jaguare, Brazil, and to FAPESP, Fundacao de Amparo a Pesquisa do Estado de Sao Paulo, Brazil, for the initial support. To CAPES, Coordenacao de Aperfeicoamento de Pessoal de Nivel Superior, Brazil, my appreciation for completing the financial support.

Finally, I would like to thank all of those who helped in one way or another during the course of this work.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure		Page

1.0 INTRODUCTION

The effects of shock and vibration forces on packages in the distribution environment system have been studied extensively by engineers. Although there are various different ways of transporting goods from one place to another, trucks and trains are the most common means of bulk surface transportation of most packaged goods. To evaluate the protective role of packaging systems, measurements of the environment (roads, rails, etc.) have been done to quantify the input shock and vibration levels. Based on environmental conditions and a knowledge of the fragility of the product to be protected, packages may be designed to isolate the product from shock and vibration.

All packaged products go through several handling and transportation environments before reaching the end users. Both human and mechanized handling such as conveyor belts, material handling equipment, trucks, ships, aircraft and trains are some examples. Focusing on trucks which are the main carriers, the origin of all shock and vibration comes from two sources:

1) External sources such as road or surface irregularities, braking, and forward acceleration;

2) Internal sources due to the vehicle itself such as engine vibration, drive mechanisms, and wheel unbalance. (Harris, 1961).

Therefore, not only the route but also the type of vehicle to be used plays a fundamental role when designing an optimum package for shock and vibration protection. In addition, the size and position of the load within the truck changes the dynamics of the truck and therefore the response of the package inside the truck. This is important since many commodities are shipped by Less Than Truckload (LTL) shipments from regional distribution centers to sale outlets.

Objectives:

The purpose of this study was to understand the dynamics of a loaded truck and how they effect packages in a LTL shipment. The specific objectives were as follows:

1) Quantify the response of the truck suspension to a given vibration input.

2) Observe the effect of the truckbed vibration on instrumented packages to determine vibration transmission levels.

3) Establish relationships between package resonance and packages position within the truck.

4) Compare the results for LTL shipments from this study to Full Truckload (FTL) shipments based on prior research.

5) Compare contributions of each axle or wheel to the overall acceleration of the package.

The above objectives were chosen to raise and answer some questions which have not yet been considered in LTL shipments. The results show that some previously held ideas are incorrect and that there are dynamic phenomena associated to LTL shipments as opposed to FTL shipments which may be the cause of increased damage.

2.0 REVIEW OF LITERATURE

There have been many studies aimed at uncovering the primary features of shock and vibration that relate to product damage during transportation using common carriers. Several of these studies use analytical concepts such as RMS acceleration, magnification factors and Power Spectral Density (PSD) plots. In order to hold the literature review to a non-mathematical format, the reader is referred to Appendices A and B for the mathematical details of these concepts. Also, since there appears to be some overlap in the areas studied, no attempt will be made to present the literature in an order which suits the development of analytical ideas. Rather, the literature will simply be presented in chronological order.

The studies, Preliminary Analyses of Data Obtained in the Joint Army/AEC/Sandia Test of the Truck Transport Environment (Foley, 1960) and The Environment Experienced by Cargo on a Flatbed Tractor-Trailer Combination (Foley, 1966) were undertaken to gather and analyze data on heavy load shipments. These tests were aimed at determining the dynamic environment for truck shipments using a 15-ton radioactive cask as the cargo and at developing techniques for the reduction, presentation, and analysis of data. The results were given in PSD plots for blacktop and concrete highways at 35 and 50 mph, at a location 24 in. to the rear of the cask.

Harris and Credes (1961) presented concepts and definitions related to shock and vibration levels encountered by road and rail vehicles. A table of typical shock and vibration data for tractor-trailer combinations under normal operating conditions was presented. Similar data for trucks and trains were also published.

The Dynamic Environment of Spacecraft Surface Transportation (Schlue, 1966) studied shock and vibration characteristics for the purpose of designing transportation vehicles for spacecraft. The objective of this work was to measure shock and vibration forces over both rough, irregular roads and smooth highways. Comparisons were made for vans on three trips over smooth and rough roads, and the conclusions reached were:

a) Air-suspension systems are adequate for spacecraft shipment.

b) Equipment should be hard mounted to the floor, especially if dynamic characteristics of support structures or shock mounts are not known.

c) The position on the truckbed corresponding to the lowest vibration input was a function of frequency, individual van characteristics, and to some extent, input excitation amplitude. The aft and center positions are the most consistent with respect to vibration levels (aft highest, center lowest).

d) Vibration levels at frequencies greater than 100 Hz were insignificant.

Preliminary Measurement and Analysis of the Vibration Environment of Common Motor Carriers (Sharpe et al, 1974) was a study aimed at assessing the nature of the dynamic vibration environment of commercial motor carriers carrying package loads. Some conclusions reached were:

a) Vertical accelerations were the most severe and therefore the only ones that need be measured.

b) Simply enveloping PSD data was equivalent to selecting the worst possible conditions and therefore may be an excessively severe test requirement. Some type of product validation testing at various PSD levels for specified periods of time would be better.

The objectives of the Joint Services Highway Shock Index Project (Grier et Al, 1975) were to conduct static loading and dynamic impact tests on a representative series of commercial highway cargo trucks to obtain data for the shock levels transmitted to the cargo bed and to analyze the static and dynamic data collected from these tests to develop a method for determining the shock index of commercial cargo trucks.

The results of the study were:

a) A practical graph method which uses planned payload(s) and vehicle payload axle spring rates, was developed for determining the shock index of commercial cargo trucks.

b) For a two-axle cargo-truck, the roughest ride on the truckbed was over the rear axle. For a truck-tractor-semitrailer combination, the roughest ride occurred either over the rear axles of the trailer or over the tractor rear axles (fifth wheel), depending on which axle had the higher payload.

c) The tests showed that of the three major variables (percentage of maximum payload, tire pressure, and speed), percentage of maximum payload had the greatest influence on the shock index. Tire pressure and speed were relatively unimportant.

d)For highway travel, vertical accelerations were generally greater than either lateral or longitudinal accelerations and were a major factor in potential cargo damage.

e) High, erratic shock values occurred with either very light or very heavy payloads. The most erratic results occurred in the area of the fifth wheel.

f) Under maximum load conditions, independent of load location, tire pressure, and truck speed, vertical accelerations exceeding 10 g's were recorded on several occasions by each of the forward, middle and rear impact registers when the test vehicles ran over test bumps.

g) The shock index graph may be employed to define practical shock parameters for selecting cargo trucks based on ride performance and for preparing cargo truck specifications or standards.

Advances in Shipping Damage Prevention (Caruso and Silver 1976) was another study aimed at observing cargo losses in tractor-trailers for different suspension systems, degrees of loading, rear wheel positions, road types, and drivers. Their conclusions were:

a) The results did not necessarily apply only to heavy cargo rigidly attached to the trailer bed.

b) Similar suspension systems on different trailers produced similar responses.

c) PSD levels at frequencies greater than 50 Hz were negligible.

d) Each suspension type had a different power density at the first frequency peak (about 5 Hz) but similar power density spectra beyond the second peak (about 13 Hz).

e) Single-leaf steel suspension springs produced the worst ride conditions.

f) The worst ride occurred in a lightly loaded trailer over the rear wheels.

g) The worst ride for typical roads occurred during high-speed driving on interstate highways.

h) Different drivers had little effect on the overall results.

Shock and vibration environment studies for large shipping containers with heavy cargo during truck transportation were undertaken by Magnuson (1977 and 1978). The results of his work are:

a) The vibration history observed was random and had a normal Gaussian distribution with respect to acceleration levels.

b) The highest vibration level was generally always in the vertical direction.

c) Although the vertical acceleration was almost always the most severe, there were exceptions unique to the vehicle and its operating conditions.

d) Shipments weighing more than 15 tons showed little difference in vibration amplitude regardless of the type of suspension system.

The objective of the study, Shock and Vibration Environment in a Livestock Trailer (Turczin et al, 1980) was to ascertain the shock and vibration environment on the floor of a livestock trailer typical of those used in the shipment of cattle. Some conclusions reached were:

a) Vibration levels above 40 Hz were insignificant.

b) The highest energy levels were produced in the vertical direction (peaks were measured to 0.08 g rms).

Tevelow (1983) characterized the military logistical transportation vibration environment with respect to the shipment of fuzes using various types of vehicles (truck, sea, rail and air). His study summarizes twelve previous reports on trucks.

Pichler (1984) presented condensed information on dynamic environment data used in Brazil. The measurements were done on the most important highway connecting the Northwest to Southern part of the country where the highest volume of goods are transported.

Goff et al (1984) reported vertical disturbances caused by large amplitude transients. Accelerometers mounted on both packaged products and the truckbed itself monitored the degree of vibration magnification. Data was taken from a half-hour test over city and county roads, interstate expressways, bridges, and railroad grade crossings. However, only accelerations in the rearmost portion of the trailer were recorded. Three different suspension systems were studied:

a) moveable leaf spring tandem axle trailer with axle at rear portion;

b) same as a), but axle at forward position;

c) fixed position air-ride tandem axle trailer.

The study was divided into two parts using different cargo: cartoned freezers in the first part and uncartoned furniture in the second part.

American President Companies (1986) concluded a study in December 1986 describing the double-stack/truck/vessel ride characteristics. Accelerations were recorded while the products were transported by ship, APL stack train, and truck trailer. Instruments were positioned so that the acceleration of each container and product could be recorded. An overview of the results of the test revealed that vessel transportation generated the lowest acceleration levels during the entire trip. The double-stack intermodal cargo transportation system proved to be the smoothest of the available inland transportation modes. Trucking remained the highest vibration environment among the tested modes of transportation. The study states: "A key factor emerging from the tests results is that, in the movement of this type of cargo, the loading of the container itself is of prime importance. All packages should be tightly loaded from front to rear of the container in such a manner that no movement of the cargo is allowed. This will reduce the levels of acceleration experienced by the product to the same levels by the container, thereby further improving the ride quality". Results were presented in graphs of G level as function of the percentage of occurrence for the following situations:

a) Cargo performance in a double stack railcar (combined, lateral, longitudinal and vertical acceleration).

b) Cargo performance in a truck (combined acceleration).

c) Container performance trucking (lateral, longitudinal and vertical acceleration).

d) Cargo performance under handling & stowage (combined, lateral, longitudinal and vertical acceleration).

Goodwin and Holland (1987) studied the rail distribution environment from Rochester, NY, to Los Angeles, CA, via Chicago, IL. They took into consideration two modes of shipment, Trailer on a Flat Car (TOFC) and Container in a Well Car (CIWC). A change in the mode from TOFC to CIWC was made in Chicago. This intermodal shipping channel generated dynamic data on shock and vibration which was analyzed and used as a basis for packaging lab test simulation. The end result was a laboratory preshipment test using the random vibration characteristics of this route to simulate the dynamics of other distribution channels. Power Spectral Density (PSD) plots were developed for each of the three axes, longitudinal, lateral and vertical for the same loading in both transportation modes. Percentages of shocks measured were then reported for similar loading in both modes and four directions, longitudinal, lateral and two vertical axis (rear and front).

This study is aimed at expanding the information base on LTL shipments, specifically on the effects of truck vibration on low, medium and high natural frequency packages. A knowledge of the effect of the truck vibration on the different dynamic characteristics of packages is crucial for planning LTL shipments with emphasis on arranging packages in the truck to minimize transportation damage.

3.0 EXPERIMENTAL DESIGN

The experiment designed to meet the objectives of this study was set up at the laboratories of Fruehauf Corporation located in Detroit, Michigan. All of the data for this study was collected at the Fruehauf laboratories using a vibration system consisting of six independent vibration actuators positioned under the truck wheels to vibrate the truck and simulate road conditions. (Figures 6, 7 10, 11 and 12). Each wheel and axle is referred according to its position in the truck as described later in the text. As the vehicle travels over the road, irregularities in the surface create independent excitation inputs at the individual wheels, which may be vertical, side to side (transverse) or front to back (longitudinal). In this experiment, only vertical excitations were used since earlier studies show these to be the most critical.

3.1 Package Systems:

The description of each package tested on the truck is as follows:

Package A: (Figure 1)

Corrugated fiberboard box, 8.75" x 8" x 10.50", containing a concrete brick weighting 11.5 lbs, with cushioning material, having a natural frequency of 9 Hz.

Package B: (Figure 2)

Corrugated fiberboard box, 12.75" x 10" x 9.50", containing 23 cans of tomatoes. The 24th. can was replaced by a wooden block containing an accelerometer. This block was placed in one of the corners of the box. Package B has a total weight of 28 lbs and natural frequency of 21 Hz.

Package C: (Figure 3)

Wooden block measuring 9.75" x 8.75" x 8.75", weighing 15.5 lbs and having a natural frequency in the range 64 to 74 Hz. The accelerometer was located in the center of the wooden block.

3.2 Truck System:

The truck used in this experiment was a tractor/trailer type provided by Yellow Freight Systems, Inc. Figure 4 shows the particular truck used in the experiment and Figure 5 shows the location and designation of the axles and wheels. The truck's main characteristics are as follows:

Dry freight Doubles Van Trailer:

Manufacturer: Fruehauf Corp.

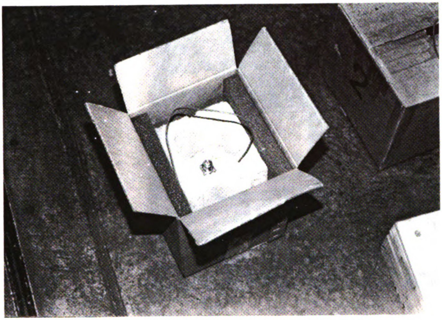


Figure 1. Package System A

Figure 2. Package System B

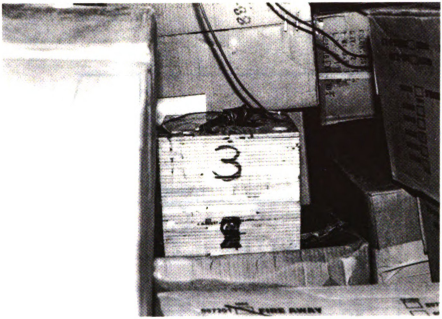


Figure 3. Package System C

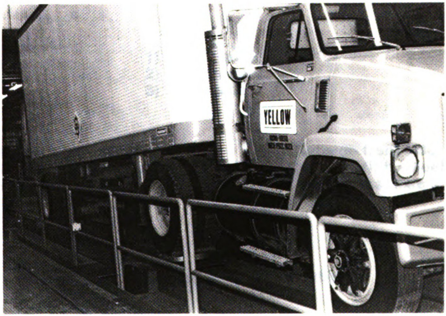
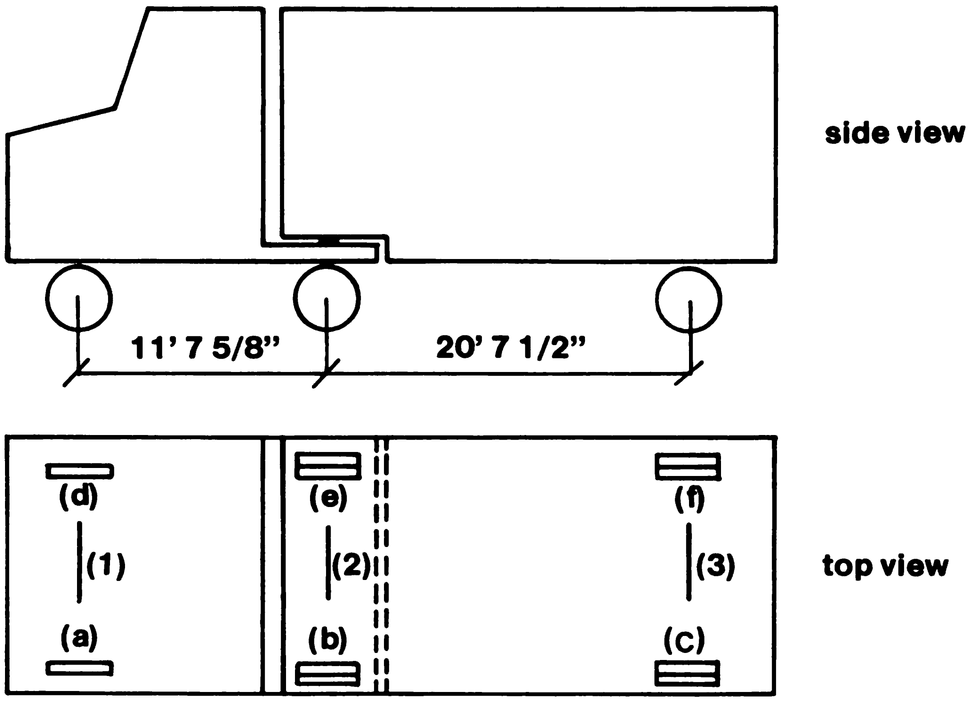


Figure 4. Tractor/Trailer Truck System



1) Front axle
2) Fifth wheel
3) Rear axle

a) Roadside front axle
b) Roadside fifth wheel
c) Roadside rear axle

d) Curbside front axle
e) Curbside fifth wheel
f) Curbside rear axle

Figure 5. Position of axles and wheels

Model No.:	FBBX - F1 - 27
Serial No.:	1H2V02714BE003223
Date of Manufacture:	January, 1981
Tires:	11" x 24.5" ø
Single Axle Tractor:	
Manufacturer:	General Motors Corp.
VIN:	IGTM9C1W6HV511593
Max load at front (rated):	9,000 lb
Max load at rear (rated) :	20,000 lb

3.3 Actuator System:

The vibration system consists of six independently controlled vibration actuators, each of which restricts the wheels from moving back and forth and side to side, thereby permitting only vertical motion. Figure 6 shows a picture of one wheel placed on an actuator and Figure 7 shows the back of the truck and the positioning of the rear wheels over their respective actuators with interlocking restraining devices. The drive signal used to control the actuators replicates six minutes of driving at approximately 40 mph on Interstate Highway I-94 in the Detroit area.

3.4 Accelerometers:

Strain gage accelerometers were used to monitor accelerations. For each experimental run, four

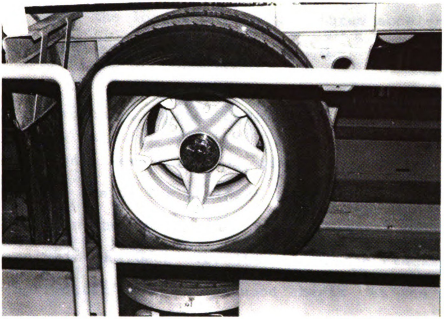


Figure 6. Curbside rear wheel on its actuator

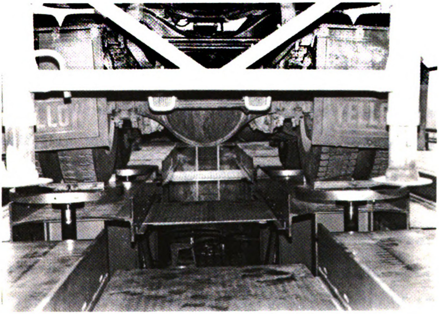


Figure 7. Truck positioned on the actuator system (rear view)

accelerometers were used. One was mounted under the truckbed just below the load and the other three accelerometers were installed inside packages A, B and C.

3.5 Instrumented Package Positions:

Measurements were made with stacks located in two different positions inside the trailer:

1) only in the front, over the fifth wheel with no stowage.
2) only in the back over the axles; the stacks were stowed by horizontal bars and plywood sheets at the front and by the doors at the very back. These package arrangements are shown in Figures 8 and 9.

3.6 Experimental Set up:

a) In the front of the trailer, boxes A, B and C were positioned at the top of the LTL shipment as shown in Figure 12. The LTL shipment consisted of various kinds of packages ranging in weight and size. Figure 10 shows the location of the LTL shipment in the front of the truck.

b) In the back of the trailer, the packages were positioned in the same way as described in Figure 12. Figure 11 shows the instrumented packages located in the back LTL shipment.

Figure 8. Packages in the front (before testing)

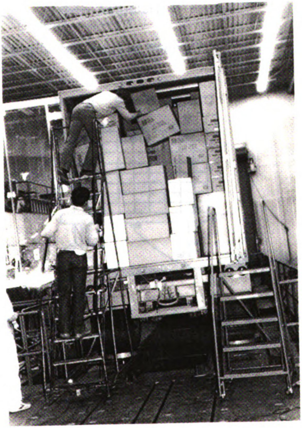


Figure 9. Packages in the back (before testing)

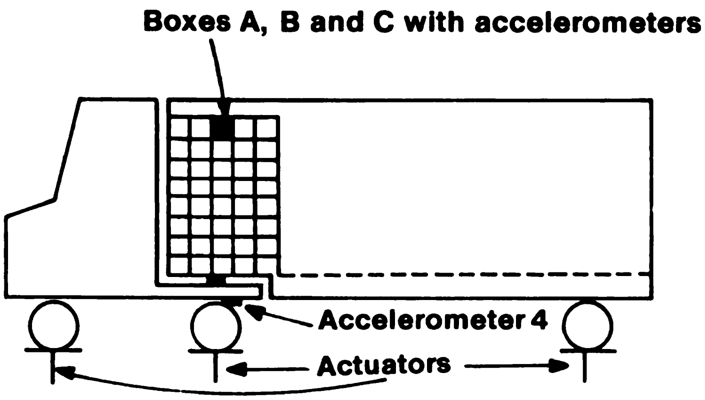


Figure 10. LTL shipment (front of trailer)

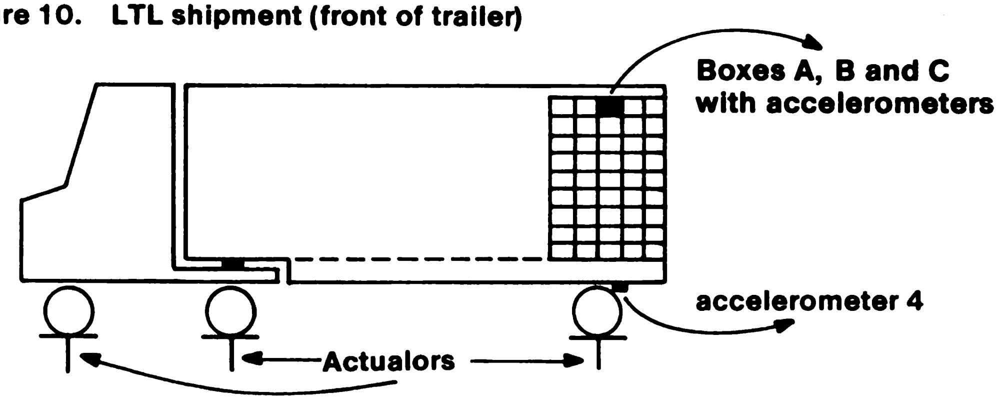


Figure 11. LTL shipment (rear of trailer)

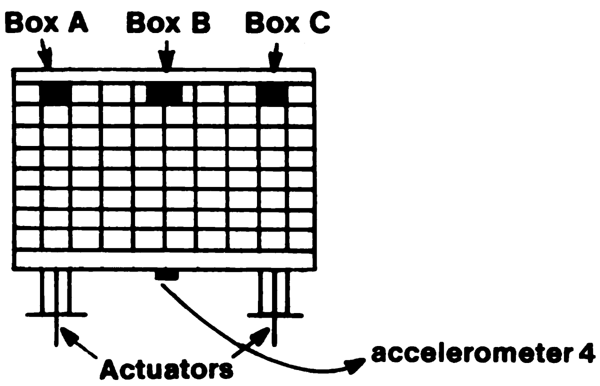


Figure 12. Positioning of instrumented packages in LTL shipment

Figures 10,11 and 12 also show the location of the control accelerometer located underneath the truckbed during different simulations.

4.0 DATA AND RESULTS

4.1 Vibration Input:

Table 1 describes the location of the six recording channels relative to their respective actuators and Table 2 shows the statistics of the input drive signal amplitude. Refer to Appendix A for a description of the terms used. The data presents the mean, and maximum and minimum amplitude of each actuator during the 350 seconds of run time. The average distribution function of occurrences as a function of actuator amplitudes (Figures 13 and 14) shows an approximately normal Gaussian distribution.

Table 3 shows RMS stroke values determined for different locations at frequencies that generated high RMS values. Figures 15, 16 and 17 show the RMS stroke as a function of the frequencies of actuators.

It is evident that the roadside of the trailer generally produces larger maximum and minimum values for input amplitudes than the curbside at high frequencies. We can say in general that for this particular road, there were more severe vibration levels for the roadside axle. All input

Table 1. Location of Recording Channels.

Channel	Wheel
1	Roadside front axle
2	Roadside fifth wheel
3	Roadside rear axle
4	Curbside front wheel
5	Curbside fifth wheel
6	Curbside rear axle

Table 2. Amplitude of Input Drive Signal (Inches).

Channel	Mean (μ)	Maximum	Minimum	Std. Dev. (σ)
1	2.51E-04	1.62	-1.07	0.158
2	-6.37E-05	1.86	-1.34	0.181
3	3.39E-04	2.96	-1.66	0.159
4	-9.71E-04	1.33	-1.44	0.178
5	2.36E-04	1.22	-1.22	0.139
6	4.44E-05	1.47	-1.54	0.157

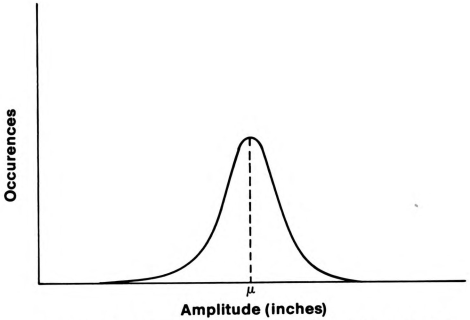


Figure 13. Normal distribution of input drive signal

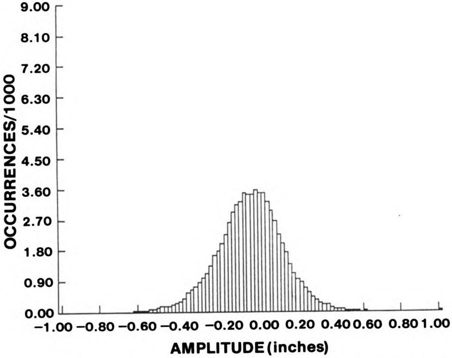


Figure 14. Histogram of Input Drive Signal

Table 3. RMS Stroke Values Versus Frequency.

Frequency (Hz)	Location	RMS stroke (inches) Roadside	Curbside	Average
1	Front axle	0.075	0.091	0.083
	Fifth wheel	0.084	0.062	0.068
	Rear axle	0.060	0.063	0.062
6	Front axle	0.021	0.010	0.016
	Fifth wheel	0.028	0.028	0.028
	Rear axle	0.023	0.027	0.025
12	Front axle	0.006	0.005	0.006
	Fifth wheel	0.009	0.005	0.007
	Rear axle	0.009	0.007	0.008
18	Front axle	0.005	0.003	0.004
	Fifth wheel	0.008	0.004	0.007
	Rear axle	0.006	0.003	0.005
24	Front axle	0.004	0.002	0.003
	Fifth wheel	0.003	0.003	0.003
	Rear axle	0.004	0.002	0.003
30	Front axle	0.002	0.002	0.002
	Fifth wheel	0.002	0.002	0.002
	Rear axle	0.003	0.002	0.003

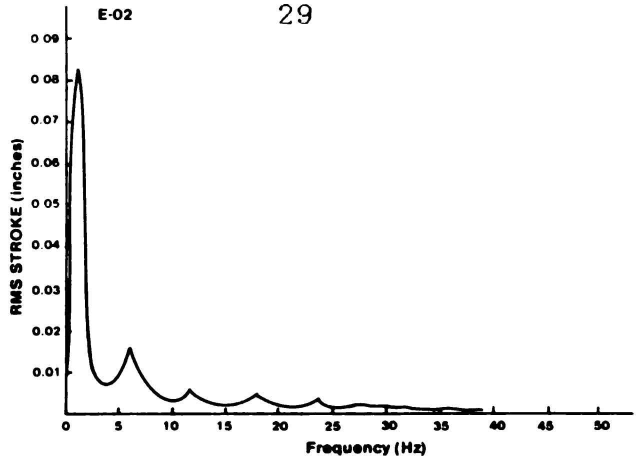


Figure 15. Front axle: Average between curb and roadsides front axle of RMS stroke (inches) as a function of frequency

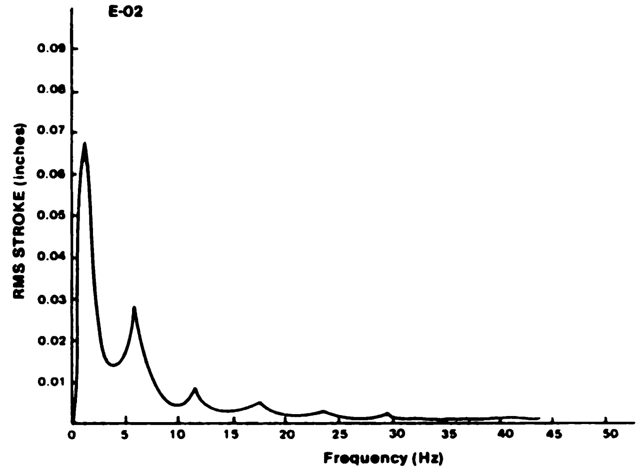


Figure 16. Fifth wheel: Average between curb and roadsides fifth wheel of RMS stroke (inches) as a function of frequency

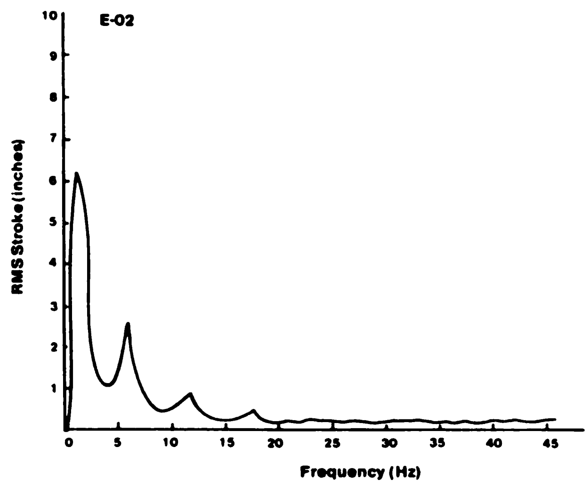


Figure 17. Rear axle: Average between curb and roadsides rear axle of RMS stroke in inches as a function of frequency

functions for displacement amplitudes are very close when compared to each other.

For the rear axle, the numbers of occurrences for both wheels are also very close to one another (Table 2). Maximum amplitudes however do vary for each location. Comparing values of RMS stroke for the input at the fifth wheel and at the rear axle we observe that they are very similar. In general all inputs at different locations are similar because similar wheels were used for both axles and the distances between them were equal.

In general, the average input stroke to the different actuator varies between 0.14 and 0.18 inches, but isolated strokes as high as 3 inches were measured.

4.2 Truckbed Vibration:

Table 4 shows the vibration response of the truckbed due to the effect of the actuator input described in section 4.1.

According to Table 4, the maximum and minimum accelerations exceeded 1 g for both the fifth wheel and the rear axle which means that anything not hard mounted to the floor is very likely to bounce. Values of RMS acceleration at different frequencies and peak accelerations are presented in Table 5.

Table 4. Truckbed G Levels for Front and Rear.

Position	Mean (μ)	Maximum	Minimum	Std. Dev. (σ)
Front	-6.33E-06	1.24	-1.34	0.201
Rear	8.26E-05	2.40	-3.11	0.292

Table 5. Peak G Acceleration of the Floor (Truckbed) Versus Frequency.

Frequency (Hz)	Location	RMS acceleration (g)	Peak G acceleration (g)
1	Front	0.008	0.0113
	Rear	0.010	0.0141
6	Front	0.055	0.0778
	Rear	0.100	0.1414
12	Front	0.020	0.0283
	Rear	0.030	0.0424
18	Front	0.035	0.0495
	Rear	0.025	0.0354
24	Front	0.018	0.0255
	Rear	0.008	0.0113
30	Front	0.015	0.0212
	Rear	0.010	0.0141

Figures 18 and 19 show the behavior of RMS acceleration in g´s as a function of the frequency of vibration at the truckbed.

The acceleration level of maximum occurrence (average acceleration level) at the truckbed is approximately zero for both front and back of the trailer. It can also be observed from Table 4 that 68% of the time, acceleration levels are between -0.2 and + 0.2 g´s for the front and between -0.3 and +0.3 g´s for the back of the truck. However, acceleration levels between -0.6 and +0.6 g´s for the front and between -0.9 and +0.9 g´s for the back were also measured. Maximum and minimum accelerations do exceed 1 g for both the fifth wheel and the rear axle, but occur infrequently. These rare events however can generate damage in packaged goods and should not be overlooked.

4.3 Package Responses:

The acceleration responses of the three instrumented packages that were put on the top of the stacks are presented in Table 6.

Figure 20 shows the truckbed vibration acceleration and the responses for packages A, B and C at the front of the trailer. Figure 21 shows the response for the packages at the back of the trailer. The response curves of the packages, especially at the back, show a skew distribution

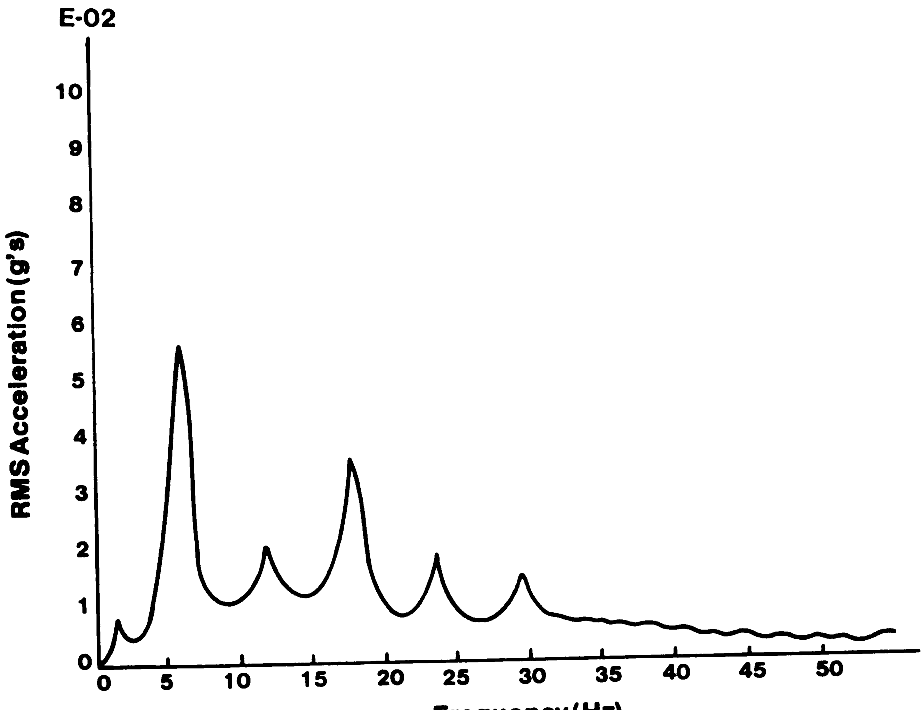


Figure 18. Truckbed RMS acceleration as a function of frequency (front)

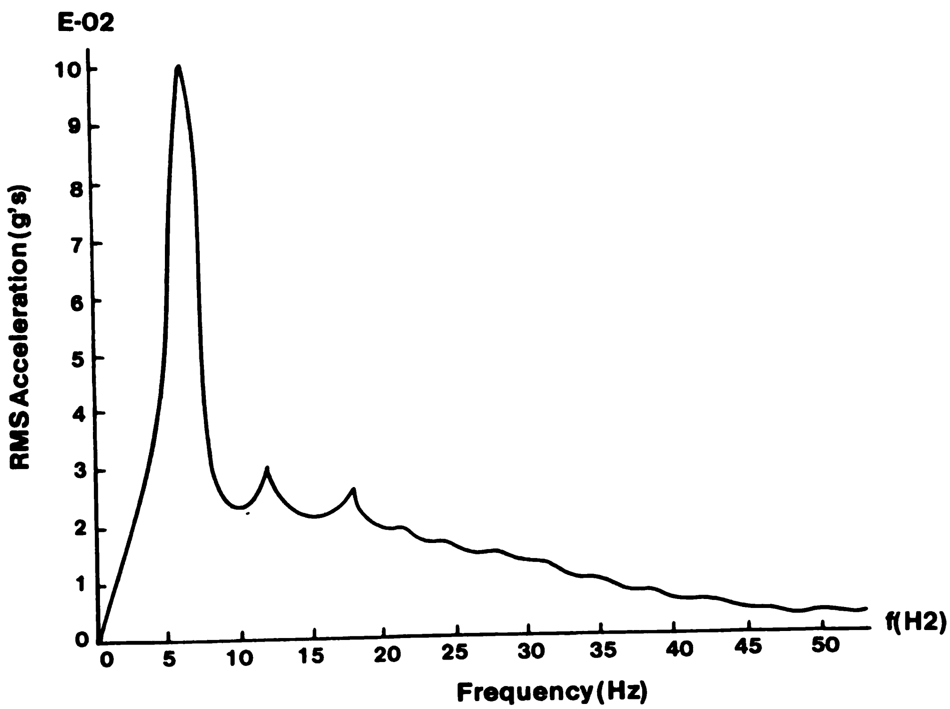


Figure 19. Truckbed RMS acceleration as a function of frequency (rear)

Table 6. Output Acceleration (g) on Packages A, B and C.

Package	Location	Mean (μ)	Maximum	Minimum	Std. Dev. (σ)
A	Front	-9.32E-05	1.47	-2.83	0.555
	Rear	6.77E-05	1.48	-8.38	0.850
B	Front	-1.06E-04	1.68	-5.57	0.508
	Rear	-6.33E-05	1.98	-9.56	0.651
C	Front	1.34E-04	0.98	-2.06	0.326
	Rear	5.96E-05	1.20	-4.31	0.435

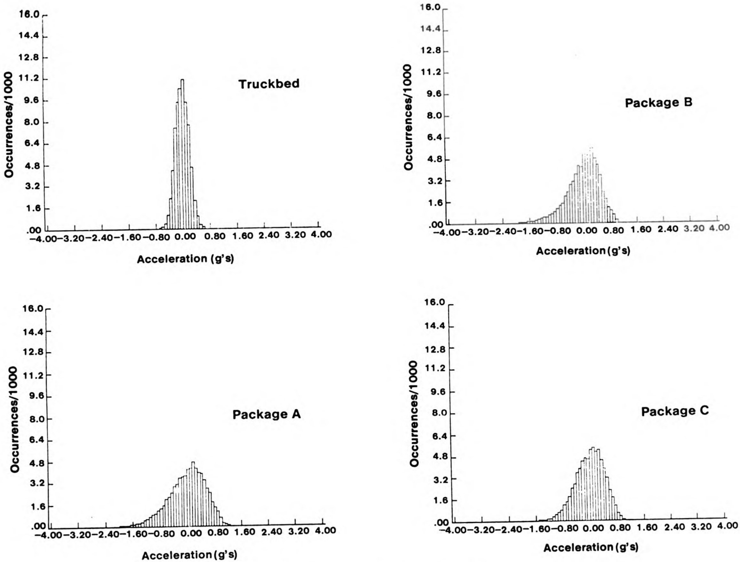


Figure 20. Response of Packages to Input Acceleration (Fifth Wheel)

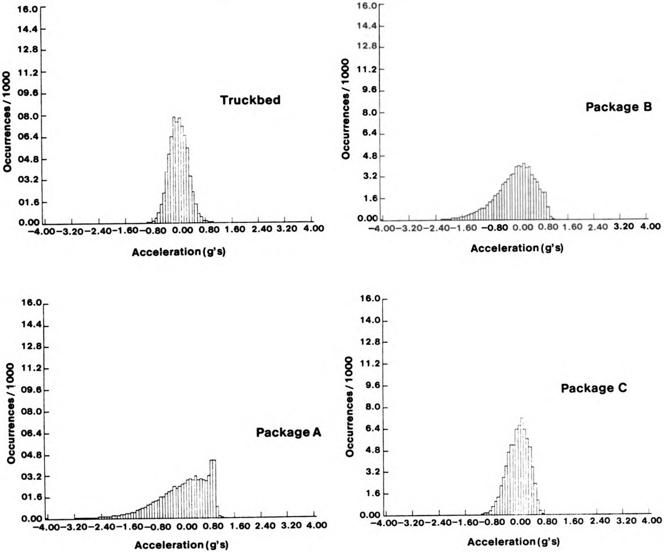


Figure 21. Response of Packages to Input Acceleration on Truckbed (Rear Axle)

to the left, indicating the presence of decelerations generated by repetitive impacts.

Table 7 shows the values of RMS acceleration and peak acceleration for the three different packages at the front and at the rear.

Table 8 compares values of peak acceleration of the truckbed and of the packages at the top showing relative Magnification Factors.

The data shows vibration isolation at high frequencies as expected. See Appendix B for a description of vibration magnification and isolation. At frequencies higher than 30 Hz, all three packages show Magnification Factors below 1.0.

4.4 Random Vibration:

A broad band Random Vibration input signal was also used to simulate over-the-road vibration for 600 seconds. Frequency response functions were developed between the various locations in the truck and the packages to the input signal. Table 9 shows the FRF between the three axles (front, fifth wheel and rear) to the floor at the rear. Table 10 shows the values of Magnification Factors for the packages A, B and C due to the input acceleration at the rear of the truckbed. Table 11 shows Magnification Factors at different forcing frequencies.

Table 7. Packages RMS and Peak G Accelerations Versus Frequency.

Freq (Hz)	Location	Acceleration (g)					
		RMS			Peak G		
		Pkg A	Pkg B	Pkg C	Pkg A	Pkg B	Pkg C
1	Front	0.010	0.010	0.010	0.014	0.014	0.014
	Rear	0.015	0.015	0.010	0.021	0.021	0.014
6	Front	0.320	0.265	0.145	0.453	0.375	0.205
	Rear	0.350	0.295	0.192	0.495	0.417	0.272
12	Front	0.048	0.060	0.032	0.068	0.085	0.045
	Rear	0.075	0.055	0.032	0.106	0.078	0.045
18	Front	0.025	0.032	0.020	0.035	0.045	0.028
	Rear	0.025	0.032	0.020	0.035	0.045	0.028
24	Front	0.008	0.020	0.007	0.011	0.028	0.010
	Rear	0.010	0.020	0.015	0.014	0.028	0.021
30	Front	0.005	0.010	0.005	0.007	0.014	0.007
	Rear	0.008	0.015	0.008	0.011	0.021	0.011

Table 8. Relative Magnification Factor (from Truckbed to Packages) Versus Frequency.

Frequency (Hz)	Location	Relative Magnification Factor		
		Package A	Package B	Package C
1	Front	1.25	1.25	1.25
	Rear	1.50	1.50	1.00
6	Front	5.81	4.82	2.64
	Rear	3.50	2.95	1.92
12	Front	2.40	3.00	1.60
	Rear	2.50	1.83	1.07
18	Front	0.72	0.92	0.57
	Rear	0.99	1.28	0.80
24	Front	0.44	1.11	0.39
	Rear	1.24	2.21	1.86
30	Front	0.33	0.66	0.33
	Rear	0.78	1.49	0.78

Table 9. Maximum FRF Between Actuators Stroke and the Floor Acceleration at the Rear.

Freq. (Hz)	(1)	(2)	(a)	(3)	(4)	(b)	(5)	(6)	(c)
0 - 5	0.5	0.3	0.40	1.0	1.2	1.10	2.8	2.9	2.85
5 - 10	0.4	0.2	0.30	1.1	1.2	1.15	2.5	2.8	2.65
10 - 15	0.8	1.0	0.90	2.0	2.1	2.05	2.7	2.2	2.45
15 - 20	0.7	0.7	0.70	2.0	2.5	2.25	2.6	2.2	2.40
20 - 25	1.2	1.2	1.20	1.8	1.8	1.80	2.3	2.2	2.25
25 - 30	2.7	1.2	1.95	2.1	2.8	2.45	3.3	5.1	4.20
30 - 35	2.8	2.2	2.50	3.7	3.8	3.75	5.8	6.8	6.30
35 - 40	3.2	2.9	3.05	5.2	4.1	4.65	6.0	7.4	6.70
40 - 45	5.2	3.5	4.35	3.3	4.2	3.75	8.2	7.8	8.00
45 - 50	5.1	3.4	4.25	4.2	4.5	4.35	6.7	7.5	7.10

Col (1): FRF between the roadside front axle actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (2): FRF between the curbside front axle actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (3): FRF between the roadside fifth wheel actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (4): FRF between the curbside fifth wheel actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (5): FRF between the roadside rear axle actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (6): FRF between the curbside rear axle actuator stroke (input) and floor acceleration at the rear of the trailer (output).

Col (a): Average of columns 1 and 2.
Col (b): Average of columns 3 and 4.
Col (c): Average of columns 5 and 6.

Table 10. Magnification Factor of Acceleration in Packages A, B and C Due to the Acceleration at the Rear Truckbed, Based on Random Vibration.

Frequency (Hz)	Magnification Factor		
	Pkg A	Pkg B	Pkg C
0.0	1.0	1.0	1.0
2.5	1.2	1.2	1.2
5.0	3.7	2.1	2.0
7.5	1.7	2.5	2.7
10.0	0.6	1.5	1.0
12.5	0.5	1.0	0.4
15.0	0.4	0.8	0.3
17.5	0.4	0.7	0.3
20.0	0.3	0.6	0.3
22.5	0.2	0.5	0.3
25.0	0.1	0.4	0.2
27.5	0.1	0.3	0.1
30.0	0.1	0.2	0.1

Table 11. Values and Frequencies of Peak Magnification Factor for the Three Different Packages.

Package	Peak M	Frequency (Hz)
A	3.9	6
B	2.7	7
C	3.2	7

5.0 CONCLUSIONS AND RECOMMENDATIONS

This study concludes the following:

1) For the same input at the axles, the response at the back truckbed of the trailer is 50% larger than that at the front (Figures 18 and 19). The average Magnification Factor from the input at the wheels to the truckbed is 2 for the front and 3 for the back for this truck.

2) Acceleration levels around 1.5 g´s for the products are very common in LTL shipments. However, levels as high as 10 g´s were recorded indicating severe bouncing. This shows that when testing for LTL shipments it may be important to conduct repetitive shock testing to check survivability of the product.

3) It cannot be generalized from LTL shipment data that one position is better than another inside the truck. Magnification Factors (between the truckbed and the packages) between 0.4 and 6.0 are present both at the front and at the back.

4) Low natural frequency packages have in general larger Magnification Factors, showing more bouncing in LTL shipments because the stacks exhibit resonance around 6 - 7 Hz, but the Magnification Factor does not vary in much when comparing packages at the top of the same stack. This is because the multiple degree of freedom system formed by the stack itself and the instrumented package at the top have almost the same fundamental mode, with very little difference. (Table 11 shows that the peak Magnification Factors occur around 6 - 7 Hz indicating this to be the resonant frequency of the whole stack).

5) Roadside or curbside locations input little difference along the same axle.

6) Low natural frequency packages (below 10 Hz) show more bouncing and larger accelerations than high natural frequency packages (Figures 20 and 21). The number of impacts with small package acceleration generally increases with the natural frequency of the package. This shows that packages with high natural frequencies move less than packages with low natural frequencies because resonance of the truckbed occurs at low frequencies. This causes higher magnification to low natural frequency packages in LTL shipments.

7) Low natural frequency packages have a broader distribution for the response acceleration levels than high natural frequency packages. Package responses do not show normal Gaussian distribution. Instead they are skew to the negative accelerations. Large negative values for acceleration (deceleration) are caused when the package impacts the stack.

8) The front axle and the fifth wheel also contribute with the input to the rear truckbed. Based on the Frequency Response Functions (FRF) determined (Table 9) it can be seen that the front axle contributes about 20% of the vibration input to the rear truckbed. Similarly the fifth wheel contributes about 30%. The remainder (50%) is due to the input from the rear axle. The FRF´s were determined by measuring the response at the rear of truckbed due to the individual axles. This was obtained by holding the two other axles fixed, giving a known input to the axle being studied and measuring the response.

APPENDICES

APPENDIX A

STATISTICAL CONCEPTS

All phenomena described by mathematical functions relating several variables are said to be deterministic which simply means that a given set of conditions produces a different result. As an example is the mathematical relationship between speed, distance and time, v = d/t. Non deterministic phenomena cannot be characterized by a mathematical function because of inherent uncertainties in one or more casual factors which affect the outcome. An example is the tossing of a coin. The case of vibration in the transportation environment is another example of a non deterministic phenomena because of the many factors such as road conditions, truck suspension characteristics, driver quality, and traffic situations which cannot be accounted for exactly. Since any of the above conditions are likely to change at any time, any data gathered from non deterministic phenomena of this type are described by random time functions.

A random time function, such as the time history of a vibration signal, despite its apparent irregularity when graphed, shows a certain degree of statistical constancy. Because of its random nature, no single parameter is enough to characterize the whole phenomena. Therefore, averaging procedures are applied to identify important governing characteristics of the phenomena. Consider for example the time function x(t) which represents the acceleration of an object due to vibration over the time interval t = 0 to t = T where T is called the period of the signal.

Some important averaging statistics related to this signal are given below. For each case, the general definition for a random signal will be given and the definition will then be applied to the special case of a half sine wave of amplitude A and period T described by the expression:

$$x(t) = A \sin (\pi t/T)$$

Refer to Figures 22 and 23 for a graphical representation.

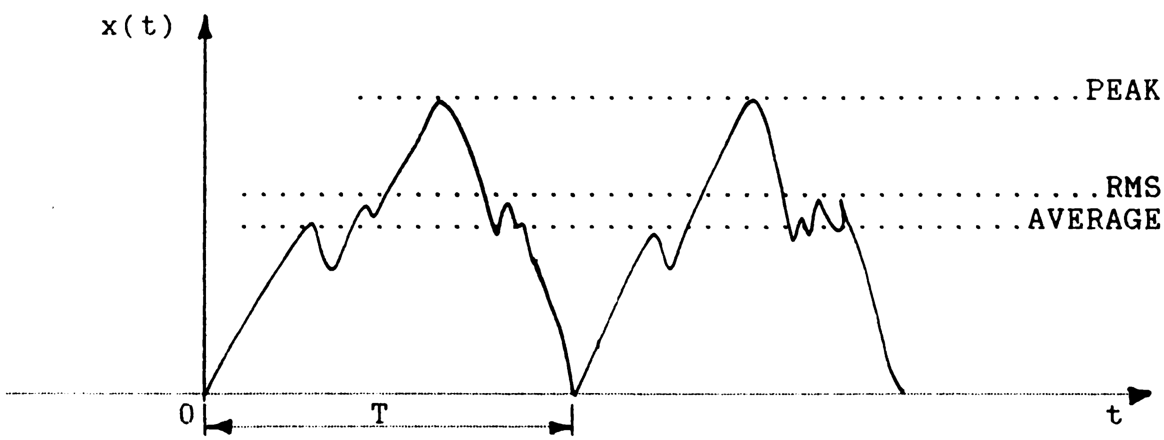


Figure 22. Random Signal

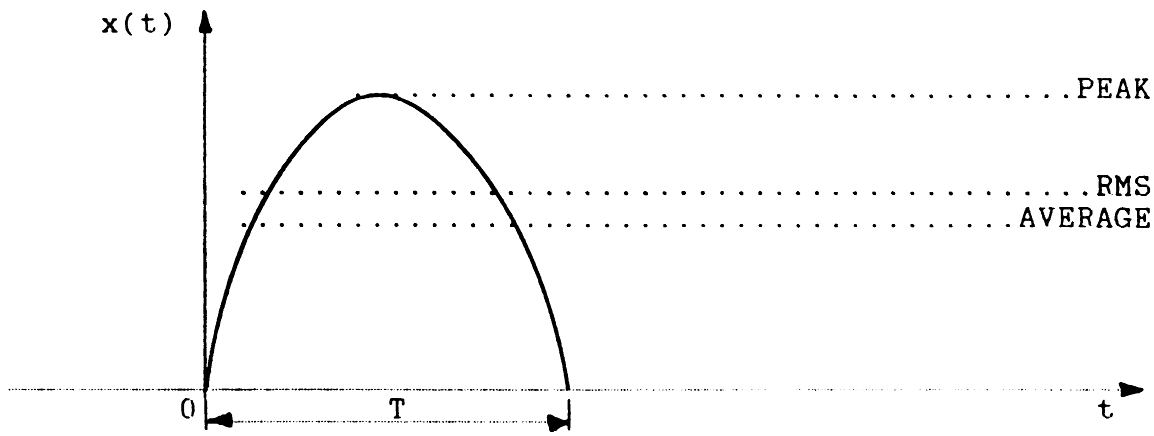


Figure 23. Half Sine wave

Peak Value: Generally indicates the maximum acceleration that the vibrating part experiences over the period of the signal. For the sine wave, the peak value is A.

Average or expected value: Over the duration of the signal, the average acceleration is defined to be:

$$\overline{x(t)} = E[x(t)] = (1/T) \int_0^T x(t)dt$$

For the sine wave:

$$\overline{x(t)} = (1/T) \int_0^T A \sin(\pi t/T)dt = (2/\pi)A$$

Mean Square Value: This statistic provides a measurement of the energy of vibration.

$$\overline{[x(t)]^2} = E[x^2(t)] = (1/T) \int_0^T x^2(t)dt$$

For the sine wave:

$$\overline{[x(t)]^2} = (1/T) \int_0^T A^2 \sin^2(\pi t/T)dt = A^2/2$$

Variance: This statistic is a measure of the spread in data about the mean value.

$$\sigma^2 = (1/T) \int_0^T (x - \bar{x})^2 dt = \overline{x^2} - (\bar{x})^2$$

For the sine wave:

$$\sigma^2 = (1/2 - 4/\pi^2)A^2 = 0.0947\ A^2$$

Root Mean Square (RMS): This is the square root of the mean square value.

$$\text{RMS} = \sqrt{\text{Mean Square Value}}$$

For the sine wave:

$$\text{RMS acceleration} = \sqrt{A^2/2} = 0.707\ A$$

Standard Deviation: This is the average deviation from the mean value.

$$\sigma = \sqrt{\overline{x^2} - (\bar{x})^2}$$

For the sine wave: $\sigma = 0.308\ A$

Gauss' Distribution: The distribution of accelerations measured during transportation over time has an approximately normal distribution. The probability of measuring an acceleration between the limits x and x+dx, where dx is small is:

$$P(x) = \frac{1}{\sigma\sqrt{2\pi}} \exp(-x^2/2\sigma^2)dx$$

<u>Frequency Domain Representation:</u> This is the most common method of representing the characteristics of random vibration. Every random signal may over a suitably chosen time period T be represented as a superposition of pure sine waves each with its own amplitude and frequency. The process of obtaining these component sine waves is known as Fourier decomposition. The random signal x(t) over the duration T may then be represented in graphical form as component sine wave amplitude versus corresponding frequency. In general, this frequency domain representation exhibits strong contributions from certain frequencies and only minor contributions from others.

<u>PDS - Power Density Spectrum:</u> The Fourier representation of a complex waveform decomposed into simple sine waves is an exact description of the real waveform only over the time interval T. One of the characteristics of random vibration however is that the complex waveform does not repeat itself exactly for each successive period T. Therefore, an alternate but related procedure is used. A complex waveform is first fed into an electronic bandpass filter which electronically extracts from the signal the component sine waves present within a narrow band (BW) of frequencies. Over a long period of time, accelerations are sampled and from n samples, the RMS acceleration is determined. The power density is then calculated according to:

$$PD = \frac{\sum_i (RMSi)^2}{n * BW}$$

PD = Power Density (g^2/Hz)

RMSi = RMS acceleration value (g)

n = Number of instants sampled

BW = Bandwidth (usually normalized to 1 Hz)

A Power Density Spectrum shows the values of Power Density as a function of frequency.

APPENDIX B

TRANSPORTATION CONCEPTS

In statistical terms, the power density at any given frequency is the variance about a mean value of zero acceleration. Therefore, based on the probabilities associated with the normal Gaussian distribution, we may predict the acceleration levels associated with any of the component frequencies of the complex waveform.

Accelerations of ± 1 PD values occur 68.3 % of time (interval from $-\sigma$ to $+\sigma$); accelerations of ± 2 PD values occur 95.4 % of time (interval from -2σ to $+2\sigma$); accelerations of ± 3 PD values occur 99.7 % of time (interval from -3σ to $+3\sigma$). (Brandenburg, 1985)

It is commonly accepted that the worst vertical acceleration of a truck ride is about 0.5 g for low frequencies. (Brandenburg, 1985)

When considering trucks and trailers, there are three major vibration modes. Following the terminology accepted by Tevelow (1983), these modes are:

1)Bounce: oscillation resulting in a up and down motion.
2)Pitching: creates rocking motion, longitudinal or side to side.
3)Frame bending or beaming: describes the flexure of the vehicle body such that the ends of the frame are moving up and down, at the same time and same direction while the center goes to opposite direction.

Bounce and pitching are rigid body motions and bending is not.

Magnification Factor: (Also called Transmissibility) is the ratio of dynamic output to dynamic input of a vibration system.

$$M = \frac{\text{Output acceleration}}{\text{Input acceleration}} = \frac{\text{Output stroke}}{\text{Input stroke}}$$

When the damping in the system is negligible, the theoretical magnification factor is:

$$M = \frac{1}{1 - r^2}$$

where r = Frequency ratio = f/f_n with f the input vibration frequency and f_n the natural frequency of the object being vibrated. It is clear from above that when r = 1, M goes to infinity. This is the point where the system experiences resonance. When r is several times greater than 1, M is insignificant, and the system reaches vibration isolation. For values of r between 0 and 1 the system is said to vibrate "in-phase" (same direction of oscillation for both input and output) and for values of r greater than 1 it is said to vibrate "out-of-phase" (opposite directions of oscillation) . Figure 24 shows a graph of the Magnification Factor M versus r, which is the ratio of forcing to natural frequency.

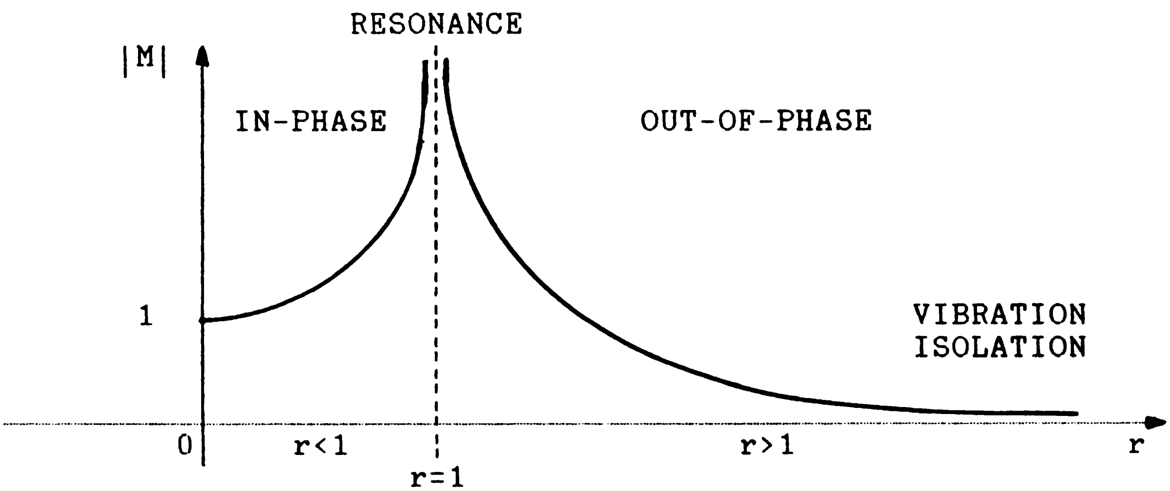


Figure 24. Magnification Factor

LIST OF REFERENCES

LIST OF REFERENCES

American President Companies, Double-Stack/Truck/Vessel Ride Characteristics Study, Equipment Engineering, Dec.1986.

Brandenburg, R.K. and Lee, J.J.L, Fundaments of Packaging Dynamics, MTS Systems Corporation, 1985.

Caruso, H., and W. Silver, Advances in Shipping Damage Prevention, Shock and Vibration Bulletin 46, Part 4, 41-47, 1976.

Foley J.T., Preliminary Analysis if Data Obtained in the Joint Army/AEC/Sandia Test of Truck Transport Environment, Shock and Vibration Bulletin 35, Part 5, 1966.

Foley, J.T., The Environment Experienced by Cargo on a Flatbed Tractor-Trailer Combination, Sandia Laboratories, SC-RR-66-677, 1966.

Fruehauf Corp., Yellow Freight Project, Service request number 411-11995-500, test date-Friday, February 26th, 1988.

Goff, J.W.; Grebe, J.; Twede, D. and Townsend, T., You'll Never See It From The Road, Technical Report # 26, School of Packaging, Michigan State University, 1984.

Goodwin, D.L., and Holland, R.R., Shock and Vibration Dynamics of Intermodal Distribution Environment - Trailer On a Flat Car, Container In a Well Car, 1987.

Grier, J.H.; Macleod, N.J.; Fisher, F.R., and L.S. Marr, Joint Services Highway Shock Index Project, Military Traffic Management Command, 75-17, 1975, AD 850571.

Harris, C.M., and C.E. Crede, Shock and Vibration Handbook, Vol. 3, Mc Graw Hill Book Company, NY, 1961.

Magnuson, C.F., Shock and Vibration Environments for Large Shipping Container During Truck Transport, (Part 1), Sandia Laboratories, SAND77-1110, 1977.

Magnuson, C.F., Shock and Vibration Environments of Larges Shipping Container During Truck Transport, Part 2), Sandia Laboratories, SAND 78-0337, Nuclear Regulatory Commission, NUREG/CR-0128, 1978.

Pichler, E.F., Embalagem e Acondicionamento para Transporte e Exportacao, Instituto de Pesquisas Tecnologicas, SP, Brazil, 1984.

Schlue, J.W., The Dynamic Environment of Spacecraft Surface Transportation, Shock and Vibration Bulletin 35, Part 5, 1966.

Sharpe, W.N., Jr.; Kusza, T.J.; Sherman, F.W. , and J.C. Goff, Preliminary Measurement and Analysis of the Vibration Environment of Commom Motor Cariers, Shock and Vibration Bulletin 44, Part 4, 1974.

Tevelow, F.L., The military Logistical Transportation Vibration Environment: Its Characterization and relevance to MIL-STD Fuze Vibration Testing, U.S. Army Electronics Research and Development Command, Harry Diamond Laboratories, 1983.

Thomsom, W.T., Theory of Vibration With Application, 3rd Ed., Prentice Hall, 1988.

Turczyn, M.T.; Stevens, D.G., and Camp, T.H. , Shock and Vibration Environment in a Livestock Trailer, Shock and Vibration Bulletin 50, Part 2, 91-101, 1980.