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NBS SPECIAL PUBLICATION 652

U.S. DEPARTMENT OF COMMERCE/National Bureau of Standards

Damage Prevention in the Transportation Environment

MFPG
34th Meeting

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NBS special publication

Damage Prevention in the Transportation Environment

Proceedings of the 34th Meeting of the
Mechanical Failures Prevention Group
held at the National Bureau of Standards,
Gaithersburg, Maryland, October 21-23, 1981

Edited by:
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This meeting was sponsored jointly by NBS and the:
Office of Naval Research
U.S. Department of the Navy
Arlington, VA 22217

Naval Air Systems Command
U.S. Department of the Navy
Washington, DC 20360

National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, MD 20771

U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, Secretary
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, Director

Issued April 1983

Library of Congress Catalog Card Number: 83-600521

National Bureau of Standards Special Publication 652
Natl. Bur. Stand. (U.S.), Spec. Publ. 652, 349 pages (April 1983)
CODEN: XNBSAV

U.S. GOVERNMENT PRINTING OFFICE
WASHINGTON: 1983

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402
Price $8.00
(Add 25 percent for other than U.S. mailing)

PREFACE

The 34th Meeting of the Mechanical Failures Prevention Group was held October 21-23, 1981 at the National Bureau of Standards in Gaithersburg, Maryland. The program was organized by the MFPG Technical Committee on Design under the chairmanship of Jesse E. Stern of Trident Engineering Associates. Where possible, the papers in these proceedings are presented as submitted by the authors in the form of camera ready copy. Limited editorial changes, some retyping, and some photographic reduction were required.

Support of the National Bureau of Standards, the Office of Naval Research, the Naval Air Systems Command, and the National Aeronautics and Space Administration is gratefully acknowledged.

Appreciation is expressed to Kathy Stang of the NBS National Measurement Laboratory for handling financial matters and to Sara Torrence and Greta Pignone of the NBS Public Information Division for the meeting and hotel arrangements.

T. Robert Shives
Executive Secretary, MFPG

Fracture and Deformation Division
National Bureau of Standards

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ABSTRACT

These proceedings consist of 27 submitted entries (25 papers and 2 abstracts) from the 34th Meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards, Gaithersburg, Maryland, October 21-23, 1981. The subject of the symposium was damage prevention in the transportation environment. Areas of special emphasis included packaging for the transportation environment, research in the railroad industry, damage prevention in the railroad industry, designing for the transportation of hazardous materials, and highways.

Key words: Airline environment packaging; hazardous materials transportation; highway design; marine environment packaging; packaging techniques; railroad environment packaging; shipping; transportation; transportation damage prevention; trucking environment packaging.

UNITS AND SYMBOLS

Customary U. S. units and symbols appear in some of the papers in these proceedings. The participants in the 34th Meeting of the Mechanical Failures Prevention Group have used the established units and symbols commonly employed in their professional fields. However, as an aid to the reader in increasing familiarity with the usage of the metric system of units (SI), the following references are given:

NBS Special Publication, SP 330, 1981 Edition, "The International System of Units."

ISO International Standard 1000 (1973 Edition), "SI Units and Recommendations for Use of Their Multiples."

IEEE Standard Metric Practice (Institute of Electrical and Electronics Engineers, Inc., Standard 268-1979).

DISCLAIMER

Certain trade names and company products are identified in order to adequately specify the experimental procedure. In no case does such identification imply recommendation or endorsement by the National Bureau of Standards, nor does it imply that the products are necessarily the best available for the purpose. Views expressed by the various authors are their own and do not necessarily represent those of the National Bureau of Standards.

SESSION I

PACKAGING FOR THE TRANSPORTATION ENVIRONMENT

CHAIRMAN: JESSE E. STERN
TRIDENT ENGINEERING ASSOCIATES

ALLOCATING LOSS AND DAMAGE TO THE RAILROAD TRANSPORT CYCLE

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Abstract: This paper presents the results of allocating loss and damage cost to various elements of the railroad transport cycle. Estimates of loss and damage attributed to linehaul shock and vibration and flat and hump yard coupling impacts are provided. In addition, loss and damage estimates are provided for various levels of overspeed impacts in hump yards.

Key words: Loss and damage; lading damage; freight damage.

INTRODUCTION

In calendar year 1978, the Association of American Railroads (AAR) statistics indicate that North American railroads lost a total of $304,368,908 in freight loss and damage (L&D); this represented approximately 1.3% of the total gross operating revenues for all railroads. Industry sources indicate that the indirect costs to railroads and shippers to process and handle L&D claims may be as much as eight times greater than the direct L&D payments. If this is true, then the total costs of L&D to railroads and shippers may be in excess of $2.5 billion dollars per year.

There is a current lack of information and understanding of the L&D costs which can be attributed to shock and vibration (rough handling) in the transport cycle. In particular, we would like to know how much L&D can be attributed to railroad yards versus the linehaul. In yards, we would like to know how much L&D can be attributed to hump yards versus flat yards, and how L&D increases with impact speeds in yards. Over the linehaul, we would like to assess the contribution to L&D of longitudinal train slack-action versus vertical vibration.

This L&D transport cycle information is important to assist in planning the priorities of future L&D countermeasures to investigate the effects of shock and vibration. Furthermore, this information is necessary to assess the benefit-cost impact of any specific proposed countermeasures. Examples of potential areas for countermeasure development include:

- Countermeasures to reduce L&D due to train slack-action over linehaul

- Countermeasures to reduce L&D due to vertical vibration over the linehaul

- Countermeasures to reduce L&D due to overspeed impacts in hump or flat yards

RESULTS

A systematic methodology was developed to allocate 1978 loss and damage direct payment costs as detailed in the Association of American Railroads Circular No. FCDP-133 entitled: "Freight Loss and Damage, 1978." The results of the methodology are presented below.

The 1978 L&D direct payments costs caused by shock and vibration is estimated to be $101.3 million; $55.7 million is due to yard impacts and $45.6 million is attributed to vibration and train-slack action in the linehaul. On a per loaded trip basis the average L&D direct payment associated with shock and vibration is $4.33; $2.38 is associated with yards and $1.95 with linehaul. Of the $55.7 million due to overspeed impacts in yards, $22.5 million is associated with hump yards and $30.2 million is associated with flat yards. It is to be noted that 20% of the switching activity occurs in hump yards and 80% in the flat yards.

The average L&D direct payment costs per hump yard coupling is estimated to be approximately $0.91. For a hump yard which processes 2,000 cars per day over the hump of which half the cars are loaded, this implies L&D direct payments costs of $910 per day or $322,000 per year due to overspeed impacts. The expected L&D direct payment costs per occurrence of hump yard overspeed impact level is detailed below.

Overspeed Impact	Expected Direct Payment Costs Per Occurrence
<4 mph	$ 0.00
4 - 5 mph	$ 0.13
5 - 6 mph	$ 1.19
6 - 7 mph	$ 3.31
7 - 8 mph	$ 6.49
8 - 9 mph	$10.73
9 - 10 mph	$16.03
>10 mph	$22.39

It should be remarked that the direct payment costs detailed above represent only a fraction of the total costs and that indirect costs to the railroad and shipper to process and handle claims can be as much as eight times the direct payment payment costs. Also, it should be emphasized that the costs which have been calculated are average or mean values (also called expected values in statistics). Therefore, one must be careful in interpreting these costs. The average cost for a particular event (e.g., per coupling, per loaded trip) is calculated by taking the total costs for all events and dividing by the total number of events. Thus, the costs associated with any particular occurrence of an event may deviate significantly from the average cost; the average cost is actually surrounded by a statistical distribution of costs. For example, we estimate that the average 1978 L&D direct payment costs per hump yard coupling is approximately $0.91. This does not mean that a 100 car sample would yield a $0.91 L&D for each coupling; the actual distribution may be 99 couplings having no loss and damage and one coupling having a cost of $91, for purposes of cost-benefit economic analysis, where aggregate statistics are used, the use of average values should be relatively accurate.

METHODOLOGY SUMMARY

The AAR each year provides aggregate statistics on L&D. In particular, in 1978, the L&D payments for North American railroads amounted to $304,368,908. For reporting purposes, Ref. 1 lists L&D in the following 12 categories:

1. Shortage, package shipment
2. Shortage, bulk shipment
3. All damage not otherwise provided for
4. Defective or unfit equipment
5. Temperature failures
6. Delay
7. Robbery, theft, pilferage
8. Concealed damage
9. Train accident
10. Fire, marine, and catastrophies
11. Error of employee
12. Vandalism

Unfortunately, the above 12 categories do not indicate the L&D due to shock and vibration (i.e., rough handling) which is germane to this work. Freight damage resulting from excessive shock and vibration would probably be listed in category 3, "all damage not otherwise provided for" and category 8, "concealed damage." Almost $156.7 million or 51.5 percent of the total 1978 freight loss and damage payments was classified as category 3, and $1.6 million or 0.5 percent was classified as category 8. Other major factors that effect L&D in categories 3 and 8 are inadequate packaging, improper loading, and claims incorrectly assigned to these categories. The methodology is concerned with allocating the AAR category 3 and 8 L&D to shock and vibration in the transport cycle. This allocation is done in four steps:

- Step 1: Allocation to shock and vibration versus others
- Step 2: Allocation to linehaul versus yard
- Step 3: Allocation to flat vs. hump yard
- Step 4: Allocation to hump yard overspeed impact levels.

The details of the methodology are presented in Ref. 2; herein, we present only a summary.

- Step 1: Data from a corrogated container study (Ref. 3) and from a Whirlpool appliance study (Ref. 4) is combined with a table of gross claims paid by cause and commodity (Ref. 5) to estimate the allocation of L&D to shock and vibration versus "others."

- Step 2: Data obtained from the Southern Pacific Transportation Company on shifted loads is used to estimate the allocation of L&D between the linehaul and yards.

- Step 3: Assuming that damage is proportional to the squared difference between 4 mph and coupling speed (Ref. 6), data on coupling speeds in hump and flat yards (Ref. 7) is combined with the relative switching activity in hump and flat yards (Ref. 8) to estimate the allocation of L&D between hump and flat yards.

- Step 4: If we assume the frequency distribution of coupling impacts in hump yards (Ref. 7), can be interpreted as a probability of occurrence then an equation for the "expected" L&D per coupling impact can be written in terms of "expected" L&D occurring at various coupling speeds. The "unknowns" in this equation are the expected L&D occurring at various coupling speeds. However, if we assume that L&D at one coupling speed is related to the L&D at another coupling speed by the squared difference in coupling speeds, then a set of simultaneous equations can be generated to allow all the "unknowns" to be solved. This process is used to estimate the L&D in a hump yard for various levels of impact.

CONCLUDING REMARKS

The details of the analysis and procedures are presented in Ref. 2 In addition, Ref. 2 presents parametric curves which allows one to perform the allocation of L&D based on different assumptions. Also, in Ref. 2 the methodology is applied to estimating the allocation of coupler replacements costs due to shock and vibration to various elements of the transport cycle. Finally, an extensive bibliography on loss and damage is presented in Ref. 2.

ACKNOWLEDGEMENTS

This research was supported by the Federal Railroad Administration (FRA) under contract No. DOT-FR-9082; the project technical monitor was Ms. Claire Orth. The author wishes to acknowledge the support and encouragement of Mr. Phil Olekszyk (FRA) and Mr. William Cracker (FRA). Valuable information to the project was supplied by Mr. Barney Gallacher (retired, Southern Pacific Transportation Company) and Mr. Robert Robbins (Weyerhauser Company).

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EXPORT PACKING IN THE MARINE ENVIRONMENT

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Purpose and Scope

Before there can be any meaningful discussion about preparation of goods for export shipment, there must first be a firm understanding of the terms preservation, packaging and packing. These three terms comprise a total system which when properly implemented serve as follows:

To protect cargo from the elements and the normal rigors encountered in export transportation.

To facilitate handling.

To provide a degree of rigidity to the cargo being shipped.

To provide continued protection to the cargo in short term or long term storage.

This paper is intended to offer the shipper a ready reference on how to prepare his product for export shipment. The paper addresses, in general, the basics of preservation, packaging and packing including the selection of materials, methods of construction of skids, crates and boxes and a brief look at containerization.

Definitions

Preservation is the operation by which goods are protected from the elements, i.e. humidity, fresh water, salt water, heat, cold and corrosive atmospheres. Preservation usually consists of the direct application of a coating specifically designed for protection against a known hazard such as rust. Other types of preservation are VCI and VPI which are not contact preservatives, but, rather, additives in the form of powder or chemically treated paper that serve to inhibit the formation of water vapor and rust resulting from that water vapor. Desiccants are also a means of preservation.

Packaging is an operation consisting of wrapping or boxing an item as a preparatory step prior to the final (outer) shipping container. The type and extent of packaging depends on marketing requirements, segregation requirements, known or assumed levels of transportation shock and vibration, and is usually performed as an intermediate step in the protection of the item from the effects of transportation.

Packing is the final step in the preparation of goods for shipment. Packing consists of the outer or external shipping container, be it a box or crate, and includes such measures as cushioning, blocking, bracing or anchoring the item within this outer shipping container. If the shipper will conscientiously evaluate his cargo and develop the system of preservation, packaging and packing against known or assumed transportation hazards, his cargo will stand an infinitely better chance of arriving at its destination in sound condition.

VCI/VPI are vapor corrosion inhibitors or vapor phase inhibitors which when incorporated in waterproof packaging can add a substantial measure of protection against moisture.

Desiccants are usually moisture absorbing powders contained in packages such as boxes or cloth bags. When placed in relatively watertight or waterproof voids, these desiccants will absorb excess moisture. They can not, however, be expected to continue to absorb water or moisture beyond their absorbtion capability and, therefore, the amount of desiccant used must be carefully determined considering the expected temperature/humidity levels and the volume of the spaces which they are to protect.

Marks and numbers are extremely important. The outer or external shipping container must clearly show certain critical information essential to the safe handling and expeditious delivery of cargoes to their final destination. The name and full address of the consignee; an identifying number or code relating to the particular item being shipped; the full dimensions and gross and net weight in both metric and British systems; and the appropriate international shipping symbols such as "Keep Dry", "Fragile", "This Side Up" and "Lift Here" or "Sling Here", are all part of the labeling requirements.

Unitization is the assembly of cargoes of similar size and configuration into a single bundle, group, platform or pallet. The purpose of unitization is to reduce cubage, facilitate handling, and usually can provide a greater degree of rigidity to the unitized cargoes. The bundling of pipe, coiled wire, lumber and structural steel members are examples of unitization.

Palletization is a special case of unitization and consists of placing one or more similar units of cargo on wooden pallets and securing the cargoes to the pallets (usually) by means of steel or synthetic fabric tension bands. If required, overwraps may be applied to the palletized units to provide either watershed or waterproof protection. Other common methods of restraining and securing palletized cargoes are shrink wrap and tension wrap (polyethylene or polypropylene films).

Protective materials generally refers to contact preservatives such as paint, grease and oil or to the watershed/ waterproof liners and wraps applied to protect the cargo from the elements. "Protective Materials" can also refer to cushioning materials and blocking and bracing.

Skids are wooden timbers or alternative materials used either as a base to facilitate handling for heavy cargoes where cargoes possess inherent rigidity; or serve alone as a base for cargoes lacking inherent rigidity protecting them and facilitating handling. Skids are also major structural members in crate construction. It is important to note that if skids are used alone as a base to carry a cargo with or without inherent rigidity, attention must be given to the type of load, i.e. uniform for concentrated, and the skid must be designed accordingly. In addition, it is important to remember for economic reasons, that skids used as structural members in true crates can be significantly smaller in cross-section than the skids used as supporting bases because crate side and top panels contribute a significant portion of the overall strength to the crate.

A wood crate is a structural framework of members fastened together to form a rigid enclosure which will protect the contents during shipping and storage. This enclosure is usually of rectangular outline and may or may not be sheathed. A crate differs from a nailed wood box in that the frame work of members in the sides and ends provides the basic strength, whereas a box must rely for its strength solely on the boards of the sides, ends, top and bottom. Crate framework can be considered to be similar to a type of truss used in bridge construction. It is designed to absorb most of the stresses imposed by handling and stacking.

A wooden box is simply a rectangular enclosure consisting of wooden boards or planks nailed along their ends or sides to adjacent boards or planks. Boxes generally do not employ a framework such as that used in crates. However, boxes can and should employ cleats and battens, wooden stiffening members, which act as nailers and stiffening members.

The wood species most commonly used in crate construction are divided into four (4) groups, largely on the basis of density. In general, it is good practice to use species in the same group for similar parts. Group I consists of the softer woods such as pine, spruce and chestnut. Group II consists of heavier woods such as Douglas fir and southern yellow pine. Group III woods are generally hard woods of medium density and generally possess relatively good nail-holding capacity. Examples of these woods are ash, soft elm and soft maple. Group IV woods are heavy hardwood species, the heaviest and hardest domestic woods. They have the greatest capacity to both resist shock and hold nails. They are difficult to nail and tend to split when nailed, but are especially useful where high nail holding capacity is required. Oaks, hickory, birch and white ash are examples of Group IV wood.

Wood defects should be carefully considered when selecting wood for skid, crate and box construction. Generally speaking, seasoned wood is preferable to green lumber. Knots and other obvious visible defects such as slope of grain, decay, wane, shakes, checks, splits, and warping are usually obvious physical defects which affect the strength and long-term serviceability of wood. Guidelines for selecting wood with defects are contained in great detail in the Wood Handbook and the Wood Crate Design Manual (Agriculture Handbook No. 252).

Rub strips or runners are lumber strips secured to the underside of skids and skid-based crates to facilitate handling. Rub strips are an essential part of skid assemblies, crates and in box construction for loads in excess of 200 pounds.

Tension bands are metal or fabric bands or straps used to secure or bind cargoes to pallets or skid bases or as an added measure of binding the external members of crates and boxes. Tension bands come in different dimensions and are applied by means of mechanical or hydraulic tensioning equipment. Tension bands are also an essential part of good packing.

Reinforcing straps are mild steel metal bands fitted with nail holes which are affixed at the corners of crates and at the junction of side panel vertical members and top panel horizontal members. Reinforcing straps should be applied in two (2) directions where possible. They serve as an added measure of preventing separation of joints.

Headers are timbers or alternative materials which are bolted across the ends of skid assemblies. They are generally to be of at least the same cross-section as the skid members they join. Headers are an essential part of skid assemblies both in skids acting alone as well as in skids as part of a crate structure.

Load bearing floorboards are wooden timbers or alternative materials which are fastened transversely across the tops of skid assemblies and carry the load of the item for which the skid assembly is intended.

Splicing refers to the joining, end to end, of two (2) lengths of timbers of similar cross section in order to make one longer section with approximately the same longitudinal strength as a single continuous piece. The ends to be spliced are to be notched to one half the timber thickness, and through bolts are to be placed vertically through the joining notched sections. Lapped along each side of the splice is to be a ¼" steel plate of length no less than three (3) times the thickness of the members joined and through-bolted, side to side, with at least two (2) bolts in each half of the splice.

A uniform load refers to a load that is distributed more or less evenly over the entire unsupported length of the member on which it rests. As was pointed out above, consideration must be given to the type of load, i.e. uniform or concentrated, when determining the size of members to be used as skids and load bearing floorboards.

A concentrated load is a load which is not distributed evenly over the unsupported length of the member on which it rests. Rather, it is centered over a shorter segment of the member on which it rests resulting in a higher bending moment than a uniform load of similar magnitude.

Rigidity refers to a strength-property of materials which allows them to resist deformation such as longitudinal bending or racking. For example, a 20" diameter pipe with ½" wall thickness and length of 20' may be able to be picked up by a sling at its mid-length without any noticeable deformation while a 2" diameter pipe with wall thickness of 3/16" and length of 20' would droop noticeably if lifted with a sling at its mid-length. Items with no inherent rigidity, which are intended for export shipment, should be skidded or crated in order to facilitate handling and prevent damage.

Fragility is a term used to describe the degree of susceptibility to damage from the normal rigors of transportation, i.e. shock and vibration.

Bending moment, as used within the scope of this paper, refers to the product of a load acting over a distance to produce deformation of a structural member which is subjected to the load at some point along its length. The load may be uniform or concentrated as described above. As an example, consider a beam supported only at its extreme ends and subjected to a load at its mid-length such that the resultant of the load acting downward and the reaction foces at the extreme ends acting upward result in a sagging of the beam. What actually happens here is that the fibers of the beam in contact with the load are in compression while the fibers on the underside of the beam opposite the load are in tension. The bending moments described above can be written in the form of the following mathematical expression: BM = (W x L) ÷ 4 (Foot . Pounds), where W is the applied load, L is the unsupported length of the beam and the denominator 4 takes into account that W is a concentrated load. Similarly, the mathematical expression for the bending moment resulting from the application of a uniformly distributed load on the same beam is as follows: BM = (W x L) ÷ 8 (Foot . Pounds), where the denominator 8 accounts for the fact that W is a uniform load. In some calculations to determine bending moments on beams where the applied loads are neither strictly uniform nor strictly concentrated, it is convenient to use the denominator 6.

Stress is directly related to the bending moment. Mathematically, Stress = BM ÷ SM.

The term Section Modulus denotes a property of a particular beam cross-section. It is defined as the moment of inertia divided by the distance of the most remote fiber from the neutral axis, and is denoted by the symbol S or SM. A simple mathematical expression for section modulus is (b x d2) ÷ 6 where b is the width of the cross-section and d is the depth of the cross-section. It is clearly seen that Section Modulus is extremely helpful in determining the actual stress to which a beam is subjected under a known Bending Moment.

In this paper, the term "engineered" means that an analysis must be made of the types of loads and their magnitudes on the various structural members of crates, in order that their required dimensions may be correctly determined by calculation.

Marks and Numbers

The purpose of Marks and Numbers is to identify the shipper and consignee, to aid in safe handling, to provide general information and to identify hazardous materials. Never identify the contents unless it happens to be hazardous materials.

Marks and Numbers should be made in large (2" minimum height), block letters and numbers. All numbers such as gross and net weight and dimensions should be in both British and metric units. Marks and Numbers should be placed on three (3) sides of boxes and crates to insure that they can be seen, and these should be accompanied by the international symbols such as the umbrella denoting

"Keep Dry", the wine glass denoting "Fragile", and the link chain denoting "Sling Here". Marks and Numbers should be in indelible ink or paint which contrasts with the background.

Preservation

Some items, by their method of manufacture, are already preserved. Others require protective coatings such as oils, grease, paint or some other contact preservative which may be specified by manufacturing, engineering, quality control, or customer requirements. Some items, in addition to contact preservatives, require wrapping in wax paper or some other protective film.

In addition to contact preservatives, certain items may require additional protection from the elements, especially water, such as shrouds, liners and linings, wraps, VPI or VCI (moisture inhibitors) or desiccant materials. Wraps applied over a contact preservative must be grease proof and require a means of assuring that they remain in place during handling and transportation. Liners and lining are waterproof barrier materials applied between framing and sheathing to provide watershed and diversion capabilities for the items crated. When shrouds are applied, all projections in sharp corners must be cushioned to prevent penetration or abrasion of the shroud from either internal or external sources. The bottom edges of all shrouds should be fastened down with batten boards and additional restraints provided to prevent ballooning or flapping of the shrouds in transit. Adequate bottom ventilation must be provided either by spacing floorboards (3/8" minimum) or by drilling one 1" diameter hole for each 5 cubic feet of shrouded volume. When VPI/VCI materials are used, a sufficient quantity of treated paper must be used to completely enclose the item.

For general watershed and diversion protection, polyethylene films (4 mils minimum thickness), asphalt laminated kraft paper, filament reinforced polyethylene or polyethylene coated kraft paper may be used. When applying water vapor resistant liners or lining, such as asphalt laminated kraft paper or polyethylene films (6 mils minimum thickness), it is essential that they be sealed tight. Whenever VPI/VCI treated papers are used, they must be inserted within a tight enclosure, and the liner or lining must be sealed with waterproof tape. This is also true of desiccants.

Skids, Crates and Boxes

The materials used for skids must be Group IV (hard) woods such as Beech, Birch, Hard Maple, Hickory, Oak, Rock Elm, and White Ash. These woods have the greatest nail-holding power and the highest overall strength and resistance to shock. However, they are very susceptible to splitting. These woods should be used for skids, headers, load bearing floorboards and critical joists.

Group II and III are generally acceptible for exterior box and crate construction. Group II woods include Douglas Fir, Hemlock, Southern Yellow Pine, Tamarack and Western Larch. These woods are harder soft woods. They possess greater nail-holding power, greater strength and greater shock resisting capacity than Group I woods. They are more inclined to split and their grains often deflect nails. Examples of Group III woods are Ash (except white) Cherry, Soft Elm, Soft Maple, Sweet Gum, Sycamore and Tupello. These woods are similar to Group II in nail-holding power and beam strength but have less tendency to split and shatter under impact.

When using plywood as sheathing or floorboard material, type C-D interior plywood with exterior glue is the best economical selection.

Lumber Selection and Defects

A great deal of consideration must be given to a standard of quality of the wood selected for skids, crates and boxes. Often the cheapest grade of lumber will contain a large number of defects which are undesirable. Therefore, it may be necessary to select a higher grade of lumber to eliminate the majority of disqualifying defects. Generally, knots are not to exceed ¼ of the width of the member. No knot, regardless of size, will be permitted on the edge of any member. Obviously, this is because the member may be subjected to bending and a knot may fall in an area subjected to compression or tension. Cross grain or slope of grain, disregarding slight local deviations, along the general direction of the grain as related to the longitudinal axis of the wood member, should not be steeper than 1" in 15" of length. Wane is either bark or lack of wood on the edge or corner of a piece of lumber. Generally, wane is permitted on one edge only, but should not exceed 1/6 of the width, thickness, or 1/3 of the length of the member. Checks and splits, lengthwise openings from separations during seasoning, may reduce wood's resistance to shear. Checks and splits which extend through the entire thickness of the piece are not permitted. Shake is a separation along the grain, largely between the growth rings, which occur while the wood is seasoning. Shakes in members subjected to bending reduce the resistance to shear and therefore, should be closely limited in structural members. Decay, a disintegration of wood, results from the action of wood-destroying fungi. It seriously affects the strength properties of wood and its resistance to nail withdrawal. However, if it is determined that the total amount of decay beneath the surface does not extend beyond the surface outline, the dimensional limitations for knots apply. Warp along the longitudinal axis of the lumber should not be more than 1" in 8' of length. Warp, widthwise, should not be more than one-eighth inch in 4" of width. The acceptable range of moisture content in lumber used for skids, crates and boxes should lie between 12% and 19%.

No opportunity should be overlooked to utilize materials besides wood or plywood for crate, box or skid construction where practically and economically feasible. Steel alternatives for wood skid members may be the simplist substitution. For example, a 4" x 4" timber may be replaced by a 3" x 4.1 lb. steel channel iron or by a 3"x 5.7 lb. I-beam or by a 3" x 3" x ½" angle iron. Wherever material substitutions are contemplated for heavy loads, they should be engineered for the specific purpose intended.

Design

A proper analysis of forces to which skids, crates and boxes may be subjected must include consideration of compression, lateral thrust, impacts, repeated handlings, abuse, tension and inertial forces on the structure and the cargo. In this regard, it must be remembered that inertial loadings encountered aboard ship often exceed those encountered in the other modes of transportation.

Basic Design Criteria

Skids must be of Group IV woods and engineered for uniform or concentrated loads. Headers are to be bolted using washers and double nuts or upset threads. Skids are to be spaced not more than 48" apart and are to be of single piece construction or, if over 12' long, spliced according to an approved method. The ends of skids are to be chamfered and are to rest on chamfered rub strips. Cargo is to be bolted or tension banded to skids using engineering design methods for determining the sizes of bolts and tension bands. For very heavy loads, consideration must be given to the crushing strength of the wood or alternative material. Load bearing floorboards can be analyzed as beams using the formulas for bending moment to determine the required section modulus. The allowable stress used in the formula for section modulus can be taken as 1,000 psi. For hard woods, this allowable stress provides an adequate factor of safety. Where the distance between skids might exceed 48", it will be necessary to add an additional skid. The bending moment on the load bearing floorboards, then, should still be calculated on the basis of the length between outside skids.

Example: A 15,000 lb. load is to be supported on skids whose length has been determined to be 15'. Calculate the required timber size.

Using the formula BM = (W x L) ÷ 6, the bending moment will be 37,500 Foot · Pounds. Calculating the required section modulus = BM ÷ allowable stress = 450 cubic inches. Remember to multiply the bending moment by 12 in order to convert Foot· Pounds to Inch · Pounds. Since section modulus = (b x d^2) ÷ 6, you can solve by trial and error by substituting cross sectional dimensions as follows: Try a 10" x 10" timber. This yields a section modulus of 166.7 cubic inches. Two (2) skids of these cross sectional dimensions yield of total of 333.4 cubic inches. It is obvious, then, that two (2) 10" x 10" timbers do not provide the required section modulus (450 cubic inches) for a quasi uniform load. The next logical choice would be two (2) 12" x 12" timbers. Note, however, that if the load is truly uniform (BM = W x L ÷ 8), the required section modulus is only 337.5 cubic inches. The two (2) 10" x 10" timbers then come very close to making the required section modulus, and with the factor of safety provided by using the allowable stress of 1,000 psi, the two (2) 10" x 10" timbers would serve satisfactorily for a strictly uniform load. For most hard woods, the allowable stress of 1,000 psi provides a comfortable factor of safety of approximately 1.5. In calculating the basic stresses for structural lumber, impact loading is generally ignored, but long-time loading and a safety factor are considered. A piece of wood will carry less load for a long time than it will for a short time. Consideration should be given to the expected interval of storage, i.e. short time verses long time. In addition, if research or historical data indicate that a particular port of loading and/or a particular port of discharge have a higher frequency of cargo claims due to rough handling, timber sizes may be increased by applying an appropriate factor of safety or by requiring that the actual timber size will be one full unit larger than derived by calculation. Remember that rub strips must be applied to the skids. An added precaution against the accidental loss of a rub strip is the use of reinforcing straps from the rub strip to the underside of the skid members.

Crates

In general, Group II and III woods are to be used in crate construction except for skids, headers and occasionally load bearing floorboards which should be Group IV woods. All nails used are to be cemented coated. Barbed, screw, or serrated nails are occasionally used, but for practicality and economy, cement coated nails make the best selection.

All skid members in crate construction generally follow the same construction requirements as skid members used as skids alone. However, skid members sizes in crates can be substantially smaller in cross sectional area because, in crate design, it is assumed that a large part of the load imposed by the contents is carried by the side panels acting as trusses. Therefore, large skids are not necessary as load-carrying members because the sides act integrally with the skids in this function. While this assumption results in smaller size skids, it does not permit handling and moving a loaded base alone without the sides and ends fastened in place.

Crates may be either open or closed (sheathed). It is not always necessary to sheathe a crate if the contents do not require a degree of protection from the elements. Opened crates require less lumber and, therefore, if properly designed, can be less expensive to fabricate. This may not always be true, and, to achieve greater economy, it is necessary to make an engineering economy analysis to determine whether an open crate or a sheathed crate is more economical in terms of labor and cost of materials. This paper will discuss the design and construction of open crates since the contents impose loadings on structural members which must be determined either by graphical analysis or engineering methods of resolving forces. The sides and ends, as well as some members of the base, are considered as part of a bridge truss. The size of members are determined by analyzing this truss. The Howe truss, with its parallel upper and lower chord and its vertical and diagonal members, has the same general pattern as the frame members of the side of a wooden crate. Since the truss is a framed structure composed of straight members, the stresses in the members due to loads must be either compression or tension. The magnitude and character of the stress in each member can be determined. The required size of all tension members in the truss is determined by the formula $A = P \div f$; where A is the required cross sectional area of the member in square inches, P = the total load in pounds (as determined from the resolution of forces or from graphical analysis), and f is the working stress in pounds per square inch (in tension parallel to the grain). With support at each end, the major stresses in strut and lower frame members are tension stresses and those in the diagonals and upper frame members are compression stresses.

Compression members must be designed as columns and column formulas must be used to determine their sizes. In crate design, there are essentially three column formulas. The selection of the proper formula depends on the unsupported length of the column and, columns are therefore classified as short, intermediate and long. It is beyond the scope of this paper to discuss the various factors and the application of these in the appropriate column formulas. The Wood Crate Design Manual (Agriculture Handbook No. 252) is an excellent reference text for the more determined crate builder. It must be remembered that it is most probable that crates of similar dimensions will be stacked, one upon the other, aboard ship or in storage. If crates are of exactly the same overall dimensions, the corner posts will carry the superimposed loads as columns. These anticipated superimposed loads must be included in the strength analyses. If superimposed loads consists of smaller crates or other types of loads that do not rest upon the corner posts, the structural members of the top panels will be subjected to bending as well as to compressive loadings resulting from handling especially with slings or chains. Therefore, these structural members must be analyzed as beams subjected to bending and axial compression. If highly concentrated loads (in the center 1/3 of the crate) are to be crated, the skid depth must be increased. This, however, will be apparent if the side panels are properly analyzed.

Crates should not be higher than they are wide. Side panels are to be divided by a suitable number of vertical members so that diagonals are as close to a 45° degree angle as possible. Top and bottom horizontal and vertical members are to be through members (extend to full extent of length and height dimensions). All end panels are to have one or more diagonals. Avoid splicing members less than 16' long. Splicing, where required, must be accomplished using an approved method. Intersections of diagonals are to be reinforced using plywood gussets or lumber bridge.

If the crate height exceeds 72", a through intermediate horizontal member is required at mid-height of the crate. Side panel width (i.e. between vertical members) must not exceed 42". The geometry of the panels must be altered to add an additional panel so that panel widths do not exceed 42". Lower horizontal members are to rest on the ends of the floorboards with notches cut in headers and load bearing floorboards to accept the lower horizontal members. Vertical and diagonal members must be in the same plane and to the outside of the horizontal members. Side panel diagonals should slope from the top of a vertical member down to the skid base.

End panel design should not exceed 42" in width. End panels for crates having a skid width in excess of 42" will require intermediate vertical members. End panels for crates having more than two (2) skids will require a vertical member above each skid. Filler strips will be required to allow end panel vertical members to rest across the ends of skids. The filler strips will fill the void between the headers and the lower horizontal members.

Top panels are critical since most members will be in compression under the influence of various types of materials handling equipment and superimposed loadings. Top panel lateral and diagonal members will be in the same plane and rest upon the longitudinals and extend to their edges.

Where protection from the elements is essential or to meet customer requirements, a sheathed-crate, either 3/4" lumber or ½" plywood (C-D, exterior glue) sheathing can be used. Reinforcement strapping is required on all corners and at all intermediate, horizontal and vertical member junctures after the crate is assembled.

Where crate items require additional protection from the elements, as discussed earlier in this paper, appropriate steps must be taken in the early stages of crate construction to apply the required crate liner between internal framing and external sheathing. If the crate is not to be watertight, sufficient ventilation must be provided. If the contents are to be provided with waterproof protection, the item must be properly and totally wrapped with either asphalt laminated kraft paper or polyethylene film and sealed with waterproof tape in addition to the crate liner material.

The strength and rigidity of crates are highly dependent on the fastenings. Nails, lag screws, bolts, screws and metal connectors are the most important fastenings in crate construction. This paper will cover briefly nails and nailing rules. Refer to "Nailing Better Wood Boxes and Crates"by L. O. Anderson, Agriculture Handbook No. 160, for a more complete discussion of nails and nailing rules. Cement coated nails are being more widely used today than ever before, primarily, because many crate builders use automatic nailing equipment which lend themselves more readily to the use of belted cement coated nails. Nails should be driven through the thinner member into the thicker member where possible and should penetrate both members leaving a minimum of $\frac{1}{4}$" for clinching. Clinching is one of the best methods of increasing effectiveness of nails. It is used almost entirely in the fabrication of crate panels, except when frame members or other crate parts are greater than 2" thick. Clinched nails have 50 to 150 percent greater withdrawal resistance than unclinched nails when driven into drywood. Pre-drilling the wood before the nails are driven may be necessary to prevent splitting in very dense woods or with nails of large diameter. Nails should not be over driven. They should positioned no less than the thickness of the piece from the end or half the thickness of the piece from the side edge. When members, both of which have 3" or greater thickness are joined together, bolts are to be used. Nails should be staggered to prevent any two nails from entering the same grain line of any board. Follow approved nailing techniques as outlined in the handbook described above.

Where practical, the use of tension bands will provide an added degree of a reinforcement to the crate assembly. Tension bands should be tight enough to cause a light crushing of the ends of the main structural over which they pass. Tension bands should be stapled to prevent their movement during transportation and handling.

A crate can be assembled before or after loading the item on the skid base, whichever is appropriate, but internal blocking and bracing of the item in the crate must be done at the time the item is set on the skid base. The blcoking and bracing must be of substantial design and construction to prevent the free movement of the item within the crate due to the forces encountered in transportation and handling. Blocking and bracing are not to be taken lightly. The inertial loading on the crate and its content aboard ship often exceeds twice that encountered in the other modes of transportation.

Boxes

Boxes are not to be used for loads in excess of 5,000 lbs. or lengths in excess of 16'. Group II and III woods are to be used. All nails are to be cement coated. Boxes which exceed 200 pounds gross weight will require 2" x 4" rub strips. When any unsupported span of top, end or side lumber exceeds 36", additional interior battens or cleats will be required. Cleated plywood boxes for loads over 1,000 pounds must have load bearing bases with properly size skids, headers and rub strips and are to be nailed in accordance with the preceeding nailing rules. All boxes require tension bands sized according to the load and arranged in two directions passing over cleats or battens.

Bands should be stapled to prevent movement and should be tight enough to cause light crushing of the corners of the members over which they pass. If a boxed item is of large mass (weight), appropriate filler or cushioning materials filling the voids around the item may not be effective. Therefore, dunnage or blocking and bracing, where practical, should be used. Where the item boxed requires protection from the elements, the internal surfaces of the box may be lined as described under crates and the item itself should be properly preserved.

Unitization and Palletization

Unitization provides a more economic package than individually shipped pieces. It also provides ease of handling and storage and provides a greater degree of protection than individual units. Pipe and conduit are easily bundled. They may require special plastic end protectors to protect internal or external threads. The ends of bundled pipe can be boxed or completely covered with double or triple wall fiberboard secured with tension bands. In addition, long bundles may be bucked. Bucks are wooden frames which surround the girth of the bundle. They are to be nailed at the right angle junctures and tension banded. Tension bands are to be stapled to prevent accidental loss of the bands. Bucks should be located at five foot intervals maximum.

Lumber and paneling are usually bundled. Care must be taken to avoid over-tensioning bands which will mar their edges. Paneling should be covered with water-resistant paper or polyethylene film.

Coiled wire can be bundled. Because of the historical poor bundling methods used for wire, substantial losses have occurred. Consignees frequently use coiled wire to make nails, screws and bolts. Kinked or crooked wire will interfere with the extrusion process and substantial claims may arise.

Lumber and paneling can also be bundled. Care must be taken to insure that tension bands do not damage the items bundled. Coiled wire can be bundled. The usual problem encountered here is when tension bands or wrapping wires come off allowing the coils to become disarranged or crushed. The consignee usually has a specific machine set up for manufacturing items such as bolts, screws and rods. Hence, damage to the coils can result in a claim.

Palletization is best for similar size fiberboard cartons, drums, and coiled sheet steel. Do not overload or stack the items to high. Overloaded pallets break. Too high of a stow can result in crushing of the cartons in the lower levels. Then further damage occurs when the otherwise neat stacking falls apart. In-transit handling of broken down pallets usually results in further abuse. Fabric or steel banding is usually one of the best methods of securing an entire unit load. It may be advisable to employ vertical and horizontal corner protectors to eliminate banding damage. Shrink wrapping provides a high strength securement to a palletized load. It can also be applied to protect against contact with water.

Palletized drums must be fitted with partially-framed top to allow for safe utilization of tension bands. Tensioning of the bands must not cause damage to the pallet load bearing boards or to the cargo. Fiberboard sheets across the top and partially down the sides of drums will serve to keep the tension bands positioned and provide a level surface for stacking other pallets.

In general, old or badly worn pallets should not be used. Pallets can be repaired. Some companies use secondary re-usable pallets with good success. Still, old wood and loose nails can result in broken pallets which invite further abuse in handling.

Container Problems

Containers are subject to the same shock and vibration forces as over-the-road trailers and railroad cars. In addition, very high inertial loadings are experienced by the contents due to the pitching, rolling, heaving and surging of a ship at sea.

The major causes of damage to containerized cargoes are improper stowage, inadequate dunnage, lack of securing, overloading and poor weight distribution. Any assumption that the container is a substitute for safe stowage and handling is an invitation to disaster. A good rule to adopt is, "Stow and Secure for the Worst Conditions".

Containers may travel to seaport by railroad. Constant vibration and occasional sharp humping forces must be taken into consideration. When loading a container for sea transport, always consider the six basic ship motions; yaw, heave, sway, pitch, roll and surge. Occasionally heavy seas will have a pronounced effect on cargoes in containers.

Blocking and bracing is essential to the safety of the cargo. Materials commonly used for securement are lumber, plywood, strapping, fiberboard and inflatables. Lumber can be used as a filler, for decking, blocking and bracing and constructing partitions. Plywood can be used as a partition, divider and auxiliary decking. Heavy duty strapping can be used to separate cargo and to tie down heavy and akward items. The strapping must be firmly anchored and properly tensioned for the greatest effectiveness. Fiberboard is available in sheets, rolls and structural shapes and can be used for light-duty bracing, as dividers, decks and partitions. Inflatables are available in paper or rubber and may be re-usable or disposable. They are expensive and not recommended for voids in excess of eighteen inches. Inflatables are used for light and medium duty bracing.

Consideration is another major cause of damage to cargo. It is not generally known that fiberboard boxes and crate or dunnage lumber contribute to the generation of water vapor in sea containers. Furthermore, in changing temperature and humidity conditions, condensation can form within containers and drip down from the overhead onto moisture-sensitive packaging and cargoes. These conditions can be remedied, in part, by using seasoned lumber for dunnage and crate construction. Adequate (pre-packaging) preservation of moisture-sensitive items can be the best method of protecting against water damage. Overwraps such as canvas or polyethylene sheeting will serve to divert condensation which drips from the overhead. It may be necessary with some types of cargoes to provide completely waterproof liners or wraps inside of which is VCI/VPI or desiccant materials. Cargoes are more likely to be subjected to condensation and water damage over a long time internal such as long transportation distances or long duration storage. Therefore, it is incumbent upon the shipper to consider all, these factors when preparing cargo for export.

The annual report of the cargo loss prevention committee of American Institute of Marine Underwriters contained valuable advice on the use of containers in shipping. Extracts from the report follow:

Normal reaction is for a shipper to use domestic packaging and consider the container as an additional protection. It is our experience that containers may leak and the cargo must be protected against moisture and water damage. Additional emphasis is needed on the use of dunnage and tight packing in containers.

Several offices report that losses have been experienced due to defective containers. Many shippers don't employ adequate container inspection procedures. It appears that some insureds think that because their goods are containerized the packing can be of a minimal nature. Our experience continues to show that such minimally packed goods are likely to be damaged by rain and/or sea water-when shipped in poorly maintained containers.

Many shippers continue to use containers as a means of packaging and claims due to crushing, water damage and improper or inadequate stowage, especially of heavy machinery, continue to be a problem.

Losses have also been experienced where shipments of high value electronic units have been made in containers. The units are secured on lumber bases which in turn are then secured to pallets with the units being entirely shrouded in polyethylene sheeting which also encompasses the wooden base to which the units are attached. It was found that these wooden bases were constructed of green or improperly dried lumber, which gave off moisture, which became trapped in the polyethylene shrouding. During the course of transportation fluctuations in temperature caused this moisture to condense on the exposed metal areas of the electronic unit, causing serious damage.

We have also observed that machinery which is top heavy has on occasion been stowed in containers without adequate bracing and precautions to prevent the machine from tipping On one occasion a container loaded with tractor engines arrived at destination with every engine damaged, as a result of their center of gravity being high. Although the engines were braced along the floor of the container, and would prevent lateral or fore and aft movement, there was no bracing up higher and with their relatively high center of gravity they tipped, coming in contact with one another, causing damage. Shippers should be cautioned to be aware of the center of gravity of machinery and other heavy or odd sized pieces being shipped in containers and adequate bracing to prevent tipping should be done.

Although containerization of cargo continues to grow worldwide, the packing of goods into house to house containers by shippers, as well as securing of the goods in the containers, continues to be a cause for concern. We find in many instances cargo to be damaged by condensation within the container and/or by extreme heating or freezing. Once a container is loaded and the doors securely shut, the only way in which the outside weather can affect the cargo is through changes in the temperature.

Unless the internal temperature of the container is controlled by mechanical or other means, the temperature of the air inside the container will follow the temperature of the air outside. Extreme fluctuations in the air temperature can cause condensation to settle on the cargo; if the container is exposed to very high temperatures, the cargo within the container may sustain damage or, if the container is exposed to extremely low temperatures, the cargo can be damaged, such as change in the chemical state of some goods (e.g. drugs) making them useless and sometimes dangerous, or freezing and spoiling of fresh fruit, vegetables, etc., freezing of bottled liquids with subsequent bursting of bottles.

Taking into consideration the anticipated voyage, its length, temperatures normally expected to be encountered, consideration should be given to stowing goods which are susceptible to damage by extreme heat or cold to shipping them in insulated or refrigerated containers where temperature inside the container can be controlled to some degree.

In other cases certain precision machinery or electronic equipment may require the use of VCI, VPI or powder desiccants. It must be pointed out, however, that in order for these moisture control measures to take full effect, the packaging and packing must be totally sealed.

Shipping Losses and Insurance

Under normal conditions of domestic rail shipments, material loaded in open cars is inspected by railroad representatives before the car is moved and the railroad assumes responsibility. In closed cars,however, the Association of American Railroads rules are not mandatory and are used only for guidance. Closed cars are not ordinarily inspected, but if damage occurs to the contents, they are inspected at the destination to determine the cause before a settlement is made.

Export shipping companies are specifically exempt from most forms of liabilities under the laws of many countries. The exceptions to this usually include loss or damage due to negligence in proper loading, custody, or delivery of the goods. The shipper or consignee must assume responsibility for all remaining risks during the shipment.

To prevent loss to the shipper, a form of marine insurance covers these losses. Marine insurance may be obtained to cover such perils as pilferage, theft, and leakage, as well as loss or damage if the ship should sink, burn, or be involved in a collision. However, the more hazards covered by the policy the higher the rate, so it is not economical to pay for broader protection than is actually required.

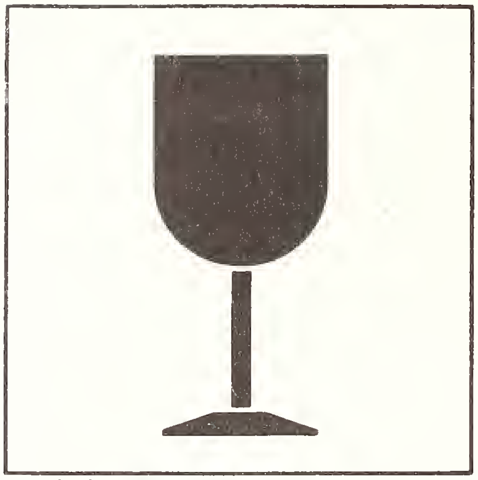
Rates in marine insurance are rather complex and are not fixed. They depend, among other things, on the type of vessel, the route, the perils insured against, the type of packing used, and the loss record of the shipper. This latter factor, of course, reflects the type of container and the method of blocking and bracing used by the shipper, because well-constructed and well-packed crates will normally receive little damage during the voyage. A shipper who uses adequate containers pays lower rates. Underwriters keep statistical records of shippers they deal with and allow lower rates for those with good records.

Damage and Claims

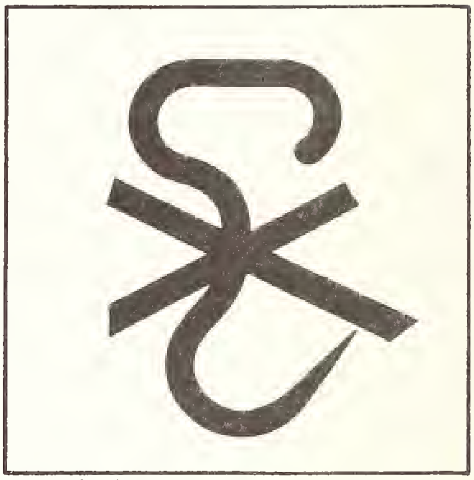
It has been the writers experience that too frequently one or more of the three steps in preparation for shipment are less than adequate and will inevitably result in damage of one type or another to the cargo. Precision machinery shipped long distances or subjected to long term storage without adequate preservation will rust. Electronic equipment shipped without serious consideration being given to adequate protection from vibration and shock will sustain physical damage. Items requiring a degree of dimensional stability (rigidity) will be damaged if the packaging and packing is not properly engineered and constructed. With regard to containerization (sea containers), a tremendous number of cargo losses occur each year due to ignorance of the proper use of these containers, specifically the requirement for adequate blocking and bracing and protection of the cargo against condensation.

A major marine insurance company has made an analysis of principal causes of losses and found that, in the Handling and Stowage Category, 39% of all losses are due to container damage including breakage, leakage and crushing. In the Water Damage Category, 10% of all losses are due to fresh water, sweat and salt water damage. Nevertheless, it concludes, 70% of all cargo losses are preventable.

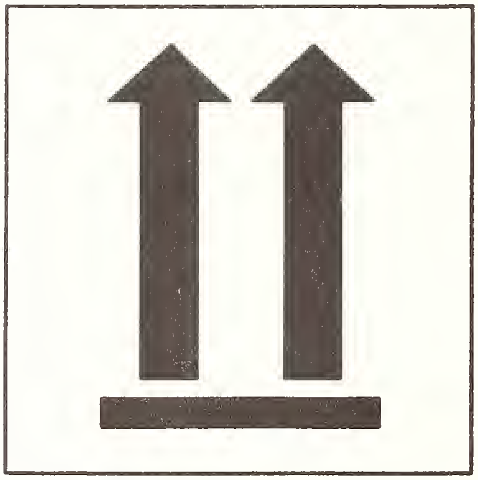
INTERNATIONAL SYMBOLS



Fragile, handle with care

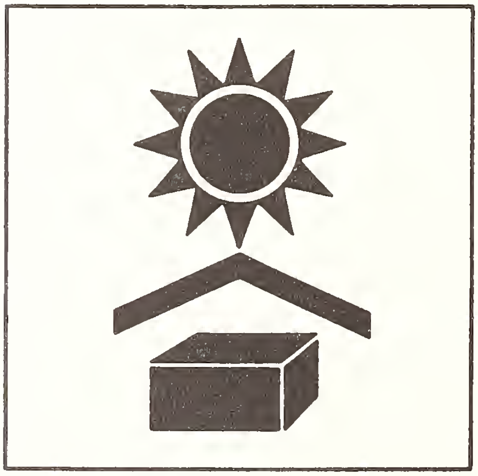


Use no hooks

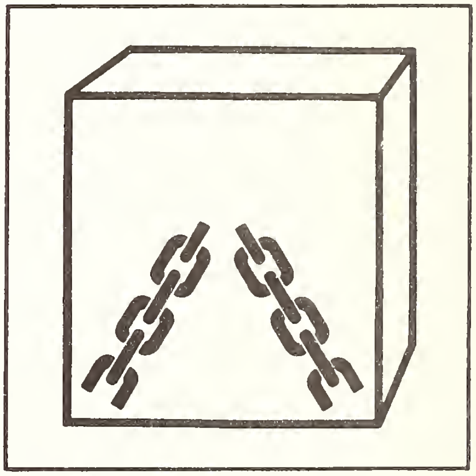


This way up

Keep dry

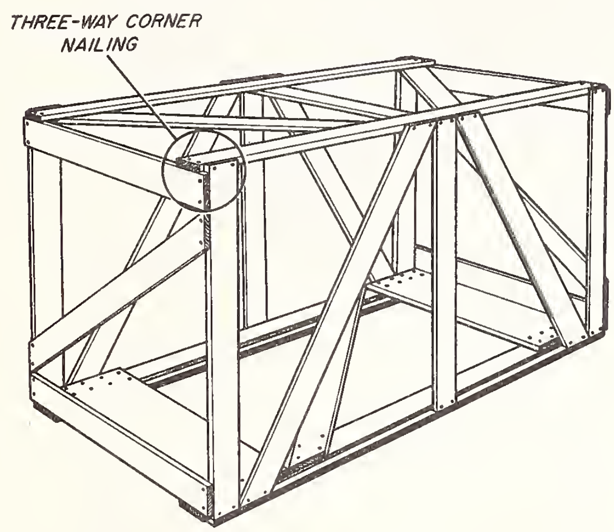


Keep away from heat

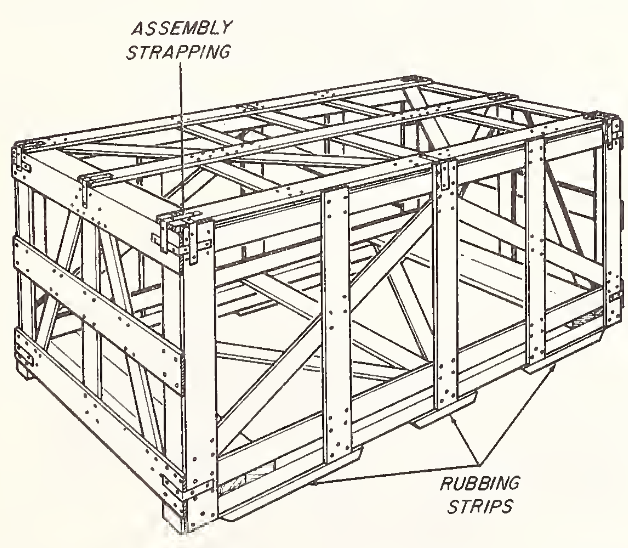


Sling here

OPEN CRATES



'igure 42.—Light-duty open crate (loads to 200 pounds) with three-way corner nailing, the key to proper assembly nailing of crates of this type.



igure 41.—Limited-military type open crate for loads to 2,500 pounds, designed for the shipment of light, bulky items that do not require a waterproof container.

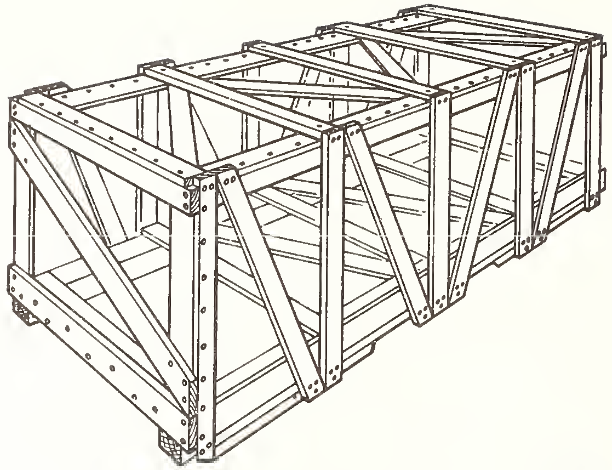


Figure 1.—Typical open crate.

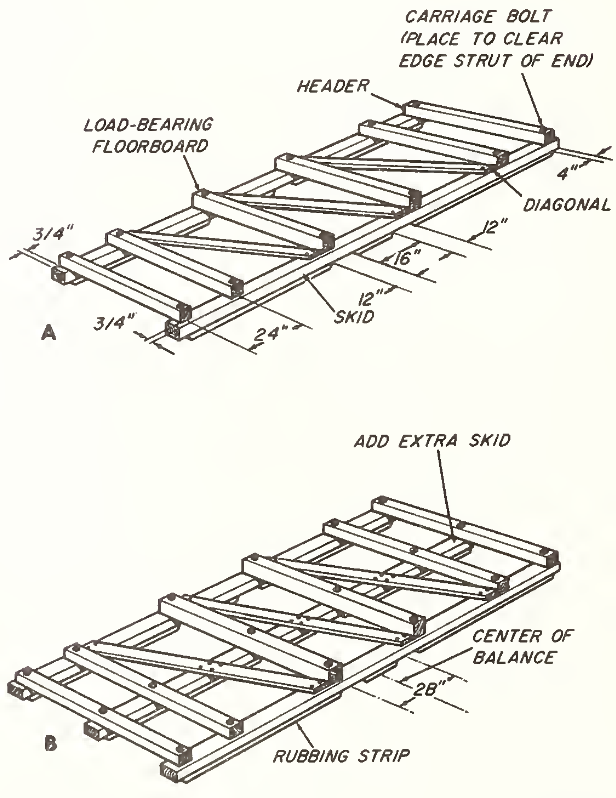
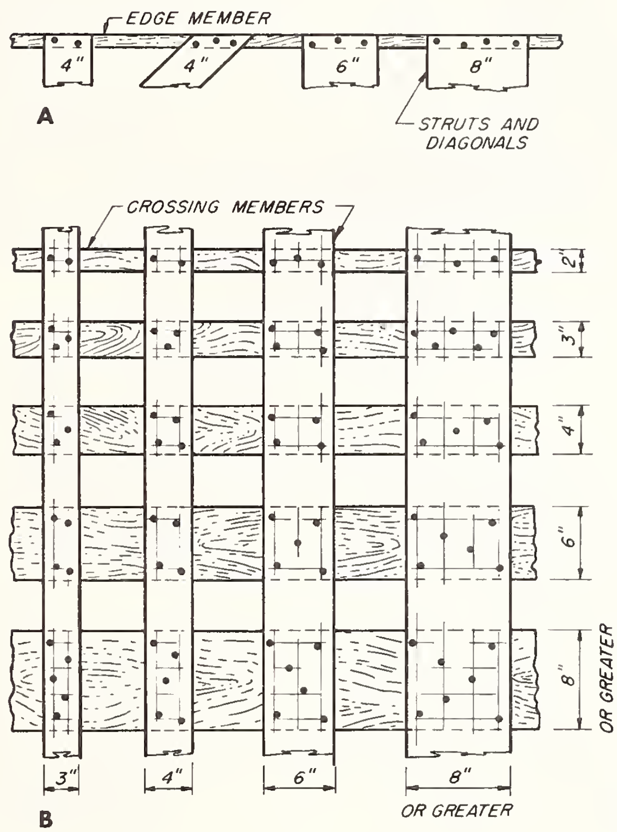
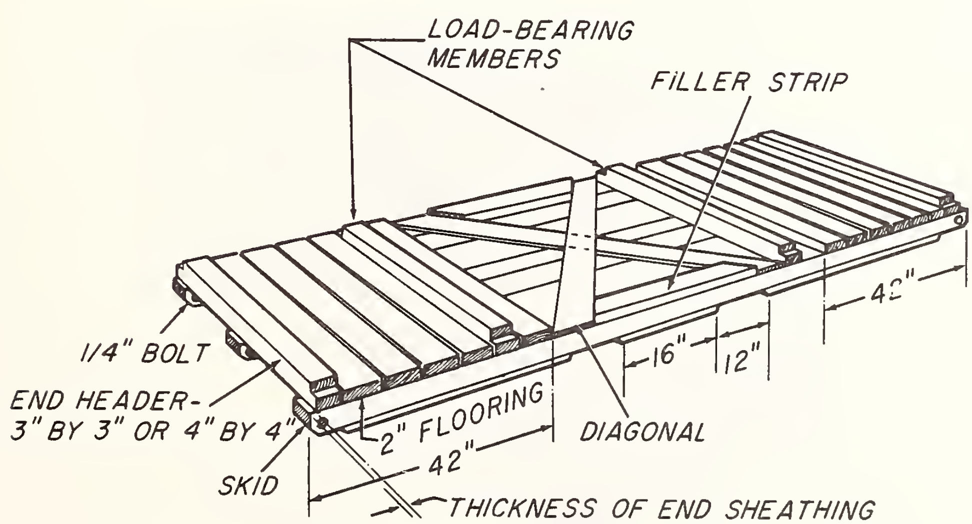


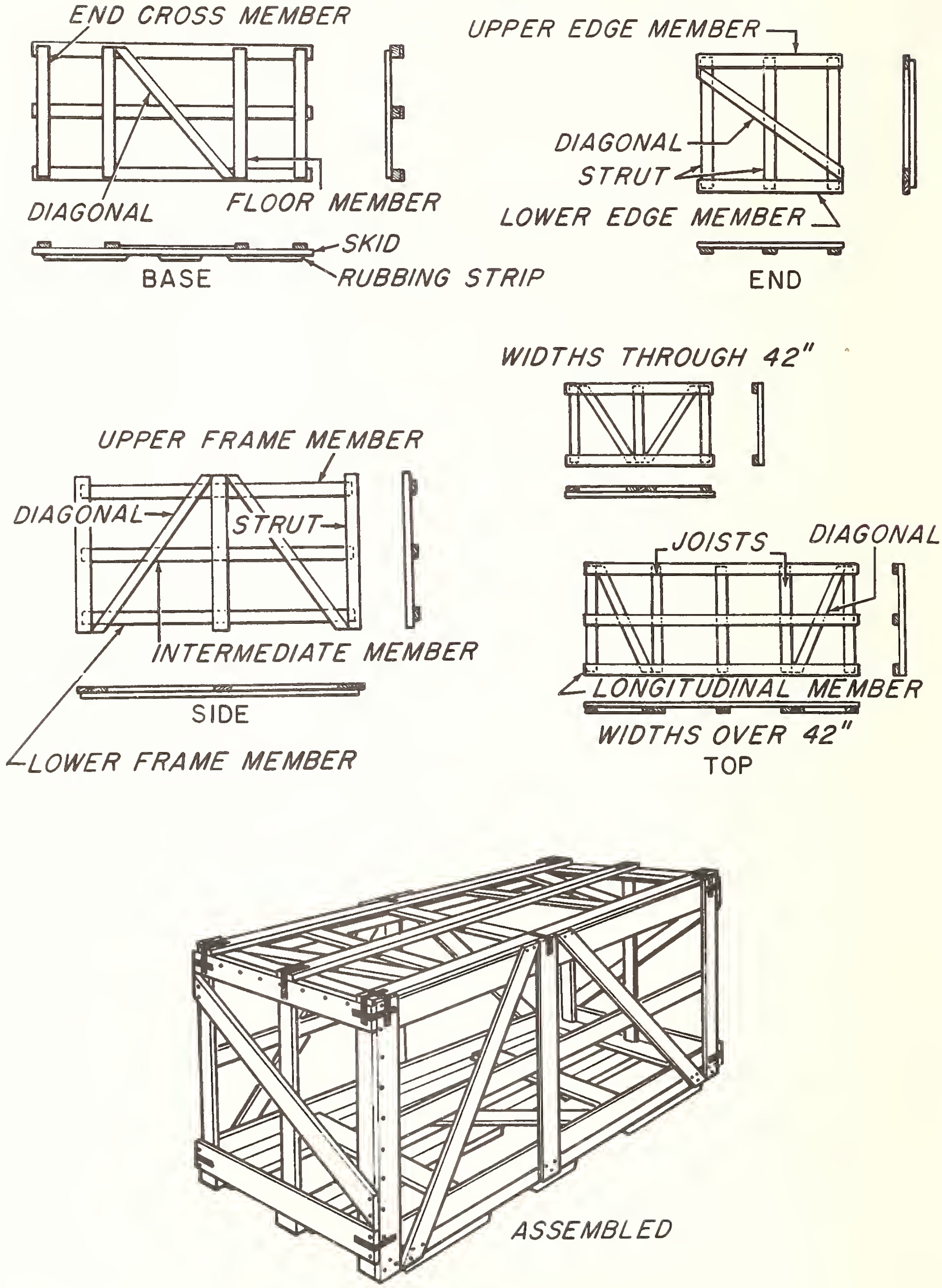
Figure 51.—Bases for light-duty open crates: *A*, Through 42 inches wide; *B*, over 42 inches wide.

TYPICAL NAILING PATTERNS



TYPICAL SKID FOR OPEN TYPE CRATE




END CROSS MEMBER
DIAGONAL
FLOOR MEMBER
SKID
RUBBING STRIP
BASE
UPPER EDGE MEMBER
DIAGONAL
STRUT
LOWER EDGE MEMBER
END
WIDTHS THROUGH 42"
UPPER FRAME MEMBER
DIAGONAL
STRUT
INTERMEDIATE MEMBER
SIDE
LOWER FRAME MEMBER
JOISTS
DIAGONAL
LONGITUDINAL MEMBER
WIDTHS OVER 42"
TOP
ASSEMBLED

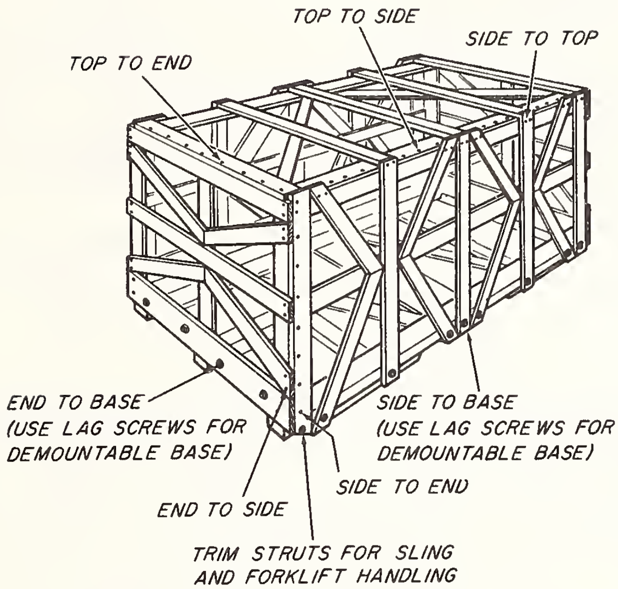


Figure 58.—Assembly of light-duty open crate.

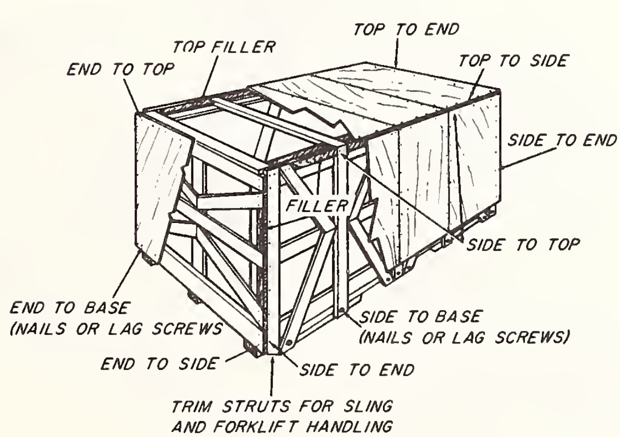


Figure 59.—Assembly of covered style light-duty open crate

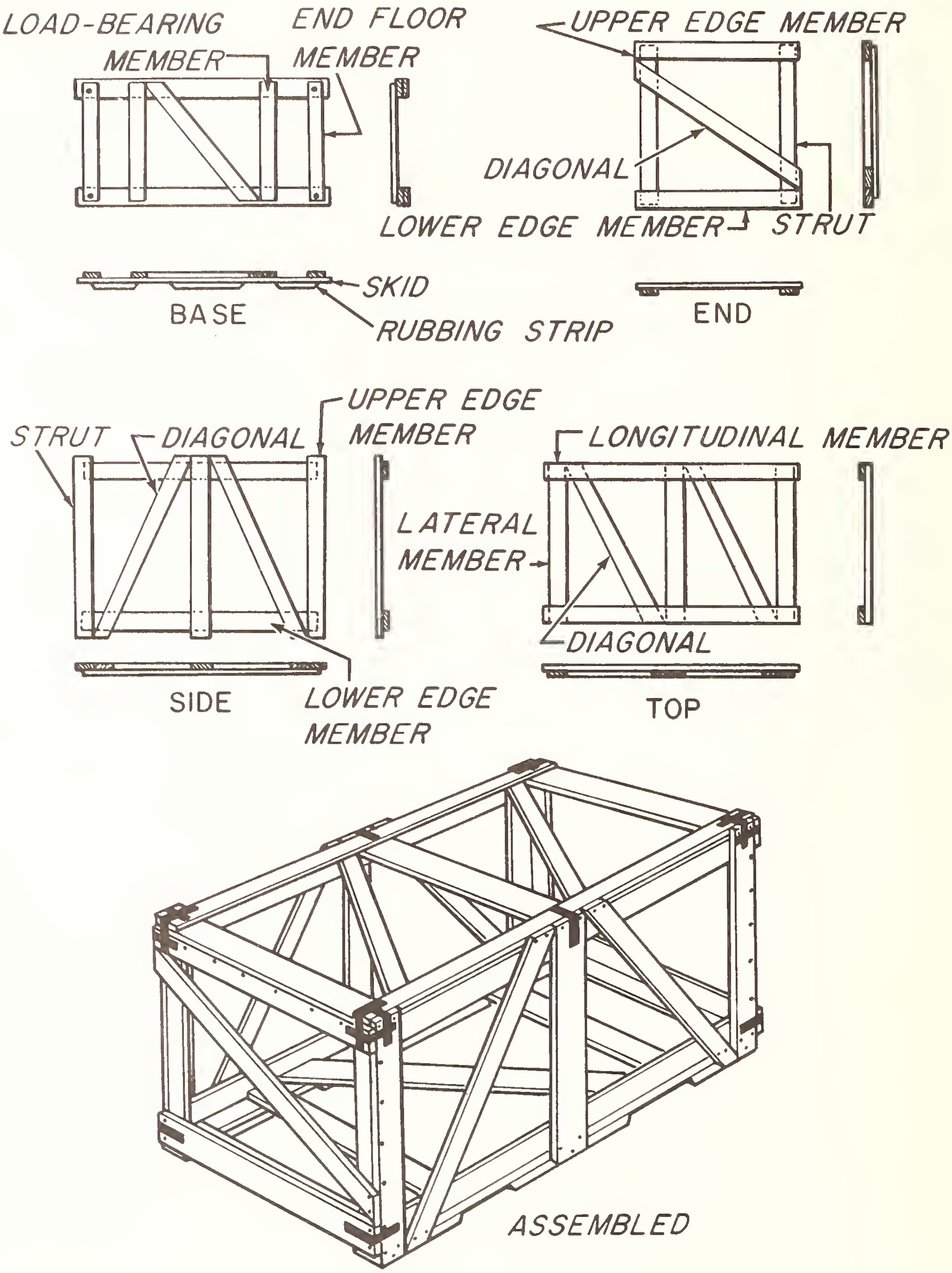


Figure 47.—Style A limited-military type open crate.

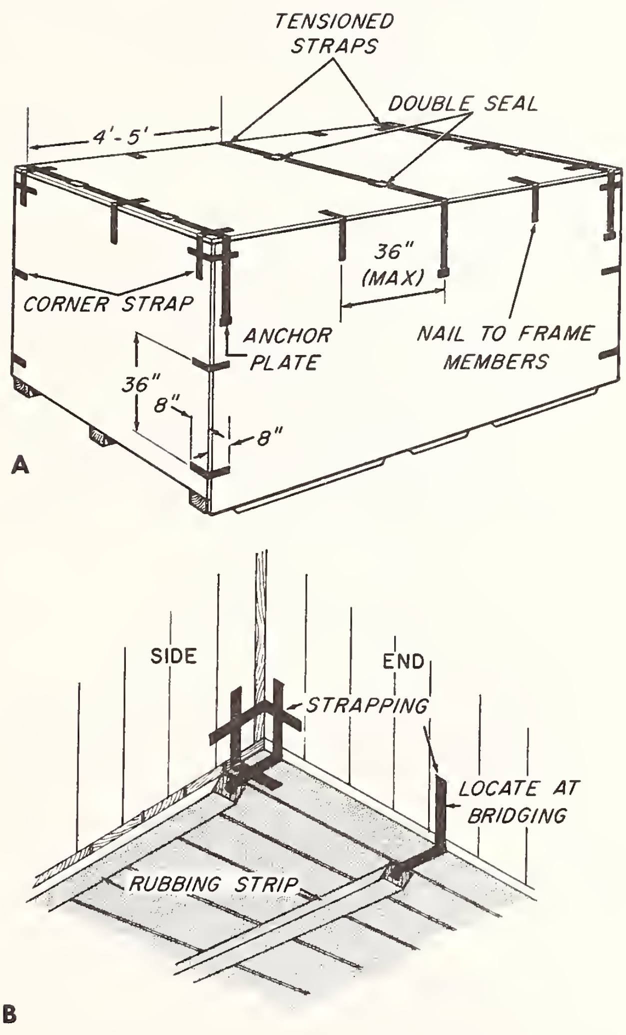


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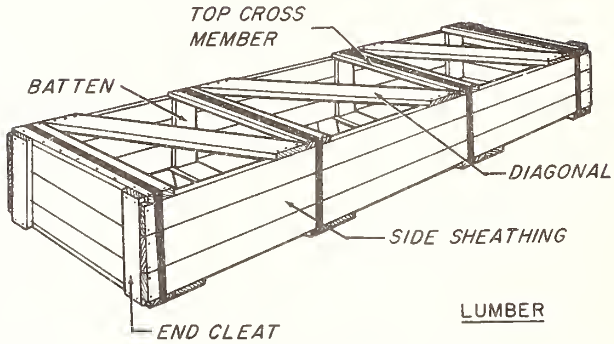
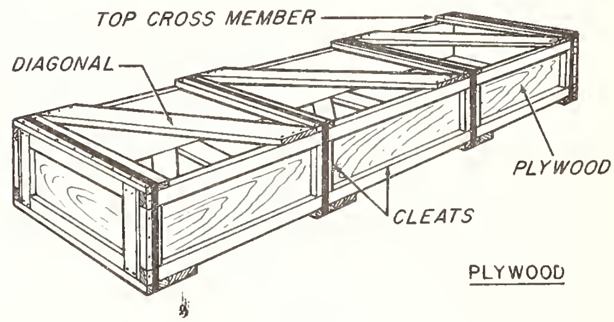
Figure 10.—Strapping for crates: *A*, Sheathed crate with skid base; *B*, additional strap for sill-type base.

TYPICAL SHALLOW CRATE





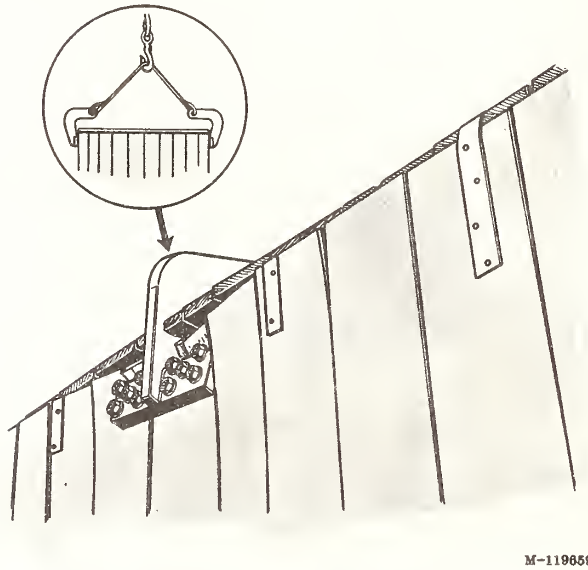


Figure 2.—Crate damage caused by a grabhook when there was insufficient joist support in the top of the crate.

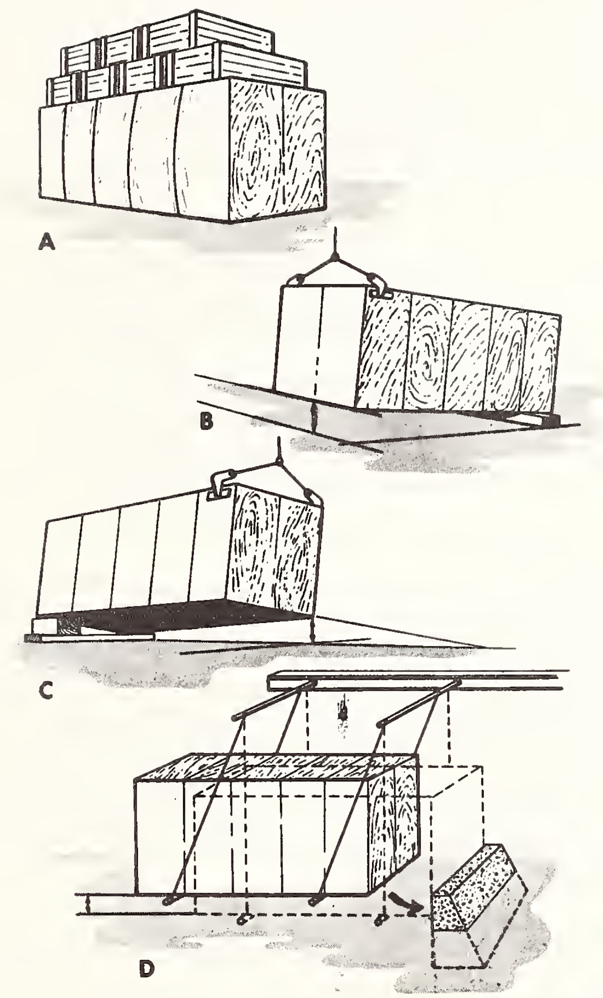


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Figure 64.—Rough-handling tests for crates: *A*, superimposed load; *B*, edgewise drop; *C*, cornerwise drop; and *D*, pendulum impact.

PACKAGING SOFTWARE FOR SHIPMENT AND STORAGE

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Abstract: The proliferation in the use of computers in all types of equipment has led to a dramatic increase in demand for computer software. Software is distributed on many different media including paper, magnetic materials and semiconductor memories. Each medium has its own handling requirements. This paper will discuss packaging, preservation methods and special considerations involved in preparing computer software for shipment and storage. Emphasis will be given to techniques designed to prevent damage to the software due to temperature, humidity, static discharge, electromagnetic emissions and other transportation environments.

Key Words: Error correction; error detection; magnetic media; memory; semiconductor memory; storage media

As computers become more complex and software costs increase the importance of storing and transporting software without failures becomes more evident. Failures due to lost data on the various storage media can be extremely costly. This paper will attempt to describe the various techniques utilized in preparing software for shipment and storage. Various storage media will be discussed, including semiconductor and magnetic media. In addition, methods by which software is internally prepared to help eliminate errors and detect and correct them when they occur will be covered.

The term "memory" refers to a medium capable of storing information and providing access to that information. Semiconductor memories are very commonly used in present day computer systems. These devices are programmed to represent either stored data or a part of the program to be carried out by the cpu. This is accomplished by storing a pattern of bits, or binary digits, at a specified location, or address in memory. The device is said to contain one byte of information at that address and the contents of that address can be read at any future time to obtain the same bit pattern that was originally stored.

Information in a semiconductor memory is specified in terms of 1's and 0's. Electrically, this is presented as the presence or lack of a charge, voltage or current, depending upon the type of device. Over the past several years the speed and density of these devices have increased to a point where a semiconductor memory may contain more than 64,000 bits on a single chip. Since a single bit error can cause an incorrect answer or a program halt, it is very important to exercise care in the handling of these devices in order to avoid damage or the loss of data.

Semiconductor memories are subject to two major types of errors. The first type is called a "hard error." This type of error is the result of permanent damage to the chip. The second type is called a "soft error." This type of error is the result of temporary alteration of data without permanent damage to the chip.

Hard errors are commonly caused by exposing the chip to an environment in excess of the limits specified by the manufacturer. The resultant failures may be mechanical or electrical. The most common causes of hard errors are excessive temperature and voltage, either steady-state, transient or cycling. Humidity, vibration and shock also contribute to device-damaging hard errors.

Soft errors in semiconductor devices have only been recognized for about three years. They are characterized by random errors which appear and disappear at several locations. For example, a byte is found to contain an altered bit pattern. When the memory cell is tested, it works normally, but the problem reappears at a new address. Soft errors have become more prevalent in the newer, denser semiconductor memories. Studies have been made and reveal that soft errors are caused by ionizing radiation which releases fast moving charged particles in the chip. It is important to note that in soft errors, the chip is not permanently damaged, but the information contained in the device may be altered.

In recognition of the possible causes of failure in semiconductor memories, most manufacturers exercise great caution in packaging these devices. Many follow the guidelines set down in MIL-M-555565 and subsequent revisions to prepare semiconductor memories for delivery. The general provisions of MIL-M-555565 are for physical protection and field force protection of the chip. The following is a discussion of these provisions and several ways in which they may be implemented.

Physical protection refers to the design of carriers, wrapping and cushioning to protect the device against several types of physical damage, such as vibration, shock, moisture and corrosion. In general, the specification requires compliance with MIL-P-116 for physical protection. It is important to note that in providing for physical protection of the device, care must be taken not to introduce or use materials which generate an electrostatic charge. The carrier must also be designed to maintain physical separation of the device leads.

Field force protection refers to the design of wrappings or bags to protect the device against several types of forces. These include electrostatic forces, electromagnetic forces, magnetic forces and radiation forces. The specification generally requires the use of materials conforming to MIL-B-81705 for field force protection. These materials are intended primarily as protection against static charges. They have a permanent conductivity on all surfaces to permit a controlled bleed-off of static charges to ground without producing a spark. For protection against other field forces, the specification recommends overwrapping the devices with additional layers of material with suitable shielding properties. Aluminum foil wrapping is recommended for electromagnetic protection, enclosure with ferrous metals for magnetic protection and enclosure with lead or lead filled compositions for radiation protection.

Protection against electrostatic forces is the most important of the four provisions. Static charges generate tremendous amounts of energy and noise which are primary causes of hard errors. Static charges may range between 5000 to 10,000 volts. In designing protective wrappings against electrostatic forces, the designer must consider: 1) internally generated static charge and 2) external electrostatic fields.

Several commercially available wrappings or bags are available which offer protection against internal and external static charges. Additionally, some manufacturers use anti-static tubular magazines which offer both physical protection and field force protection. Generally, the bags and magazines are fabricated from non-static generating plastics and are surface treated with conductive materials.

Protection from ultraviolet light must be provided for certain types of memories. Erasable and Programmable Read-Only Memories (EPROM's) are designed to be unprogrammed or erased by exposure to ultraviolet light. A clear window is provided in the case of the chip which exposes the semiconductor to the light. Typically, a covering of some type is applied to the window when it is desired to retain the contents of an EPROM indefinitely.

The presence of fast moving charged particles in semiconductors is the cause of many soft failures in new devices. Sources of this radiation are both internal and external to the device. Alpha particles are generated by the decaying atoms of uranium, thorium and other nuclei commonly found in many semiconductors. Another source of ionizing radiation is cosmic rays found in the earth's atmosphere. Although the effects of these forms of radiation may be reduced by shielding, they cannot be eliminated. As memories become higher in density soft errors will become a bigger problem. This will probably force the use of error correcting codes in future systems.

Possibly the most commonly used method of storing and transporting computer software is through the use of magnetic media. This method is preferred over the other kinds because of its non-volatility, ease of storage, and reliability. Magnetic media consists mainly of tapes, disks (hard platter and Winchester), and flexible floppy diskettes. Since this form of media is used so often, methods have been developed to protect software stored on them. This protection is both external (physical protection) and internal (error prevention, detection and correction).

Information is stored in terms of 1's and 0's on magnetic media in the form of magnetized particles. The physics involved in reading and writing the information is the same for both disks and tapes. Although the information is non-volatile, after extended periods of time the particles become less distinguishable, requiring some form of periodic backup procedure. Both tapes and disks continue to gain popularity in today's market as there is a constant trend to make storage better by increasing flux and bit density, thereby increasing storage capability. As will be discussed later, this in itself is one cause of problems in data storage.

The first magnetic media utilized was the magnetic tape. For larger systems this is an ideal form of storage; however, access time is very slow. The hard disk was developed to increase access time by providing true random access to all data on the device. The disks are very sensitive to dust and particles in the air and require an extremely clean operating and storage environment. This is often hard to accomplish and leads to high expenses. Disks are removable and when care is taken are sometimes used for transporting and storing data.

Cleanliness is extremely important when using hard disks. The disks are placed in containers in a clean room environment, and are packaged in a way which prevents foreign matter from settling on the disks. They should be shipped in containers lined with polyfoam material to provide maximum sealing and insure protection of the disks and the data on them. Recommended shipping and storing conditions are 10° - 85°F to insure data retention. In order to retrieve the data, however, a temperature range of 60° - 135°F must be maintained.

In an attempt to provide an alternative disk system with a less rigid standard for cleanliness, the Winchester type drives were created. Unlike the normal disk, the Winchester allows contact between the head and the disk surface. The disk and drives are contained in their own clean room environment which eliminates the need for special care and cleanliness in handling. Only recently have the Winchester systems been designed to be removable, allowing transportation. Cost, however, restricts serious consideration of Winchesters as mechanisms for transporting software.

Another alternative to the disk is the floppy diskette. These diskettes contain significantly less information than regular hard disks but are considerably more versatile. Diskettes require almost no special handling during use and are ideal for smaller systems. They are very widely used for transporting and storing software.

Although diskettes were designed for versatility, care must be taken with their storage and use. They have built in features to protect data, including special features to prevent particles from becoming lodged between the jacket and disk surface. The envelopes which enclose the diskettes are made of impregnated plastic, which prevents the accumulation of static electric charges which may cause loss of data. There is also a soft liner to provide a fluffiness which helps clean the disks.

In transporting and storing floppy diskettes, care must be taken to prevent damage to the diskette and loss of data. The exposed areas of the diskette should always be covered and the diskette stored in its cover envelope while not being used. Precautions should be taken to protect the diskette from any magnetic sources or sources of extreme heat. Shipping standards range from -40° to 125°F. Although data will not be lost outside these temperatures, sufficient time for cooling (min. 30 minutes) should be allowed to insure data accuracy. In addition, care should be taken not to do pencil erasures around the diskettes, since lead particles tend to cause severe damage to them. When storing or shipping, measures should be taken to prevent the diskette from sagging or leaning as this will tend to warp the diskette, causing loss of data.

As magnetic storage becomes more compact, the need for disk surfaces which can prevent loss of data increases. Attempts have been made to enhance the coercitivity by adding cobalt to the iron oxide coating. Various coatings are used, but none is superior. In addition to modifying the disks, attempts are being made to modify the read-write heads to produce a more reliable recording media.

Magnetic tapes are extremely versatile and have become more transportable due to the cartridge concept. These cassettes, similar to an eight-track, provide an easy method of storing and transporting large quantities of information. The sidewinder tape was created mainly for the purpose of backing up and storing information from Winchester drives, which contain as much as 20 M bytes of data.

It is obvious that with the increase in computer complexity there will continue to be an increase in software volume. Regardless of the amount of care taken during packaging, storage and shipping of software, the risk of errors may increase during this dormant period. Surface damage on magnetic disks is becoming a larger problem with the greater densities. Specks of dust on the reading heads of paper tape readers, card readers, magnetic tape and disk drives cause errors during the reading process. Attempting to read dirty or warped cards may result in erroneous data. To protect against such errors, error detection and correction codes have been implemented in both hardware and software. Alpha radiation in new, high density integrated circuits, as well as noisy telephone lines, lightning storms, and dirty contacts on telephone switching equipment, are additional reasons for including error detection and correction codes during system design. There are numerous error detection and correction methods in use today, a small but representative subset of which is included here. These include parity, longitudinal redundancy check, Hamming codes and cyclic redundancy check.

The simplest and most prominent error detection code used is parity. Parity is used to determine whether the bits in a character have been properly received. One additional bit is appended to each character. In the case of odd parity, that bit is chosen so that the number of 1 bits in the character, including the parity bit, is an odd number. Likewise, in the case of even parity, that bit is chosen so that the number of 1 bits in the character is an even number. If one bit is changed, the number of 1's in the character will have the wrong parity and an error will be detected. Parity on each character can detect only errors that affect an odd number of bits; i.e., characters with one erroneous bit (or three, or five, etc.). Errors that affect an even number of bits will not be detected. Parity does not provide any method of detecting which bit in a character is in error, and therefore does not provide for error correction.

The next level of error detection is known as longitudinal redundancy check. One check character is computed per block of data. Each bit, except the leftmost, in the check character is chosen such that it and the bit in the corresponding position in each character total an odd number of ones. In all characters, including the check character, the leftmost bit is a parity bit as previously described. Longitudinal redundancy check will detect multiple errors within a character as long as multiple errors do not occur in a column. If double bit errors in characters occur simultaneously with double bit errors in columns, then neither parity nor longitudinal redundancy check will detect the errors.

Richard Hamming developed a code for error correction as well as error detection, known as the Hamming code. In a Hamming code, check bits are distributed throughout the character in such a way as to provide parity checks on various bit positions. The bits are numbered starting at 1, not 0, with bit 1 the leftmost bit. All bits whose bit number is a power of 2 are parity bits. Each parity bit checks multiple data bits. Hamming codes may be illustrated using a seven-bit ASCII character. Four parity bits are added such that bits 1, 2, 4, and 8 are parity bits, and bits 3, 5, 6, 7, 9, 10, and 11 are the seven data bits. Each parity bit checks specific bit positions as follows:

Bit 1 checks bits 1, 3, 5, 7, 9, and 11
Bit 2 checks bits 2, 3, 6, 7, 10, and 11
Bit 4 checks bits 4, 5, 6, and 7
Bit 8 checks bits 8, 9, 10, and 11

The parity bit is chosen such that the total number of 1's in the checked bits is even. An error may be corrected by performing parity checks in order and designating a successful check by a 0 and a failure by a 1. The position of the bit in error is determined by the binary number formed by writing the check digits from right to left. The state of that bit may then be inverted to correct the error. Through the use of Hamming codes double-error detection is accomplished with single-error correction. Errors involving more than 2 bits per word may not be detected. Hamming codes increase the memory requirements of a system significantly, adding one parity bit for each power of 2 that the data word increases.

Another method of error detection is the cyclic redundancy check (CRC). CRC requires no parity bit (or bits) per character, as do the previously mentioned codes, but there is a check character at the end of each block of characters. The use of CRC is becoming more widespread with recent integrated circuit designs. The bit stream of length K bits is represented by a polynomial with K terms. The bits in the data stream are the coefficients in the polynomial. This polynomial is sometimes referred to as the message polynomial. To compute the CRC, another polynomial, referred to as the generating polynomial, is chosen of degree greater than zero and less than the message polynomial and with a non-zero coefficient in the X^0 term. Several standard generating polynomials exist, such as CRC-16 and CRC-12. CRC-16 is a 16-bit check character resulting from the generating polynomial $X^{16} + X^{15} + X^2 + 1$ and CRC-12 is a 12-bit check character resulting from the generating polynomial $X^{12} + X^{11} + X^3 + X^2 + X + 1$. The check character is computed by first multiplying the message polynomial by X^r where r is the degree of the generating polynomial. This multiplication will result in zeros in the lower r position of the message polynomial. The check character will be appended to the message in these positions. The product from the multiplication is then divided by the generating polynomial, yielding a quotient and a remainder. The quotient is discarded and the result is added to the product of the previously mentioned multiplication. This remainder is the check character. The check character is stored with the data stream for later verification. A practical application of the CRC procedure is with the use of integrated circuits. A CRC may be computed on the program to be stored in ROM with that CRC also stored in ROM. Any time verification is needed that the program is intact, for example upon power-up, the CRC may be recomputed and compared to the stored CRC. If they are different, then an error has occurred. Cyclic redundancy checks normally do not provide for error correction, but they are an efficient and reliable method of error detection.

It is highly recommended that some method of error detection and/or error correction be implemented in any system design. If the software and/or data within a system are to be stored or shipped, error detection techniques will aid in recognizing failures that occur during shipping and storage.

PACKAGING FOR THE TRANSPORTATION ENVIRONMENT

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Adequate packaging is needed to protect products against the hazards of the transportation environment. The level of packaging protection must be sufficient to bridge the gap between the fragility level of the product, and the highest probable level of hazard in the normal expected transportation environment.

In order to design an adequate level of packaging, it is necessary to define the anticipated environment.

The hazards of the transportation environment are most commonly perceived to be those associated with rail and truck transportation, such as switching shocks and over-the-road vibration. Actually there are many other hazards, such as production line conveyor systems, pallet trucks, clamp trucks, manual handling methods, and warehouse and vehicle stacking forces. Many of these hazards can be controlled and reduced by good packaging and material handling practices. An assessment of the common carrier shipping environment is provided in Forest Products Laboratory's General Technical Report, FPL 22.

The vast majority of non-bulk, non-hazardous products are packaged in corrugated fiberboard containers and shipped via common carrier. Corrugated fiberboard has been a relatively stable, uniform product for many years, with grade classifications based on carrier regulated packaging rules, which tacitly assumed the use of virgin softwood fiber. With increasing demand, and limited availability of softwoods, greater use is being made of excess hardwood and recycled fibers for corrugated boxes. These fibers have different mechanical properties, so corrugated fiberboard boxes procured with the existing specification system may have widely different performance characteristics.

Therefore, it is recommended that performance type tests, such as the ASTM Recommended Practice for Performance Testing of Shipping Containers be used to evaluate packaging to ensure satisfactory and economical protection against the hazards of the shipping environment.

SESSION II

RESEARCH IN THE RAILROAD INDUSTRY

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IMPROVED SUSPENSION DESIGNS AND THEIR INFLUENCE ON FREIGHT CAR DYNAMICS

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Abstract: An evaluation of freight car truck performance was accomplished through the use of extensive field testing under revenue operating conditions. The field test data were reduced and analyzed through the use of digital computers. Results from the analysis were studied, and the performance characteristics of premium design trucks were compared to those of conventional trucks. Incremental performance improvements, relating to freight car dynamics, were studied with the objective of relating them as a cause-and-effect relationship to specific design features in the truck suspensions. This paper discusses the relationship of advanced features in truck suspension design to mitigating effects on the mechanisms influencing wear and failure of vehicle system components.

Keywords: Truck design features; Type I trucks; Type II trucks, premium trucks; performance regimes; lateral stability; ride quality; trackability; freight car dynamics; suspension; vibration.

In the course of a recently completed applied research and evaluation project sponsored by the Federal Railroad Administration, several state-of-the-art design premium freight car trucks were studied and evaluated to determine performance improvements [1]. Conventional, three-piece trucks were also subjected to a parallel evaluation to provide a basis for comparison [2]. Overall freight car truck performance was classified into four distinct performance regimes, inclusive of lateral dynamics, vertical and roll dynamics, load equalization, and curve negotiation. The discussion in this paper considers these performance regimes with the exception of curve negotiation with the intent to relate rail car dynamics to vehicle system and component service life and reliability.

The evaluation of freight car truck performance was accomplished through the use of extensive field testing under revenue service operating conditions. The field test data were reduced and analyzed through the use of digital computers. Results from the analysis were studied, and the performance characteristics of premium design trucks were compared to those of conventional trucks. Furthermore, incremental performance improvements, relating to freight car dynamics, were studied with the objective of relating them as a cause-and-effect relationship to specific design features in the truck suspensions.

The objectives of this paper are (a) to review conventional freight car truck design features, (b) to discuss truck performance and component deterioration, and (c) to present the results on improved performance attainable through improved suspension designs, with an attempt to relate the new features in truck suspension design to mitigating effects on the mechanisms influencing wear and failure of vehicle system components.

CONVENTIONAL (TYPE I) FREIGHT CAR TRUCK

Apart from track, the freight car truck has long been recognized as the most critical component of the rail vehicle system. It supports the carbody and its lading, guides it along the track, transmits braking forces, and isolates the car to some extent from dynamic excitation caused by track irregularities. In performing these functions, the components of the truck suffer various forms of deterioration, such as wear and fatigue failure, [3].

The design of the standard three-piece freight car truck, last improved by the addition of friction snubbing in the 1930's, must be considered as an engineering, manufacturing, and economic achievement. It is relatively inexpensive to produce in large quantities by casting. Its loose construction, i.e., the bolster-side frame and adapter-pedestal connections, make it tolerant of vertical track irregularities (a property generally termed "equalization" which refers to the load distribution over the four wheels when their contact points are not in a plane). The truck's relatively few standardized components can be readily stocked at repair facilities, and maintaining or repairing it does not require a highly skilled labor force.

However, the demands made on the country's rail transportation systems have steadily risen over the past few decades. The gross rail weight of freight cars has shown a continuing increase, which results in higher wheel loads and accelerating deterioration of the track, while higher centers of gravity increase the risk of derailment. In accommodating shippers' requirements, railroads have introduced a number of new freight car configurations which, by virtue of extreme inertial or structural properties, such as low torsional stiffness, or high center of gravity, have given rise to problems of operation and maintenance more severe than were the case for cars of more conventional proportions. In addition, maintenance and repair problems are aggravated by the high utilization rate of unit trains.

In summary, the three-piece, friction snubbed freight car truck, in one sense, has evolved into an optimum configuration with respect to a number of criteria, such as low production cost, high tolerance to rail irregularities (i.e., good load equalization), and moderate wear. However, it must be clearly recognized that some of the performance criteria are met only under the constraint of relatively low speed and that some (like curving performance) are met much less than others. Increasing speeds have brought out a number of problems such as hunting and poor ride quality at high speeds.

TRUCK PERFORMANCE AND DETERIORATION

Figure 1 shows the frames of reference used to deal with the freight car truck. The diagram is divided into two main domains, performance and deterioration, with cost indicated as a factor in both. The interface between these two domains is formed by the truck components.

Performance may be defined either in operational or in functional terms. Operational descriptions include a combination of problem areas, phenomena, regimes, and conditions. Some of these are indicated in Table 1. For example, hunting may be considered a phenomenon, a dynamic regime, and a problem area. Other major performance regimes listed are curve negotiation, harmonic roll, ride quality, derailment, load equalization, braking response, and possibly others. Under each major heading are listed some of the characteristics associated with the particular regime. Such a classification of truck performance features is useful for demonstrating the range and variety of requirements, and there is probably little doubt as to the characteristics of the ideal truck in any of these performance regimes. For example, all railroads would like a high critical speed and low sensitivity to wheel profiles with respect to hunting, and also a minimum flange force during curve negotiation. Unfortunately, the requirements for high performance in several regimes are frequently incompatible.

To reflect actual conditions, a realistic engineering analysis cannot be based on the assumption that all of the components remain unchanged over a period of time and perform their functions according to the equations describing them when new. Almost all components of the freight car truck are subject to various forms of deterioration. The deterioration in performance may be high in proportion to the amount of metal removed or damaged. For example, a small change in wheel tread contour may greatly lower the critical speed as may a widening of the clearances between the bearing adapter and its seat in the pedestal.

Deterioration is accepted as a reality inherent in freight car operation and is allowed for in maintenance schedules and repair facilities, as well as in replacement budgets. Frequencies and rates of failure above the expected are cause for concern, especially when they occur in a particular type or class of components.

The lower half of the freight car truck universe diagram (Figure 1) shows modes or evidence of deterioration on the left, and mechanisms or causes of deterioration on the right. Some of the major modes of deterioration are:

- abrasion
- plastic deformation
- gross loss of material
- change in properties
- metal flow
- surface/subsurface cracks
- brittle fracture
- dislocation

The more important mechanisms or causes are shown in Table 2.

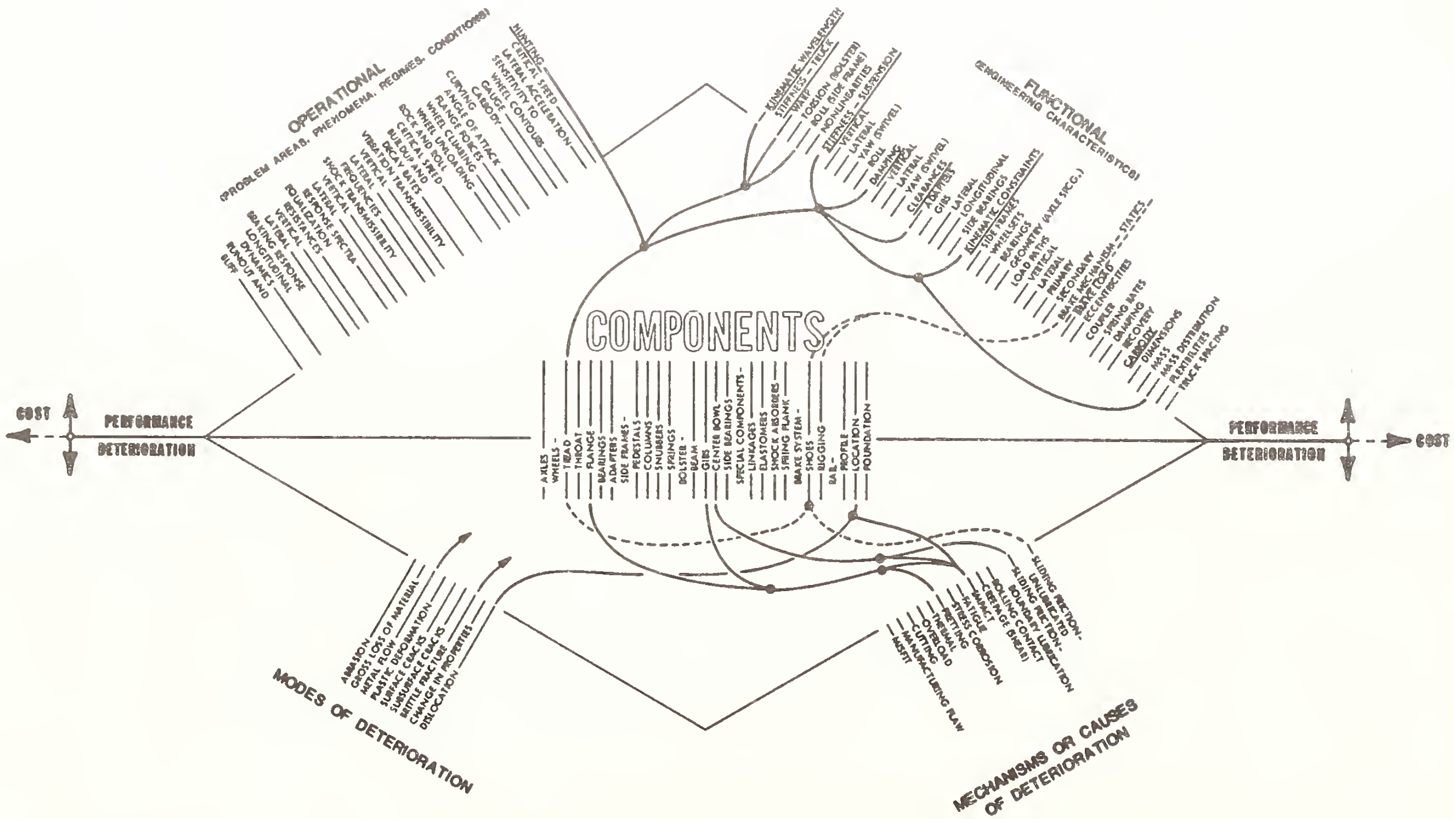

Figure 1. Freight Car Truck Performance and Deterioration

Table 1. Truck Operational Characteristics

<u>HUNTING</u> Critical Speed Lateral Acceleration Sensitivity To - Wheel & Rail Contour - Gauge - Carbody Properties	<u>RIDE QUALITY</u> Shock Vibration Isolation/Transmission Damage Potential
<u>CURVE NEGOTIATION</u> Axle Alignment Lateral Wheel Loads Wheel Climbing Wheel Unloading Resistances	<u>DERAILMENT</u> Wheel Climbing Loss of Guidance (Wheel Lift) - Pre-Existing - Forced
<u>HARMONIC ROLL</u> Critical Speed Maximum Amplitude Centerplate Liftoff Buildup and Decay Rates Wheel Lift Snubbing	<u>LOAD EQUALIZATION</u> Resistances <u>BRAKING RESPONSE</u> Crabbing

Table 2. Mechanisms or Causes of Truck Deterioration

Overload/Overstress
Fatigue
Impact
Thermal Stress
Sliding Friction - Unlubricated
Sliding Friction - Boundary Lubricated
Contact Stress
- Shear
- Fretting
- Stress Corrosion
Cutting
Manufacturing Flaw
Misfit

The identification of failure modes and mechanisms in more precise technical terms may well facilitate the solution of such problems, again by relating them to the existing technology.

Although deterioration shows up in individual components it must be treated as a system problem, just like performance, for the most effective solution. In fact, the aspects of performance and deterioration are connected and interact. This interaction may be illustrated by considering the hunting phenomenon, Figure 1.

Functionally, this is connected with a large number of engineering characteristics, shown on the right, which include the kinematics of both the suspension and the wheel and rail, the stiffness of the truck and suspension, clearances in all the mechanisms, and the characteristics of the carbody. The components of the truck directly involved are the wheel tread and throat as well as some properties of the rail, including profile and gauge, and surface condition.

The deterioration side of the diagram shows some of the possible modes and mechanisms of damage that may be involved, such as metal flow and fracture in wheels, dislocation of the rail due to lateral impact, impact at the gibs and the rim of the center bowl, and center plate wear due to boundary lubricated sliding friction. Another possible interaction begins with the configuration of the brake rigging which influences the transverse location of the brake shoes with respect to the wheel and is believed to affect the development of the worn contour. This in turn can lead to large differences in critical speed, an effect that has been verified in several road tests and in the analyses based on test results.

The interaction between performance and deterioration for the case of hunting is shown in Figure 2. Hunting in freight cars occurs when one of the lateral natural frequencies of the carbody on its suspension comes close to the kinematic frequency of at least one of the two trucks. Therefore, different cars with the same truck, or the same car with different lading, will not necessarily become unstable at the same critical speed. When two identical cars with the same lading have widely different critical speeds, under the same track conditions, this difference in performance can be shown to be due to differences in the contours of the worn wheel treads which in turn are believed to be produced by the characteristics of the brake rigging, and by differences in the brake shoes themselves.

The contributions of these various factors to the development of stable and unstable wheel profiles are not well known, and consideration is being given to the construction of a test rig that will allow a more positive identification of the importance of the various factors involved in the wear process. Thus, the left side of Figure 2 represents a combination of known facts, such as the dynamics of the spring mass system represented by the car, and the importance of the effective conicity of the wheel profile, with conjectures concerning changes in the system due to localized deterioration, that is, the removal of small amounts of metal from the circumference in the wheel.

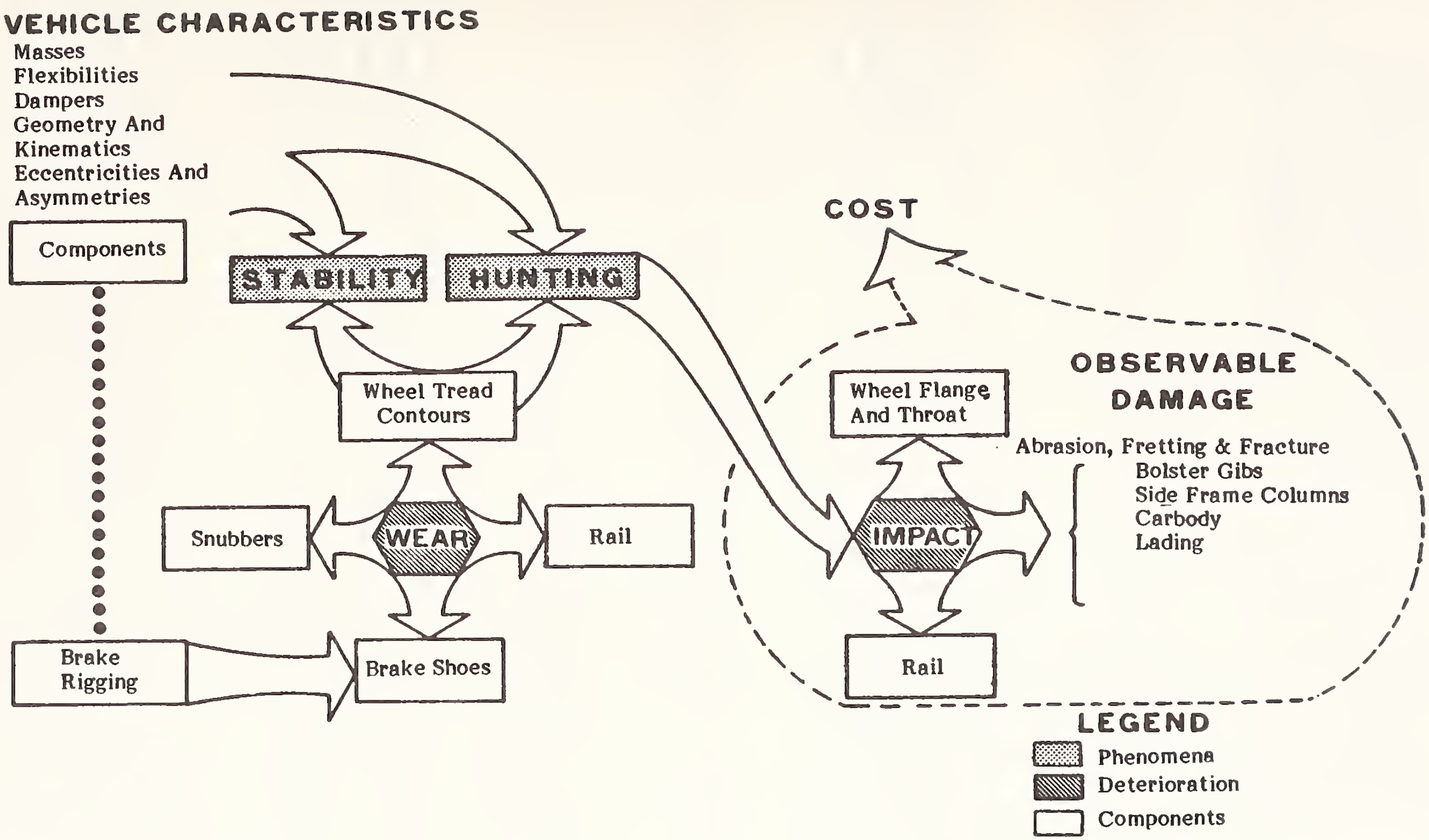

Figure 2. Hunting Environment Simplified

By contrast the right side of the diagram portrays the observable damage and the resulting cost of repair or replacement. Such damage is caused by impact between the wheel and rail, which may result in damage to both, as well as to other truck components through which the impact forces are propagated, such as the bolster gibs, the side frame columns and the center bowl. In severe cases, the car structure may be fatigued and the lading damaged. In addition, if there is relative motion between truck components during impacts caused by hunting there will obviously be abrasion and fretting, even fractures may occur.

Similar relationships between performance and deterioration aspects may be established for other major performnace regimes of the freight car truck. The consideration of these relationships is an essential basis for the establishment of performance guidelines.

TRUCK DESIGN IMPROVEMENTS

The standard three-piece truck is highly serviceable, but by no means perfect. Attempts to improve its performance and durability have a long history, with both railroads and suppliers contributing novel components as well as entirely new design concepts. The acceptance of these improvements is impeded not only by their unavoidably higher initial cost, but by the difficulty of justifying this cost in terms of improved performance.

While many details on truck performance and deterioration remain to be dealt with quantitatively, there exists a basis of knowledge and data sufficient to suggest major improvements in design. These improvements are of two main kinds:

- o Modification of the three-piece truck by addition of special hardware
- o Alternate truck configurations.

Modified Three-Piece Trucks

A number of special components have been developed to improve the roll dynamics of the carbody. Among these are constant contact side bearings, center plate extension pads (C-PEP) and adjustable pneumatic side bearings (air springs). Modifications of the suspension, intended to improve ride quality and dampen harmonic roll, include augmented snubbing, through higher column loads, by additional dampers inserted in the spring nest, or by hydraulic dampers.

Several approaches have been tried to improve lateral stability. Apart from larger center plates, which increase resistance to truck swivel, such modifications include hydraulic dampers between the carbody and truck bolster, or between the truck bolster and side frames, and structural ties between the side frames to reduce parallelogramming. The rigidity or resistance of these devices required to maintain the truck in tram against the moments applied by creep forces appears in general to have been underestimated.

Alternate Truck Designs

Almost without exception, all Type II (premium) trucks are made up of two side frames connected at their centers by a member analogous to the bolster of the Type I (conventional) trucks. What distinguishes alternate truck designs from each other, and from the conventional trucks, are the nature of the side frame/bolster connection, the suspension, and in a few cases, the method of supporting the carbody. These differences exert a significant effect on truck performance, in the four performance regimes of curve negotiation, lateral stability, ride quality, and load equalization.

The goals of the new suspension design premium trucks are to provide higher safe operating speed, greater load-carrying capacity, improved ride quality, reduced damage and deterioration to vehicle components, and reduced wheel and rail wear. To fulfill these goals, each truck possesses certain design features. An evaluation of the different design innovations was accomplished through field testing and test data analysis of both Type I and Type II trucks. In the following sections the performance regimes are discussed, the Type I and Type II performance characteristics are presented, and the incremental performance improvements relating to freight car dynamics are studied with the objective of relating them as a cause-and-effect relationship to specific features in the truck design.

PERFORMANCE REGIMES

Results relating to the three performance regimes of lateral stability, ride quality, and trackability are discussed in this paper. These performance regimes are described below.

It is known that hunting is a safety as well as an operational problem. The large relative motions of components in a hunting truck cause accelerated wear, impact between wheel flange and rail cause gauge widening, and pose the danger of wheel fracture. Severe lading damage can be caused by the lateral vibration of the carbody. The elimination or reduction in severity of hunting is, therefore, economically important because of the damage it causes to both lading and vehicle components.

Ride quality is an important characteristic of a rail vehicle, since it refers to the effectiveness with which it is able to transport lading with minimum damage, both to the lading and to vehicle components. In engineering terms, ride quality is expressed as the response of the carbody to excitations arising from track irregularities. From the operational point of view, ride quality is a function of the transmissibility for shock and vibration which is functionally determined by the spring and damping rates of the suspension, by the load paths, and by the magnitudes of the sprung and unsprung masses. The chief difficulty in optimizing the suspension and improving ride quality is the wide range of loading conditions, between an empty and a fully loaded car. Springs must obviously be designed for maximum combined static and dynamic loads, and dampers for maximum energy dissipation, but these result in a "hard" ride for the empty car. Other constraints include the limited space available in the side frame and manufacturing inaccuracies that modify the performance of even new suspension elements, such as out-of-square springs.

Trackability is the ability of the truck to maintain sufficient loads on all wheels to allow the development of guidance forces which prevent derailment for all extremes of in-service track geometry. The trackability regime discussed in this paper includes, as subsets, the ability of the truck to accommodate: (a) track twist, and (b) curve entry and exit (spirals). This ability is important for successful negotiation of low sidings, poor track in switch yards, and transition from the tangent to curve track for a wide range of operating and environmental conditions.

Typical results are shown in Figures 3 through 7, and in Table 3. The design features of each truck are evaluated and the performance is compared with Type I trucks in the following sections.

TRUCK PERFORMANCE CHARACTERISTICS

A number of Type I and Type II trucks were road tested during Phase I, [4], and Phase II, 5,6 , of the Truck Design Optimization Project. The test data were analyzed in TDOP Phase II to establish quantitative information on Type I and Type II truck performance in the principal regimes of lateral stability, ride quality, steady state curve negotiation, and trackability, [1,2].

Performance test data on the lateral acceleration of the test vehicles were available at various locations on the carbody and trucks. Indications of instability were identified from the time history data and from the power spectra. The power spectra were scanned in the range of 0-5 Hz and the characteristic frequency of hunting was identified. Centered around this frequency, root mean square (rms) accelerations were calculated for a bandwidth of 1 Hz. The calculated rms values for lateral accelerations were then plotted as functions of operating speed. Typical results are shown in Figure 3.

The high speed tangent track test runs were used in the analysis of the ride quality. Performance in this regime is given by the rms response over the wide band spectrum (0-20 Hz) as a performance index. Figures 4 and 5 give the carbody rms vertical and roll accelerations, respectively, as functions of vehicle speed.

Analysis of data in the trackability regime fell under the subregimes of track twist, and curve entry/exit. The wheel unloading index has been quantified for the track twist and curve entry/exit subregimes.

The wheel unloading index is defined as follows:

$$WUI = 1 - W_L/W_H/3$$

where: W_L = vertical force on most lightly loaded wheels

W_H = sum of vertical forces on the three most heavily loaded wheels.

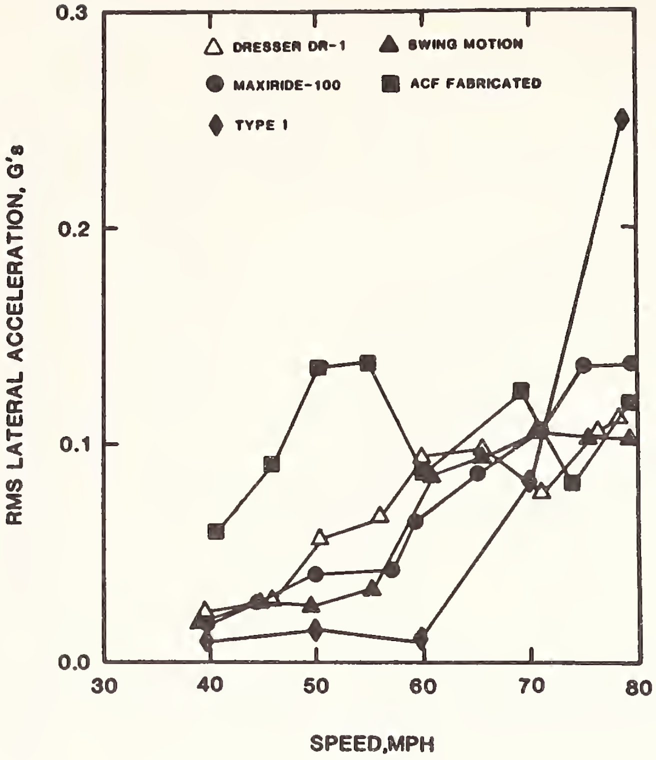

Figure 3. RMS Lateral Acceleration For Empty Cars Versus Speed

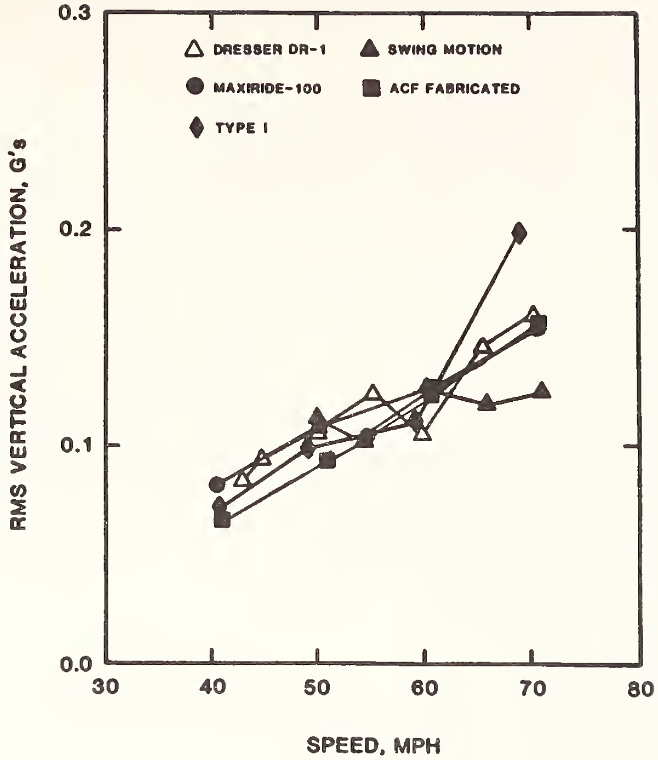

Figure 4. RMS Vertical Acceleration For Loaded Cars Versus Speed

Table 3. Wheel Unloading Index (WUI) Levels (Track Twist)

Truck	Empty Car WUI_{95}	Loaded Car WUI_{95}
Dresser DR-1	0.783	0.281
Swing Motion	0.553	0.368
Maxiride-100	0.297	0.307
Type I	0.351	0.364

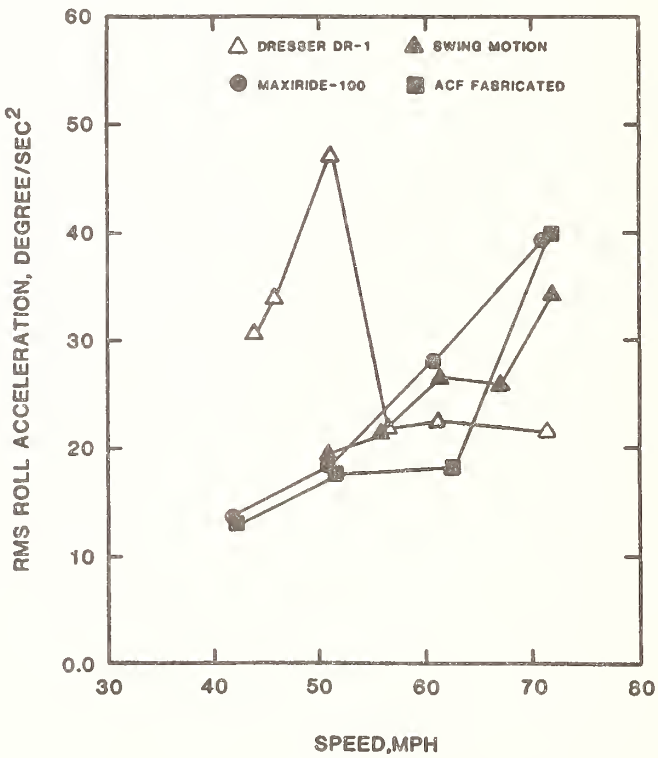

Figure 5. RMS Roll Acceleration For Loaded Cars Versus Speed

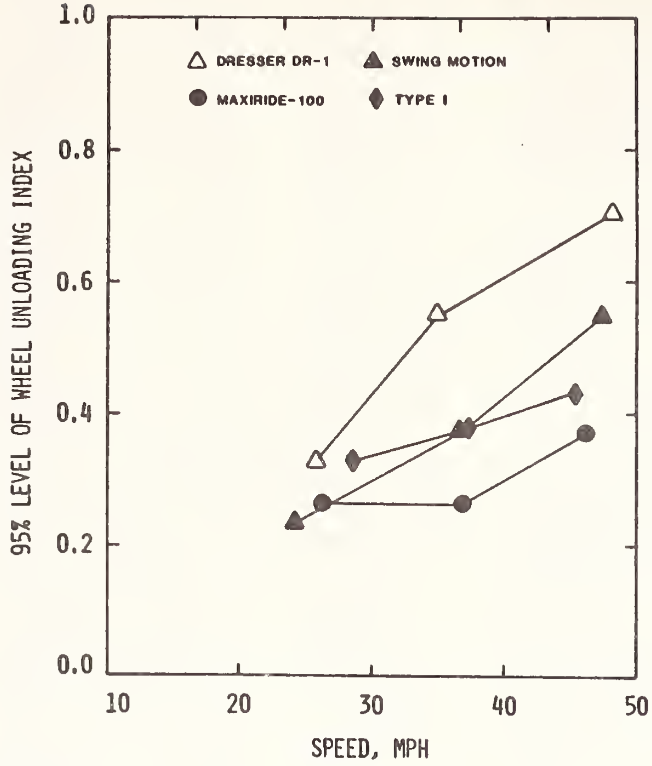

Figure 6. 95% Level of Wheel Unloading Index for Empty Cars Versus Speed - 5.1 Degree Curve

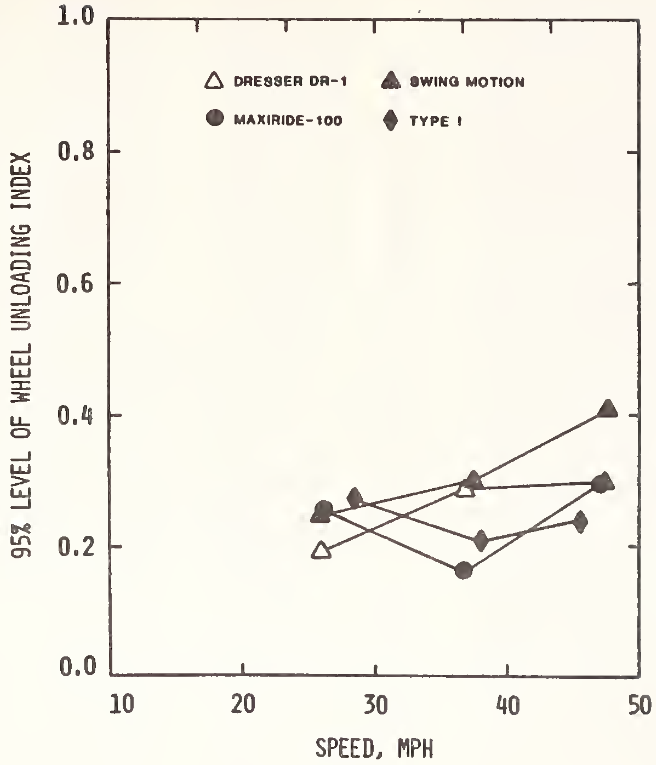

Figure 7. 95% Level of Wheel Unloading Index for Loaded Cars Versus Speed - 5.1 Degree Curve

Data relating to the track twist subregime consisted of yard track tests and those relating to curve entry/exit consisted of the segments of curved zone test runs representing the entry and exit spirals associated with each of the curves in the test track. The quantified levels of the wheel unloading index for the track twist are provided in tabulated format, Table 3. For the curve entry/exit subregime, values of the wheel unloading index with associated probability of occurrence of 95% are plotted as functions of speed, as shown in Figures 6 and 7.

IMPROVED DESIGN FEATURES AND RESULTING PERFORMANCE

As can be seen in Figure 3, which gives the rms lateral accelerations of the carbody as functions of operating speed, while the Type I truck exhibits severe hunting motion in the speed range of 70 to 79 mph, the Type II trucks undergo only low amplitude intermittent hunting in the corresponding speed range, with the lateral motions of the carbody well controlled.

The attenuation or prevention of hunting by using Type II trucks is accomplished through several different techniques. These techniques include decoupling the carbody from the truck (kinematic modification), rigidizing the truck in tram, and energy absorption. Decreasing the lateral suspension stiffness of the truck modifies the vehicle system kinematics and removes the body mode frequencies from the range of kinematic hunting frequencies, thus reducing the synchronization between the carbody and truck motions. Decoupling between the carbody and truck can also be achieved by using hydraulic snubbers. In Type I trucks, the lateral forces on an empty or lightly loaded car may be too low to overcome breakout friction, which may lead to flange impact as the carbody and truck displace laterally as one rigid body. In using hydraulic snubbers to dissipate the energy, however, provision should be made to provide damping in both the vertical and lateral directions. Mounting the hydraulic absorbers vertically only may cause a resonance phenomenon in the medium speed range of 40 to 60 mph with very high lateral motion that may result in damage to vehicle components.

The complete rigidization of a truck (i.e., rigidizing the truck in tram by means of an intertie between the side frames or steering arms capable of reducing parallelogramming action of the truck) is bound to affect truck performance by stabilizing it against hunting, and has a noticeable beneficial effect on the lateral motions developed at high operating speeds.

Increasing truck swivel resistance, either by means of larger center plates supporting the entire load on the side frames or constant contact side bearings, inhibits sustained lateral oscillations. The use of dual-rate springing suspension may also contribute to truck stability.

From the foregoing discussion, it may be said that altering the kinematics, stiffening the truck in tram (i.e., high warp rigidity) and extent of energy dissipitaion in truck swivel can result in attenuating truck hunting and raising the critical speed.

The uniform riding qualities provided by Type II trucks are due to several features implemented in their designs. A reduction in the unsprung mass, for example, greatly improves vertical ride quality. This is achieved by placing the suspension over the axle (primary springing), thus reducing the unsprung mass to that of the wheelsets, bearing, and adapters. The use of dual-rate suspension characteristics that modify the vertical and lateral behavior of the empty car can provide smooth ride in both the lateral and vertical directions in empty and loaded conditions. The ride motion can also be improved by decoupling the vertical motion from the lateral motion, and by reducing the spring rate with the use of longer spring travel combined with the load-sensitive friction snubber wedges that damp the vertical and lateral motions of the bolster.

On the other hand, in general, the Type I truck has better performance in the track twist and curve entry/exit subsets of the trackability regime than many of Type II trucks. Accommodation of vertical track irregularities is an acknowledged advantage with the Type I truck featuring loose construction at the bolster-side frame and adapter-pedestal connections. The ability to maintain adequate vertical load distribution under cross-level irregularities is impaired in Type II trucks. Increasing truck swivel resistance by dissipative means promotes derailment in curves due to inadequate vertical wheel loads maintained under simultaneous lateral forces.

It may be stated that modifications made to improve an aspect of truck performance frequently leads to impairment in another operating regime. Therfore, in an overall judgment of any particular truck, its relative standing in all regimes must be considered, and such judgment may be heavily influenced by each railroad's particular operating conditions and practices.

Within the limitations of the Type I truck, improvements in vertical ride quality appear achievable mainly through modification of the load path. Constant contact side bearings appear to decrease maximum shock loads due to roll. Dual rate springs would improve the ride quality of the empty car, but are apparently difficult to accommodate in the standard truck. Hydraulic shock absorbers would improve the vertical ride; however, provision should be made to provide some damping in the lateral direction.

Some premium trucks incorporate improved load paths. Supporting all of the load on the side frames provides greater lateral stability which benefits the lateral ride quality. The lateral ride quality of some Type II trucks is achieved by kinematically softening the lateral spring rate.

IMPLICATIONS FOR MITIGATION OF FAILURE MECHANISMS

An essential requirement of the freight car truck is that it must perform in different environment (transitions from tangent-to-curved track, from continuously welded to jointed rail, crossings, turnouts and bridges, and changes in the total gross rail weight) in a state of increasing deterioration, which is brought about by its interaction with the operating environment. In the standard three-piece truck, the only precision components are the roller bearings, the journals of the axles on which they are mounted, and the surfaces of the adapters that mate them with the truck. All other components, such as the center bowl, friction snubbers, gibs, and the adapters themselves, transmit loads by metal-to-metal contacts through surfaces capable of moving relative to each other (sliding) or normal to each other, through clearances. The resulting deterioration of these highly loaded connecting points or interfaces represents a large cost to the industry because repairs or replacements must be made to ensure safe operation and prevent delays. In addition, the changes in clearances as the connecting surfaces wear or deform changes the kinematics of the truck as a mechanism.

These changes have a profound effect on the dynamic response of the truck and the railcar body to the conditions of the track described above. For example, wear of the friction snubber shoe decreases the resistance to harmonic roll and thus results in increased probability of derailment when the vehicle is crossing staggered rail joints at certain critical speeds. Snubber wear also decreases the yaw stiffness of the bolster-side frame connection; the resulting increase in the degree of parallelogramming increases the angle of attack of the leading wheelset during curve negotiation and thus leads to increased wear of both wheels and rails. Comparable changes in truck performance can be associated with almost every other group of components.

Wheels, brakes, and rail interact to produce complex wear patterns that affect performance long before components are changed or repaired to ensure safety of operation. As with the other examples given above, the cost of impaired effectiveness due to deterioration is not easy to establish since the mechanism of damage cannot be viewed in action and may be inferred by analysis of such damage.

Improving the truck suspension systems through the use of the different design features, will reduce or attenuate the severity of hunting and provide smooth ride quality. This will result in less wear to wheel, rail, bearing adapters, gib clearance, etc., less impact between wheel flange and rail, thus eliminating gauge widening, reducing the mechanical damage to vehicle components, and mitigating the deterioration not only to the various components making up the rail vehicle, but also the track.

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DEVELOPMENT OF SMALL SCALE IMPACT SIMULATION TECHNIQUES

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ABSTRACT

In derailments and switchyard operations, tank cars carrying hazardous materials frequently experience impacts that result in impairment of the structural integrity of the tank car. Because of the large combination of tank car configurations and credible impact scenarios, it is not feasible to conduct full scale testing to verify that a particular tank car design will perform satisfactorily in impact situations. Therefore, there is a need for reliable small scale testing procedures to simulate the response of tank cars in impact situations.

Initial evaluation of tank car structures under impact was done utilizing an energy method based on Hertz contact stress energy. Dimensionless parameters were identified in this study to form the basis of scaling. Among the quantities that constitute these parameters were tank car dimensions, loading, impact speed, and others. Finite element analyses of the impact of tank cars were then performed to modify the scaling law obtained.

A METHOD TO DETECT AND MONITOR CRACK IN BRIDGE STRUCTURES

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ABSTRACT

The problem of detecting crack in bridge structures is studied with the use of Random Decrement analysis. Fatigue tests were performed on several 3300 mm steel beams using the program controlled load simulator at the Institute of Metal Construction, EPFL, Switzerland. Inputs to the simulator were obtained from actual measurements from various traffic patterns throughout Switzerland. A special technique utilizing pre-stressed thin wires mounted on the specimen was used to detect the initiation of crack in the specimen. The breakage of the wire automatically stops the simulator. Crack propagation resistance gages were then mounted on the specimen to monitor the growth in the size of the crack as the test is continued.

Accelerometers were attached to the specimen and responses were recorded periodically before and after the initiation of crack. These time history responses were used as inputs in the Random Decrement method to obtain the Random Decrement signatures. The change in shape of the signatures was correlated and compared to the experimentally determined crack initiation and growth to test the ability of the Random Decrement method to detect and monitor an unknown crack from the random data.

RANDOM DECREMENT TECHNIQUE

Random Decrement was developed by H. A. Cole for the measurement of damping and detection of structural deterioration of airplane wings subjected to wind flutter excitation (1,2). Further research has been done using the technique for crack detection and damping determination by Yang and etc. (3-14).

Random Decrement technique is a general method of analysis which is particularly suited to the class of problems in which characteristics are desired of inservice structures to unknown random excitation. The major advantage of this technique is that it requires only measurements of the dynamic response of the structure and not the input excitation causing the response. On bridges such random input forces occur from wind and traffic.

The analysis of a time-history response taken at some location on a structure produces a signature which is dependent on structural properties such as natural frequencies and damping. This signature is sensitive to changes in these structural properties and thus offers a means for detecting changes. The method analyzes the measured output of a system subjected to some ambient random input. After analysis, a signal results which is the free vibration response or signature of the structural system. The ability to obtain unique response signatures for different modes (usually accomplished by filtering the output) enables detection of early damage before overall structural integrity is affected. Figure 1 illustrates the acquisition of Random Decrement.

The Random Decrement cracks show up as small blips in the "hashy" high modal density region of the response spectral density and as they grow, the failure mode frequencies become lower and approach the fundamentals where failure becomes imminent. Flaws must be detected early enough - that is, at high enough frequencies - if corrective action is to be taken. Detection is accomplished by passing a random response signal through a bandpass filter which is set to screen high frequencies. If a failure develope, the signature will be affected dramatically because it will dynamically couple with structural modes within these bandpass frequencies.

EXPERIMENTAL TEST

Fatigue tests were performed on several 3300 mm steel beams with gusset plates using the program controlled load simulator at the Institute of Metal Construction, EPFL, Switzerland (see Figure 2). Details of the test beam and the solder goussets are presented in Figures 3 and 4 respectively. The objective of the testing program was to induce fatigue cracks into welded test specimens and to record the vibration responses at various stages of crack growth.

The desired responses were to be of a random nature with fairly uniform and wide frequency spectrums to which the Random Decrement procedure could be applied. To obtain this type of response, two different methods of randomly exciting the specimens under constant load conditions were tested:

(1) actual measurements from various traffic patterns programmed through the grips by the simulator, see Figure 5;

(2) continuously tapping on the specimen with the metal tip of a hammer, see Figure 6.

A special technique utilizing pre-stressed thin wires mounted on the specimen was used to detect the initiation of crack in the specimen. The breakage of the wire automatically stops the simulator. Crack propagation resistance gages were then mounted on the specimen to monitor the growth in the size of the crack as the test is continued. Two B & K accelerometers were positioned on the specimen as shown in

Figure 3. Responses were recorded periodically before and after the initiation of crack. These time history responses were used as inputs to the Random Decrement method to obtain the Random Decrement signatures. A block diagram of the analysis of the recorded response signal using the Random Decrement technique is given in Figure 7.

TEST RESULTS

Random Decrement signatures were obtained from the various time history responses for two frequency ranges, 400 Hz to 800 Hz, 800 Hz to 2000 Hz, using bandpass filters. A number of Random Decrement signatures were obtained for each stage of crack propagation in order to confirm the repeatability of the Random Decrement signatures. The repeatability of the Random Decrement signatures were excellent. This is very essential for the success of the Random Decrement technique in the detection of flaws and cracks in structures.

The Random Decrement signature for the no crack condition is shown on Figure 8. This signature is to be used as the baseline for comparison purpose. The Random Decrement signatures for each crack condition are shown on Figures 8-12 together with the baseline signature.

Changes in the signatures initiated and became increasingly obvious when the crack length became larger and when multiple cracks initiated. The relative deviation of each signature from the baseline signature is given on each figure and summarized in Table 1. The crack length and its condition together with the number of cycles of fatigue loading are also presented in Table 1. A drawing of the crack positions is given in Figure 13. As observed from Table 1, the change in the signature at the crack length of 6.7 mm is sufficiently large to be able to use as a working device.

CONCLUSIONS

Several features of Random Decrement Analysis should be pointed out:

- A very important feature is that the random excitation f(t) does not affect the shape of the Random Decrement signature and does not need to be known.

- From this it can be concluded that the natural excitation can be used for the analysis as long as it is reasonably random. This has the advantage that any structure can be tested on-line and does not have to be taken out of service for inspection.

- Random Decrement examines the structure in its entire integrity and can therefore detect surface cracks as well as inside defects.

- Random Decrement is applicable for both laboratory and field tests. The simple equipment used for the recordings of the response signal make it feasible in many surroundings.

- The recording of response signal takes usually a very short time and can be performed even by untrained people. The main part of the analysis can be done in the laboratory under constant favorable conditions.
- The Random Decrement signatures obtained from a fatigue crack introduced in a structure is consistent and repeatable. Signatures obtained from each crack depth agreed reasonably well with each other.
- Change in the crack depth as small as 6 mm of the model structure can be detected with the Random Decrement technique.
- The Random Decrement signatures demonstrated a marked change in their patterns when the crack depth reached 6 mm, which is considerably far ahead from the complete failure of the structure.
- It appeared to be feasible to use the Random Decrement technique for early detection of incipient flaws and cracks in structures.

ACKNOWLEDGEMENTS

This work was supported by N.S.F., the Swiss N.S.F. and by the Ecole Polytechnique Federale de Lausanne, Switzerland, as part of a major effort to develop better and more reliable techniques to insure the safety of structures.

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TABLE 1 - SUMMARY OF TEST RESULTS (800 Hz-2000 Hz)

	No. Cycles	Relative Deviations Position 1	Position 2	Crack Condition
1.	50,000	-	-	No Crack
2.	228,000	.005839	.004008	No Crack
3.	247,329	.006300	.009302	A_1 = Detection
4.	309,350	.029260	.030710	A_1 = 6.7 mm, A_2 = Detection, A_3 = 2.5 mm
5.	310,000	.033790	.036020	A_1 = 11.2 mm, A_4 = Detection
6.	367,000	.027220	.027930	A_1 = 18.0 mm

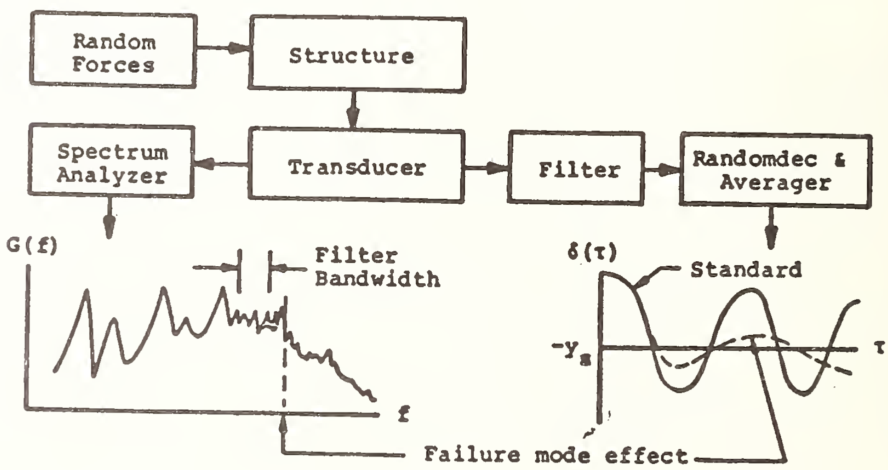

Figure 1 - Acquisition of Random Decrement Signatures

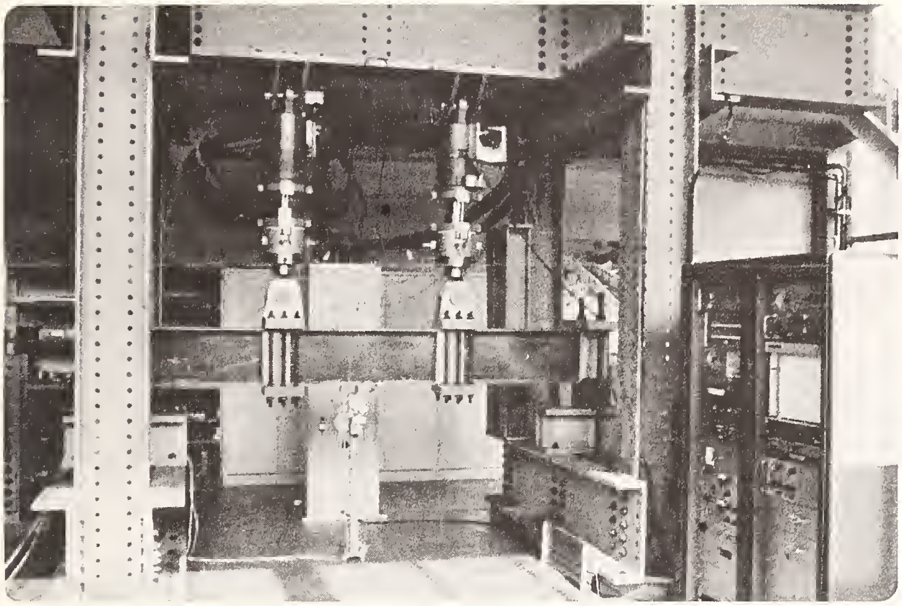

Figure 2 - Program Controlled Load Simulator
(Institute of Metal Construction, EPFL, Switzerland)

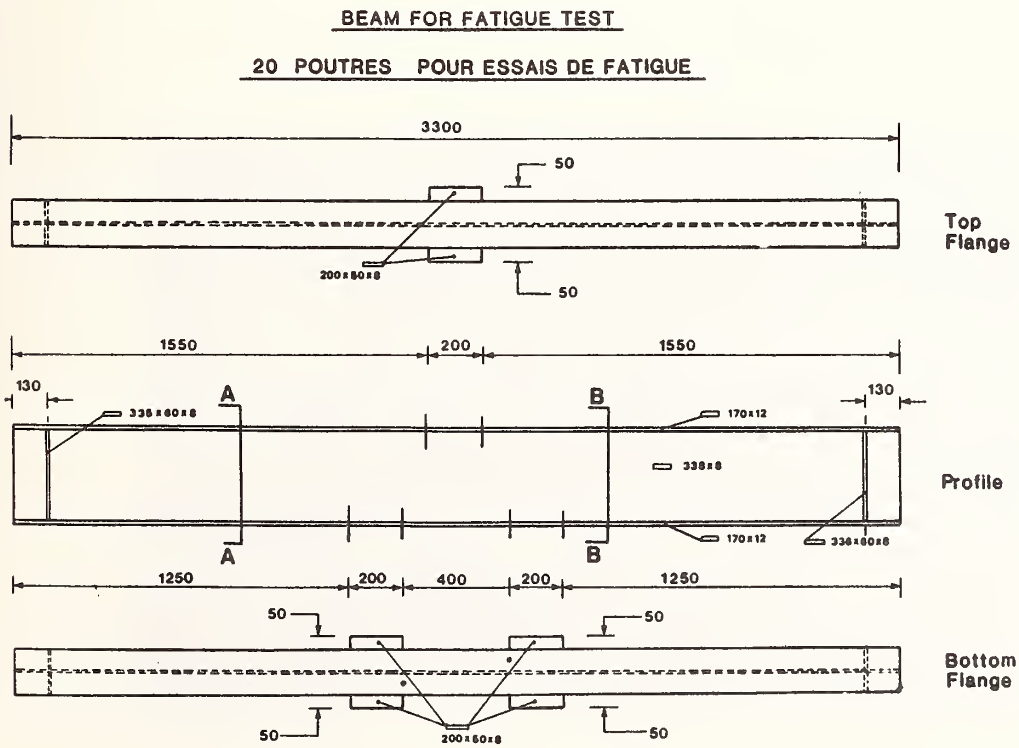

Figure 3 - Details of Fatigue Test Beam

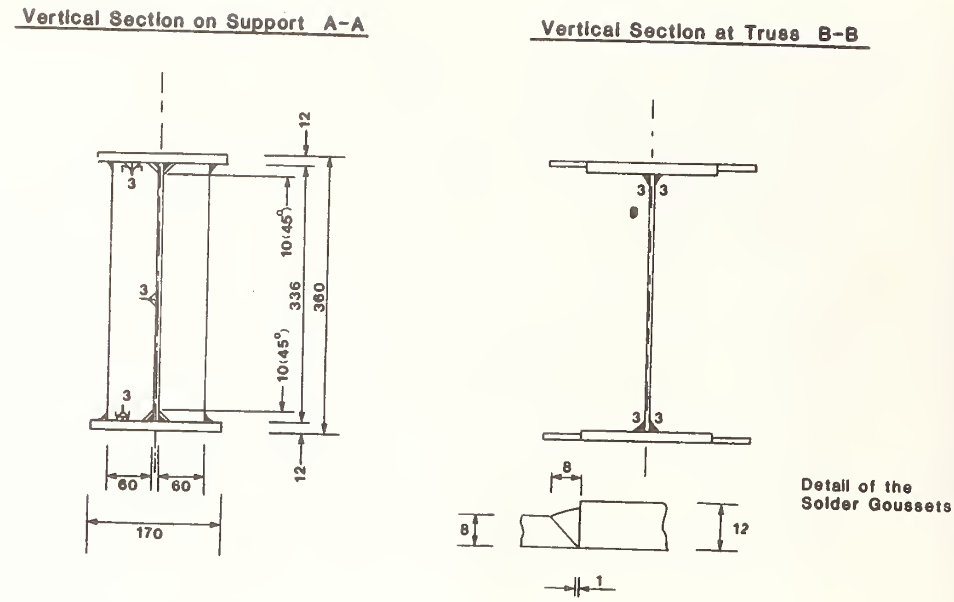

Figure 4 - Details of the Solder Goussetts

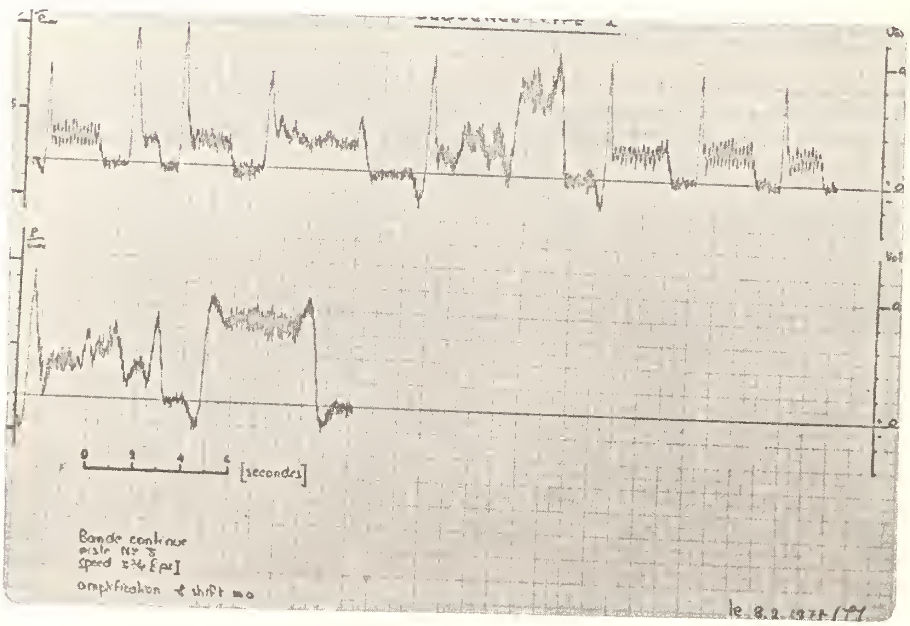

Figure 5 - Programmed Traffic Patterns

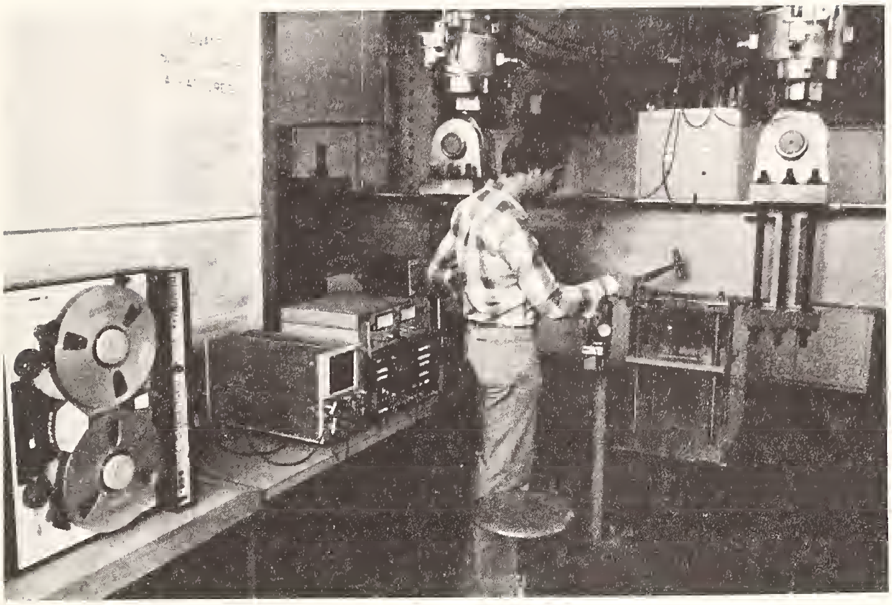

Figure 6 - Hammer Excitation

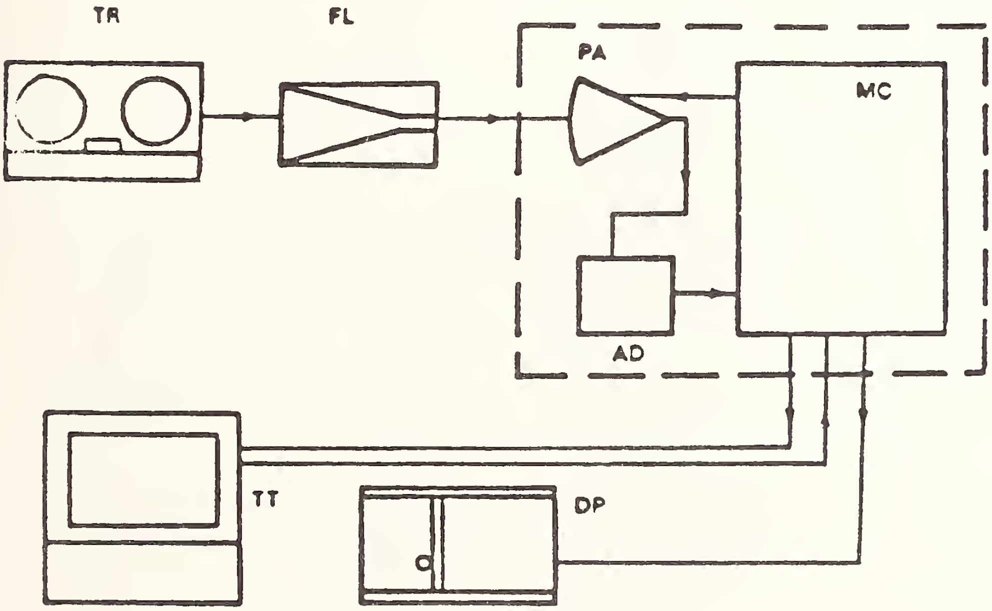

Figure 7 - Random Decrement Analysis Set-Up

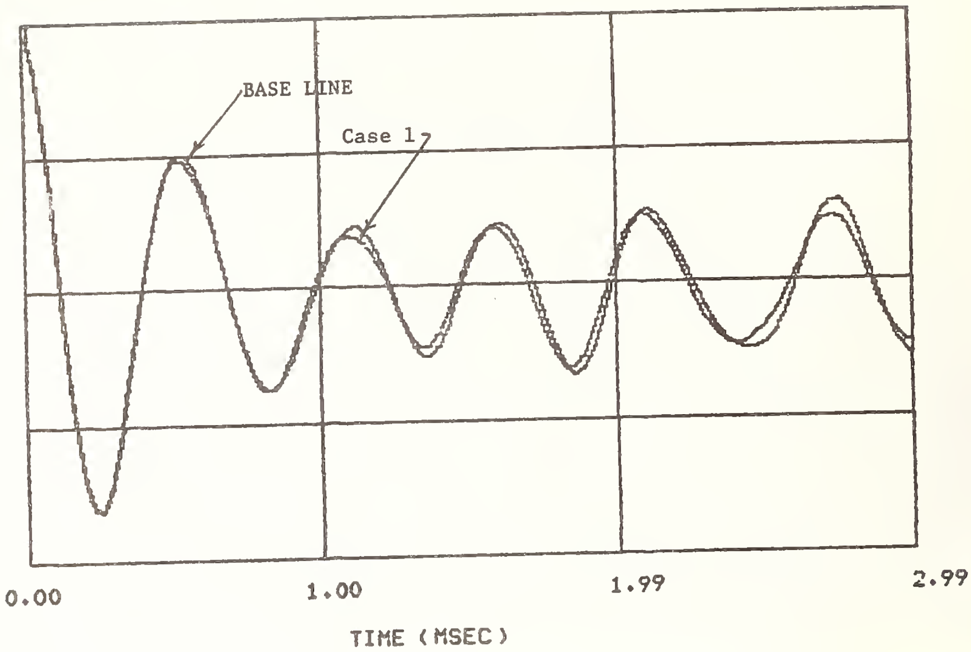

Figure 8 - Random Decrement Signatures (Base Line - Case 1)

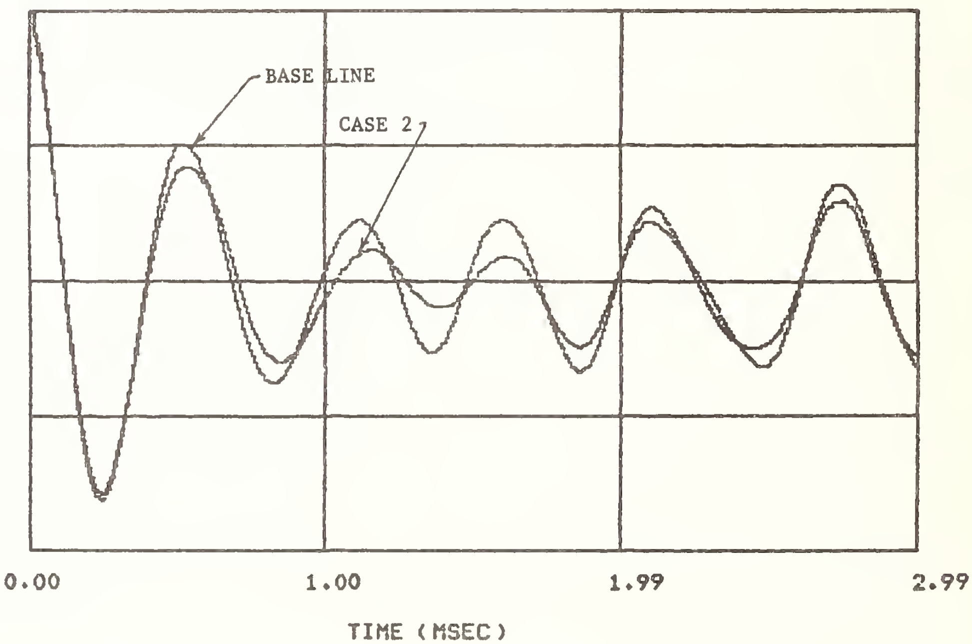

Figure 9 - Random Decrement Signatures (Base Line - Case 2)

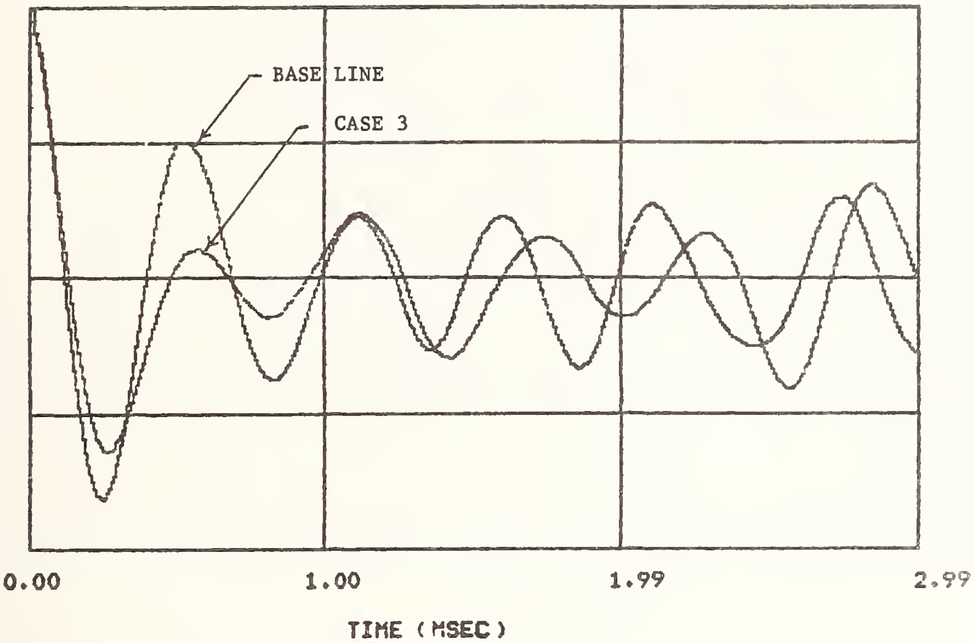

Figure 10 - Random Decrement Signatures (Base Line - Case 3)

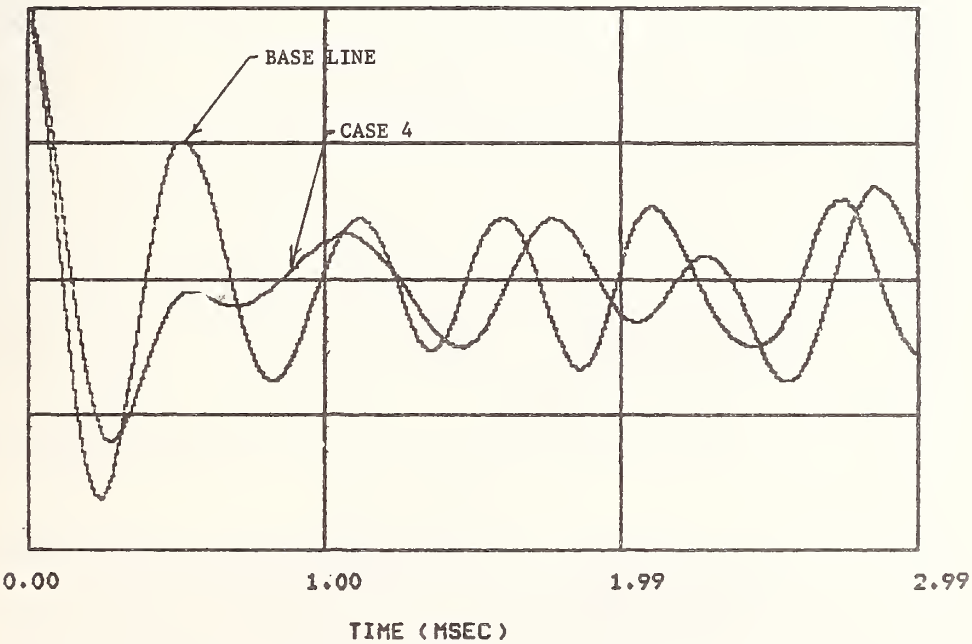

Figure 11 - Random Decrement Signatures (Base Line - Case 4)

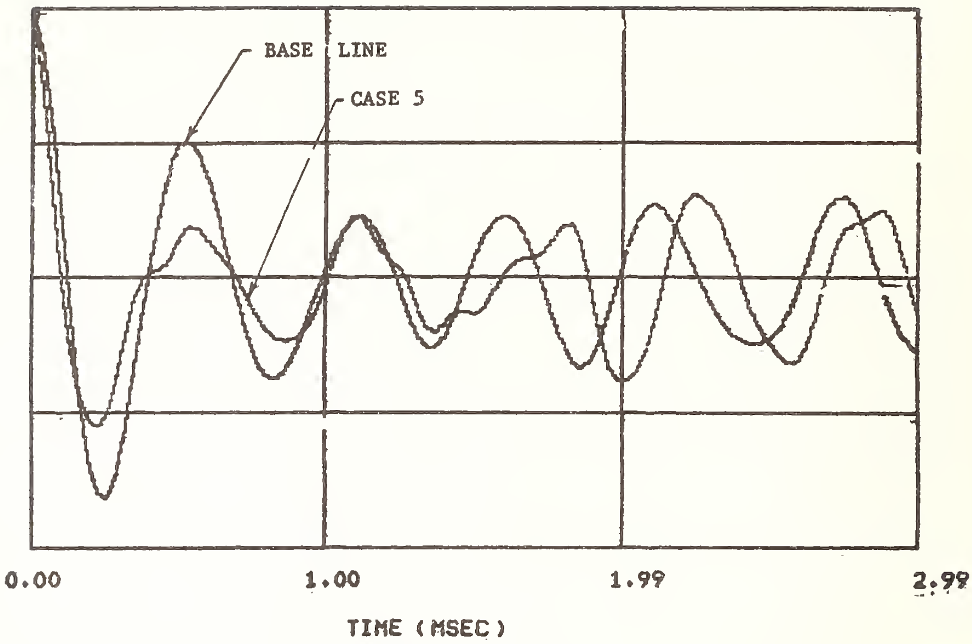

Figure 12 - Random Decrement Signatures (Base Line - Case 5)

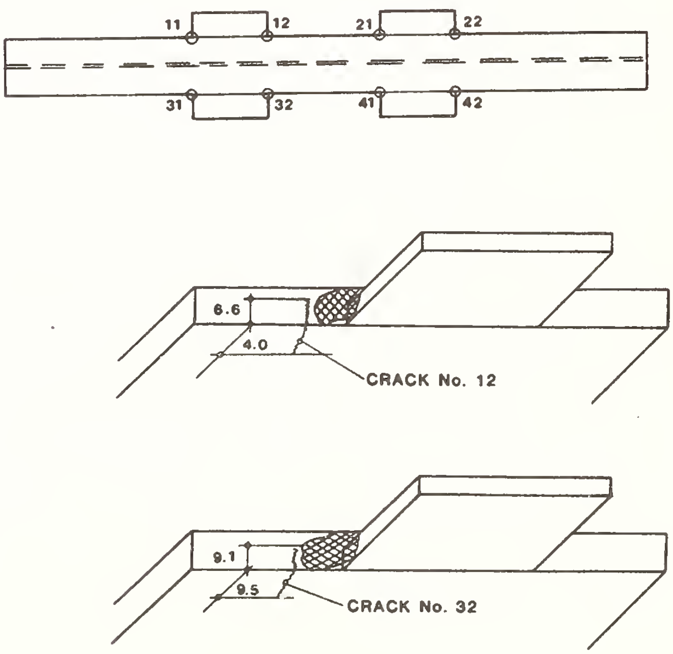

Figure 13 - Number and Crack Positions

RAILROAD LONG-TERM MAINTENANCE-OF-WAY PLANNING TECHNIQUE DEVELOPMENT PROGRAM

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Abstract: This paper describes the cooperative Federal Railroad Administration (FRA) - Conrail research program to develop techniques for long-term track maintenance planning that will better utilize the limited maintenance-of-way (MOW) resources and improve the quality of rail service and overall track safety. This research program makes use of FRA's track geometry cars and other related data (traffic, physical, maintenance) to determine track degradation and develop analytical techniques for planning expenditures for track maintenance.

Key words: Track deterioration; track safety research; track maintenance planning; track geometry; maintenance-of-way; track quality indices.

One of the major operating expenses of today's railroads are those costs related to track maintenance to assure a safe and productive railroad operation. Every railroad must decide where to devote its limited track maintenance budget in the most cost-effective manner without allowing deterioration of track into unsafe or inefficient conditions. As part of a cooperative research program with Conrail, the Federal Railroad Administration (FRA) is developing engineering techniques to help assure that track maintenance resources (greater then $4 billion per year for the industry) are utilized in an optimum manner to improve overall railroad system safety and service. This research will also identify track geometry variations and the allowable relevant vehicle speed limits, in relation to derailments and accidents due to wheel climb, wheel lift, and excessive vehicle motion.

To assist in this research, the FRA is using high speed track geometry measurement vehicles to collect detail track data.

Track Geometry Measurement Vehicle

Track geometry measurement vehicles have a long history in the United States. In 1886, the Chesapeake and Ohio Railroad operated the first domestic track inspection vehicles. Through the years, these track geometry vehicles have become automated and sophisticated in measuring and recording the following basic track geometry parameters;

- Gage - the distance between the two rails in a track structure measured at five-eights of an inch below the top of the railhead.

- Profile - the measurement of the vertical position of each railhead in a track structure.

- Alignment - the measurement of the lateral position of each railhead in a track structure.

- Crosslevel - the difference in elevation between the left and right rails.

- Curvature - the measure of the angular rate of change in track direction.

FRA has been developing track geometry measurement and analysis techniques since 1967. Since that time, FRA began its Automated Track Inspection Program (ATIP), a program in which track geometry vehicles systems (T1-T3, T2-T4, T6 and T10 cars) are used at high speeds to inspect track and monitor compliance with FRA Track Safety Standards.

Between 1971 and 1978, the FRA track geometry vehicle systems were employed in a joint Government-industry development program to investigate the use of the data generated by the track geometry cars in maintenance planning. The data from regular annual or semi-annual track geometry car surveys were supplied to two cooperating railroads (the Denver and Rio Grande Western and the Bessemer and Lake Erie). The railroads utilized the data to supplement their traditional maintenance and planning methods, and reported which track geometry statistics (termed Track Quality Indices or TQI's) they found most useful, and how these data were used. (1) In general, the end product was a system that could produce a detailed report of exceptions to the 1972 FRA track standards and railroad-selected geometry standards, as well as other geometry-based indices. These indices were used by the engineering staff of the participating railroads to direct emergency spot maintenance and plan longer term maintenance.

FRA/Conrail Cooperative Research and Development Program

On the basis of the cooperative research programs findings and successes (described above) in early 1978, FRA initiated a major cooperative research program with Conrail to investigate further the use of track geometry data in track maintenance planning. The objectives of this continuing program are to find quantitative methods of improving railroad safety and the cost-effectiveness of track maintenance through analyses of track geometry data and other necessary data (i.e., traffic, track type, and maintenance data), and prediction of track deterioration. This cooperative study has developed into three phases.

In the Phase I study, completed in June 1980, (2), two test track zones totaling approximately 290 track-miles were chosen as being typical of mixed freight operation. One was in Conrail's Fort Wayne Division, between Van Wert and Bucyrus, Ohio, and the second in the Lehigh Division between Bound Brook, New Jersey, and Penn Haven Junction, Pennsylvania. The Lehigh Division consists mainly of Class 2 and 3 track and annually carries between 7 and 14 million gross tons (MGT). A significant portion of the track is curved and has steep gradients. The Fort Wayne Division consists mainly of Class 3 and 4 tangent, low-gradient track and carries tonnage in the range of 18 to 25 MGT. This test track was observed during 1978 and 1979, the Phase I study period, and traffic and maintenance parameters were uncontrolled.

In 1978 and 1979, physical parameters data were collected in order to determine their contribution to track degradation. Table 1 divides the physical parameters into three areas: track structure, traffic, and maintenance. These parameters were selected on the bases of previous studies, experience, and cost of data acquisition. The structure parameters were quantified for the period of observation; the traffic and maintenance parameters were quantified between the observations.

Traffic parameters consisted of annual tonnage, heavy wheel loads and posted speed. Annual tonnage and heavy wheel loads were quantified for the test zone by a seasonal sampling method. Train speed information was obtained from track charts and timetables. The heavy loads were determined by adding all the carloads above 90 tons and then calculating the percentage of the total annual tonnage.

Track charts were used to determine the rail weight and rail type (bolted or welded). The curvature was quantified using curvature data collected by the FRA T-6 vehicle. Rail profile, ballast, and drainage condition were quantified by using field observations related to specified criteria. The rail profile was defined as the percentage of the surface bent rail in a bolted track segment. The ballast condition was designated as clean, dirty, pumping, and fouled. The drainage condition was designated as adequate or inadequate.

Maintenance practices were divided into two broad categories--basic and discretionary. The basic maintenance operations were regaging, tie renewal, hand smoothing, machine smoothing, and surfacing. Quantifying basic maintenance was accomplished by determining the percentage of the track segment that had received maintenance. The discretionary, or production maintenance, was categorized into three major operations; surfacing, tie and surfacing, and rail renewal.

TABLE 1

PHYSICAL PARAMETERS

Category	Type	Units
Track Structure	Curvature	Degrees
	Rail Weight	Pounds/Yard
	Rail Type	Bolted/Welded
	Ballast Condition	Clean, Dirty, Pumping Fouled
	Rail Profile	Percent Bent
	Drainage Ability	Adequate, Inadequate
Traffic	Track Speed	Miles per Hour
	Cumulative Tonnage	Million Gross Tons
	Heavy Loads	Percent
Maintenance	Basic Maintenance	Levels 0, 10%, 30%, above 30%
	Production Maintenance	Surface, Tie and Surface, Rail Renewal

The test zone was divided into segments that were homogeneous with respect to the above physical parameters. In this case, homogeneous means that a physical parameter, such as tonnage, is constant throughout the segment. The entire test zone was divided into 676 variable length segments, ranging from 0.1 mile to 1 mile.

Track geometry surveys were made in the autumns of 1978 and 1979, the FRA T6 car measured all geometry parameters including alignment at one foot intervals; and intervening traffic, maintenance and track type data were recorded. From the track geometry parameters, figures of merit, referred to as Track Quality Indices (TQI's), were computed. A TQI objectively quantifies one aspect of track condition and is a statistical summary of a track geometry parameter measured over a prescribed length of track. The TQI effectively summarizes the large number of measurements of each parameter for a given track segment. Sample statistics such as mean, ninety-ninth percentile, standard deviation, and higher order moments were investigated in this study.

Figure 1 shows a TQI rating of a track segment in a relative manner. In this example, profile is used; however, all track geometry parameters can likewise be applied. The horizontal dashed line represents perfectly smooth track; the solid line represents the actual measurement; and the shaded regions represent the area between the actual profile and smooth track. A figure of merit for profile roughness can be calculated by taking the square root of the average area. A TQI with a high relative value of 0.5 inch (for example), depicts poor track condition; a TQI with a low relative value of 0.2 inch indicates good track condition. In this example, the profile TQI is the standard deviation in a statistical sense and indicates the profile roughness.

Figure 2 illustrates how a TQI can be used in long-range maintenance planning. The track condition at time t_1 is indicated by TQI_1. If the curve, a TQI track degradation model, is known, the condition of the track at time t_2 (TQI_2) can be projected as a function of physical parameters such as tonnage and rail type.

A total of 14 TQI's were selected as representing the functional requirements of track. These were reduced to five by correlation analysis: one TQI was selected to represent each group of highly correlated TQI's in the original 14. These are listed in Table 2. The five TQI's are the gage roughness index, wide gage index, surface index, line index, and the superelevation index.

Using a linear stepwise regression analysis, track degradation models were developed for each of the five selected TQI's to describe the relationship between TQI's and physical parameters in the following form:

$$TQI_{79} = f\ (TQI_{78},\ C_i\ (\text{independent physical parameters})_i)$$

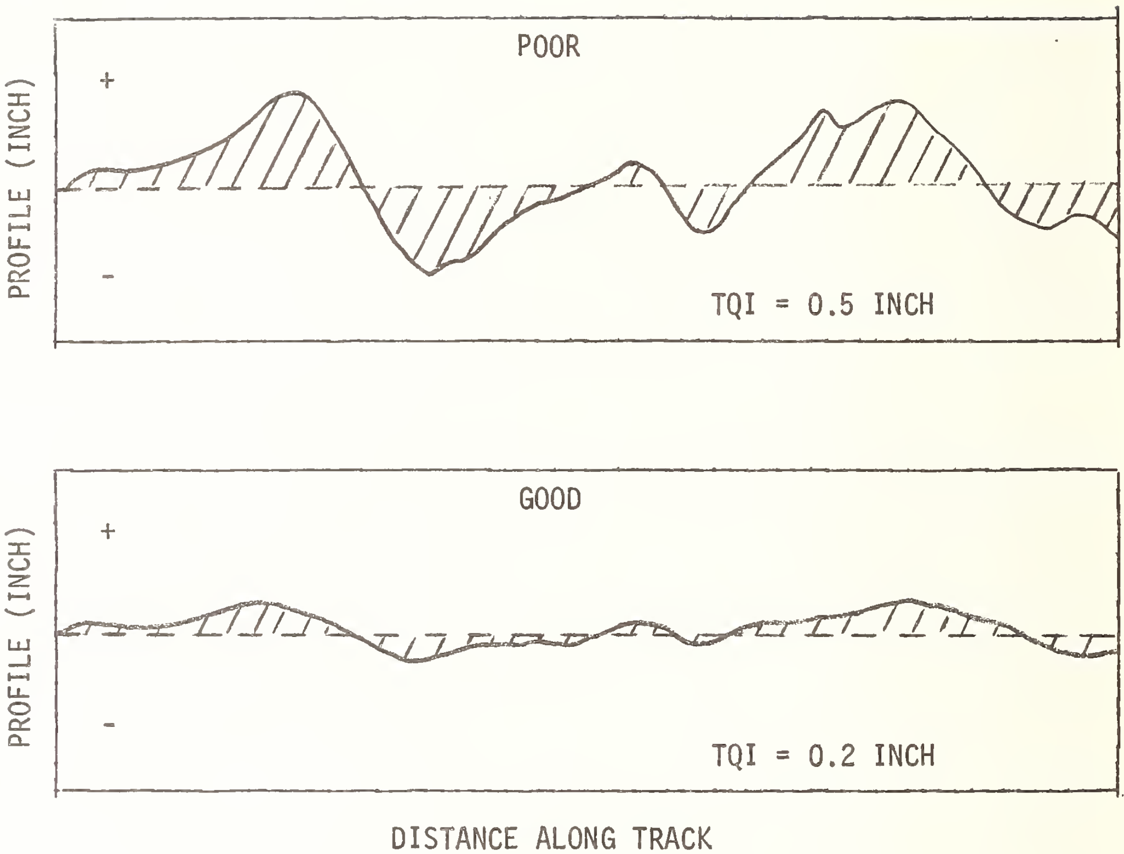

Figure 1. Concept of a Track Quality Index (TQI)

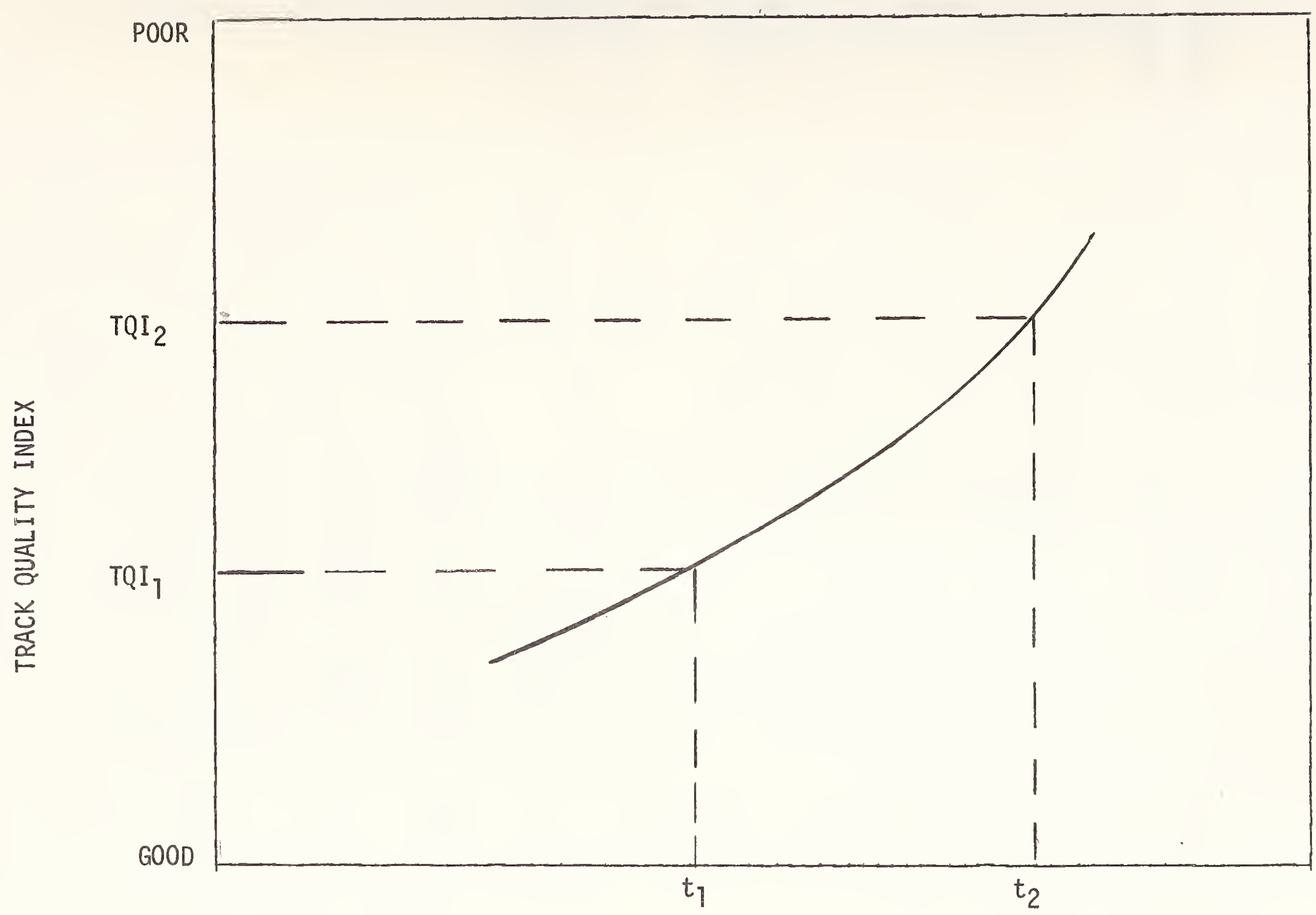

Figure 2. Projection of Track Condition in Terms of a Track Quality Index

TABLE 2

SELECTED TQI's

Variable	Code	Name	Unit	Final Choice
y_1	GAMN	Mean Gage	Inch	
y_2	GASD	Standard Deviation of Gage	Inch	x
y_3	GA99	99-Percentile Gage	Inch	x
y_4	GA3M	Third Moment of Gage	$(\text{Inch})^3/1000$	
y_5	GA4M	Fourth Moment of Gage	$(\text{Inch})^4/1000$	
y_6	XLDV	Standard Deviation of Cross-level	Inch	
y_7	WASD	Standard Deviation of Warp	Inch	x
y_8	WA99	99-Percentile of Warp	Inch	
y_9	PRSD	Standard Deviation of Profile Space Curve	Inch	
y_{10}	PRSM	Standard Deviation of Short MCO of Profile	Inch/1000	
y_{11}	PR99	99-Percentile of Intermediate MCO of Profile	Inch	
y_{12}	ALSD	Standard Deviation of Alinement Space Curve	Inch	
y_{13}	ALSM	Standard Deviation of Short MCO of Alinement	Inch/1000	x
y_{14}	BSEL	RMS Value of Cross-level Deviation from Balanced Superelevation	Inch	x

where C_i are constants associated with each independent variable. Separate sets of equations were derived for each distinct level of maintenance. Table 3 lists the predictive equations for the final five TQI's for unmaintained track. The coefficients of determination (R^2) for all TQI's were well above 0.8. (The previous value of the TQI was the most significant parameter in predicting the future TQI.) This completed the first phase of this study.

Figure 3 illustrates how the prediction equations can be used in track maintenance planning. In this example, the surface condition of seven miles of welded track section is projected, using the surface index TQI prediction equation. The solid circles indicate the measured data points in 1978 and 1979. The open circles indicate the track condition projected by the TQI equation. Note that the measured condition of track in 1979 is lower (better) than the projected condition. This section of the track received discretionary surface operation during 1978 and 1979, and the dashed line represents the effect of the surface operation. The annual tonnage on this section of track was 7 MGT during the study period. At that time the annual tonnage was estimated as 10 MGT for 1980 and 15 MGT for 1981. Track condition was then projected using these tonnage values for 1980 and 1981. If no maintenance is performed, this track section is estimated to be within the standards for Class 3 in 1981.

Fiscal Year 1981 Research

After completion of the Phase I study, the following three activities were undertaken in Phase II of the FRA-Conrail cooperative research program.

(1) The 1978-1979 TQI's and the prediction equations of the Phase I program were analyzed for validation with 1980 data. (3) An FRA T6 track geometry survey was conducted in July and August 1980 over the Lehigh Division test zone, which contained 124 unmaintained track segments. Track geometry data were than processed into five TQI's, and physical parameters were quantified for the 1979-1980 period. Results showed that the equations derived from 1978-1979 data did not predict the 1980 TQI values, except for the surface index TQI. This validation study was based on a somewhat different and smaller test zone than the zone used in the Phase I study. This study used a test zone with less traffic, more curves, and predominantly continuous welded rail. It was recommended that these prediction equations could be improved by adding new physical parameters and extending the study period used to develop such equations.

The study also found that segment length affects the accuracy of the prediction equations. Shorter track segments of less

TABLE 3

PREDICTION EQUATIONS FOR UNMAINTAINED TRACK

TQI	Prediction Equation*	R^2
Gage Roughness	$y_2 = -0.036 + 0.84\hat{y}_2 + 0.0007x_2 + 0.0008x_3$	0.89
Wide Gage Index	$y_3 = 13.88 + 0.75\hat{y}_3 + 0.003x_1 + 0.0116x_4 + 0.0526x_6$	0.85
Surface Index	$y_7 = -0.004 + 0.92\hat{y}_7 + 0.0037x_1 - 0.0032x_4 + 0.069x_6 + 0.0005x_7 + 0.029x_8''$	0.90
Line Index	$y_{13} = 3.84 + 0.71\hat{y}_{13} + 0.429x_1 + 0.349x_4 + 6.57x_6 + 3.18x_8'$	0.83
Superelevation Index	$y_{14} = 0.043 + 0.94\hat{y}_{14} + 0.012x_4 + 0.0009x_7$	0.97

*In this table $\hat{y}_i$ is the previous value of a TQI; x_1 is the tonnage, x_2 is the percent heavy wheels, x_3 is the speed, x_4 is the curvature, x_6 is the rail type (0 for welded, 1 for bolted), x_7 is the percent bent rail, x_8' is ballast level 1 (mildly dirty) and x_8'' is ballast level 2 (pumping).

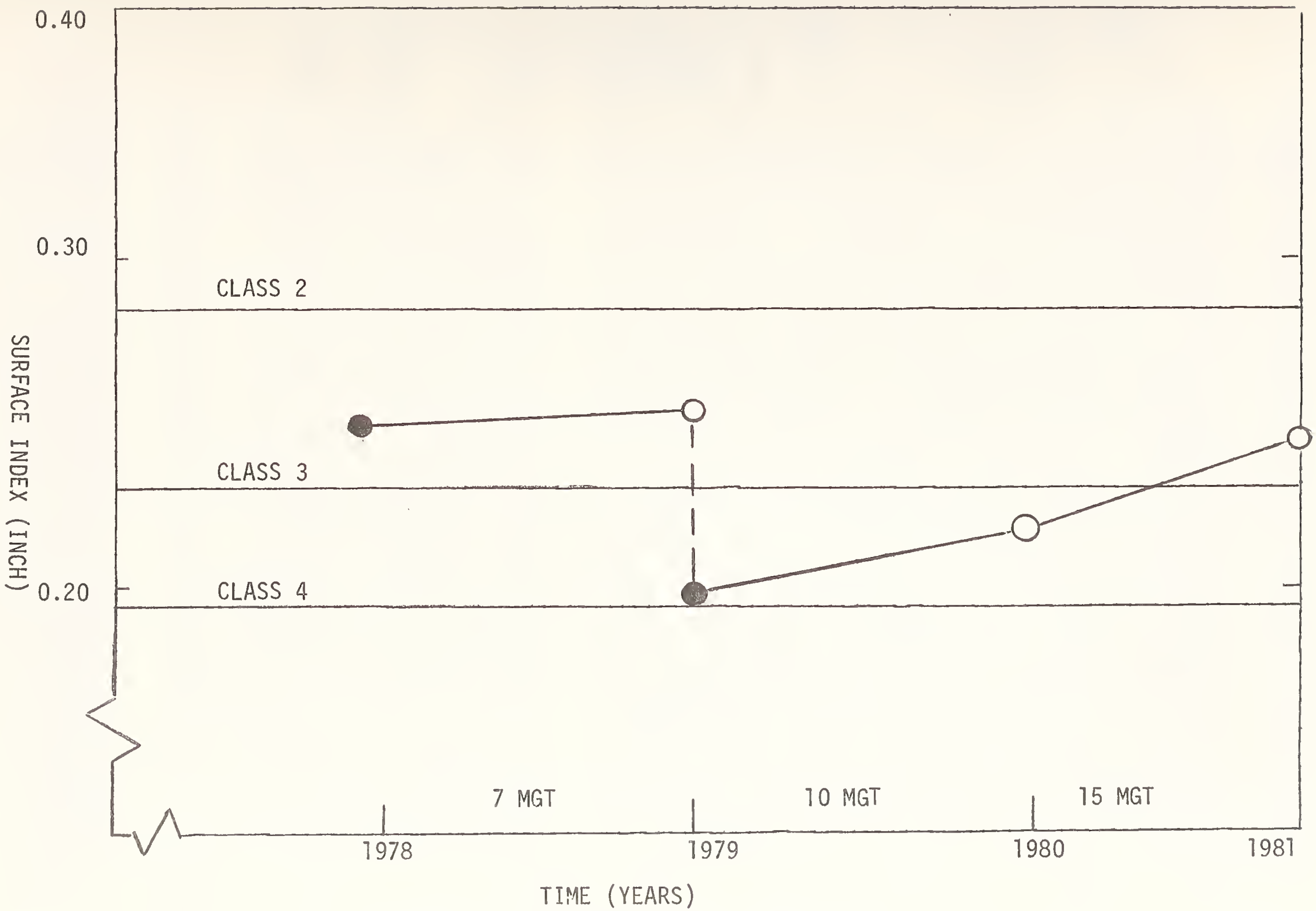

Figure 3. Projecting Track Condition

than one-half mile had more prediction error than longer segments.

(2) A small, independent study was conducted to determine the relationship between track geometry and the critical vehicle responses likely to cause derailment. In the course of their movement along the track, all vehicles respond to geometric irregularities in the track. Some of these irregularities may cause the vehicles to attain motions that are critical--primarily to safe operation, secondarily to lading damage or ride discomfort--and their amplitudes must be reduced through subsequent maintenance. The objective was to identify those geometry perturbations that (by causing critical motions to occur within a vehicle type of configuration, or within the scope of operations allowed on the track) produce an unacceptably high probability of derailment (lading damage or ride discomfort) on the track. Initial findings of this study present methods of processing loaded track geometry measurements by using filters and thresholds that are adjusted to effect a low probability of derailment for the vehicle types, configurations, and operations found on a given track. The performance-based descriptors that are being developed have a uniform threshold with a wavelength response characteristic that varies continuously and linearly with track speed. It is possible, according to the study that unnecessarily long wavelength track maintenance is being undertaken by railroads without affecting geometry uniformity at shorter wavelengths--where derailment mechanisms are at work.

(3) The final, major activity of Phase II encompassed the other two activities and is still continuing. In this study, the approach and methods used in the Phase I and Phase II were reviewed. Extensive evaluations of track deterioration modeling and track maintenance planning were undertaken. Also, there was an attempted engineering analytical corroboration of the statistical results regarding the mechanism of track deterioration developed during Phase I. Ultimately, the Phase II plan is expected to formulate an analytical model of track deterioration that should lead to an optimal track maintenance program which can be practically implemented on an operating railroad.

Preliminary results from the Phase II study activity are summarized below.

(1) The TQI's were suggested based on a space curve of alignment, profile and crosslevel, filtered to minimize the impact of wavelengths below 5 feet and above 50 feet. This is because vehicles respond to track irregularities between these wave-

lengths, and thus the TQI's are directly related to the two major track functional parameters (accident risk and ride quality). The TQI's statistical measures should be the standard deviation and 95th percentile because both cyclic and extreme irregularities influence vehicle behavior and are reasonably represented by standard deviation and 95th percentile.

(2) Segmentation of track for analysis and planning is recommended on half-mileposts and at the beginning and end of curves.

(3) Two approximate engineering formulas give the track permanent response to the vehicle track force: one for vertical deterioration (related to profile and crosslevel TQI's) and one for lateral deterioration (related to gage and alignment TQI's). Sensitivity analyses were performed on these formulas to determine which of the independent causal variables had the most influence on track deterioration for mixed freight traffic. The strong factors for vertical deterioration were previous TQI, track type (jointed or welded), vehicle type, especially the ratio of truck center distance to rail length, speed, ballast type, and condition and drainage condition. Tie type and rail inertia had little influence, mainly because of the consistency of these parameters in the test zone. The sensitivity results were similar for the lateral deterioration because of the coupling of vehicle response to lateral and crosslevel irregularities.

(4) Comparing the results of the engineering analysis with the Phase I TQI deterioration equations was unsuccessful because: Phase I equations were the result of the linear stepwise regression analysis, while the actual engineering relationships are combined multiple parameters; and Phase I TQI equations are too stongly dependent on previous TQI and thereby distort the dependence on the other terms.

(5) In the review of the physical parameters, the recommendations were to delete the surface bent rail as it cannot be visually assessed reliably and to add ballast type, rail age, and time since previous descretionary maintenance.

(6) A standard statistical computer software package should be utilized for calculating TQI's and deriving track deterioration equations.

(7) Regression models should be developed to predict change in TQI as a function of previous TQI and the independent variables (track type, traffic, etc.). Non-linear multiplication type regression models should be investigated. This should improve the ability to predict future track deterioration.

Early in 1982, this study activity is to be completed with the development of a detailed Phase III plan.

Remaining Research and Conclusions

With the collection of track geometry, track structure, maintenance and traffic data, research is continuing at FRA to develop a change in TQI prediction equations utilizing the extensive track maintenance data base started with Conrail in 1978. To help obtain a more significant statistical data base, the Conrail test zone has been expanded to 360 miles; 90 miles of the Amtrak Northeast Corridor (NEC) has also been added to this study. The Amtrak (NEC) data base contains concrete, as well as wood, ties subject to a variety of operating conditions. Resulting FRA research will assist Amtrak in determining the appropriate track surfacing cycle.

In conclusion, this research program expects to accomplish the following.

(1) Develop TQI's from track geometry data that relates to track safety and performance requirements.

(2) Establish degradation prediction equations based on change in TQI's with respect to traffic, track structure, and maintenance data.

(3) Develop an analytical basis for determining appropriate maintenance cycles to insure safe track conditions--thus completing Phase III of the study.

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ANALYTICAL INVESTIGATION OF LADING RESPONSES FROM VARIOUS PULSE SHAPES

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Abstract: The dynamic response of lading and the maximum end-wall forces from variously shaped input pulses in a struck railroad freight car were analyzed. Five different pulse shapes: rectangular, three triangular, and a half sine wave were investigated for cases where the coulomb friction between the lading and the car floor, and the lading looseness, i.e., backlash, were considered. Equal impulses, of magnitude: $I = \int_0^{t_1} F_b(t)dt$, were chosen as the comparison criteria.

The maximum lading response and the maximum end-wall forces from the lading to various pulse shapes were compared using both compressive force and period ratios. The investigation did not intend to find any optimum car cushioning characteristics, but attempted to show the effects of floor friction and lading looseness on the maximum end-wall forces for each input pulse shape, when plotted as a function of lading stiffness. The results showed that rectangular pulse shapes do not always produce the lowest maximum end-wall forces for all conditions of lading stiffness, lading looseness and floor friction.

The results for the non-dimensional, maximum lading forces from rectangular input pulses, as a function of period ratio were shown to compare very well with those obtained using the phase plane delta method. For the other input pulses, the computer solutions were compared with the analytical solutions for zero floor friction and lading looseness.

Key words: End-wall Forces; Impulse Input Wave Forms; Lading Dynamics; Lading Flexibility and Looseness with Floor Friction;

1.0 INTRODUCTION

From the instant a freight car departs from the loading platform until it reaches its destination, it is constantly subjected to many dynamic conditions which arise from railroad environmental disturbances, such as rough starts and stops, and impacts in switching yards. The latter situation creates relatively large shocks in cars and the associated lading, and it is necessary to provide the proper cushioning devices between adjacent cars, so that the lading may arrive at its destination in a relatively damage free condition. When hydraulic devices are used for cushioning impact forces, they can be designed to obtain a wide variety of force/travel characteristics. It is certainly incumbent upon the designer to select a force/travel characteristic that minimizes the maximum end-wall forces, caused by the shifting of the lading. Minimizing the maximum end-wall forces is commonly referred to as the "Min. Max. Problem."

It has been thought that a constant force/travel or force/time characteristic produced the minimum-maximum lading end-wall force. In this Report, it is shown that a constant force/time pulse does not always produce the minimum maximum dynamic response for all conditions of lading flexibility. In fact, certain combinations of delta functions and step functions can produce even lower responses than those shown [6]*; however, they would be extremely difficult to achieve for all combinations of impact velocity, lading flexibility, floor friction, and lading slack or looseness.

* The number in brackets [ ] refer to the references listed in Section 5.0 of this Report.

2.0 PROBLEM STATEMENT

The problem to be analyzed in this paper consists of a striking car of mass, M_S, traveling with an initial velocity, V_o, and impacting a loaded car, initially at rest, through a hydraulic cushioning device. It is the object of this analysis to explore the effects that various impacting pulse shapes have on the maximum lading end-wall forces or maximum dynamic load factors. The dynamic load factors of various pulse shapes, other than the one with a constant characteristic, will be compared. This analysis considers as parameters non-dimensional variables, involving lading flexibility, lading looseness and floor friction. Comparisons of the maximum non-dimensional end-wall forces are made for five selected positive pulse shapes, based on equal values of mean impulsive force over the same time of application.

3.0 ANALYSIS

This analysis is based on the following assumptions:

(1) The lumped parameter system shown in Figure 1 is valid.
(2) The lading flexibility is linear.
(3) Coulomb friction exists between the lading and the car floor.

The schematic presentation of the system under consideration is shown in Figure 1. The meaning of the symbols in Figure 1 may be found by referring to the mathematical nomenclature, listed in Table 1.

3.1 Basic Equations

When a stationary car, M_B, is struck by a moving car, M_S, at an impacting velocity, V_o, through a hydraulic buffer, F_b, the lading, M_L, in the struck car, having a flexibility, k, starts to slide through a distance, a, against coulomb friction, μ, before coming into contact with the end-wall of the struck car. It then begins to compress against the end-wall, thereby producing pressure between the end-wall and itself. This is called the end-wall or lading force, F_V.

Since friction, which in this case is assumed to be coulombic, always opposes the motion, it appears in the equations with a sign opposite to that of the velocity.

Table 1. Mathematical Nomenclature

a	=	Black lash between lading and end-wall
Fb	=	Hydraulic cushion force
Fo	=	Average cushion force
k	=	Lading flexibility
M_B	=	Struck car mass
M_L	=	Lading mass
M_s	=	Striking car mass
M_{BS}, M_{BL}	=	Equivalent masses, as defined by Equation Set (6)
n	=	Force application period - natural period ratio
p	=	$2\pi n$
T	=	Natural period of the system
t	=	Time
t_1	=	Time duration of load application
V_0	=	Impact velocity
x_1, x_2, x_3	=	Displacements of striking car, struck car and lading, respectively
$y1, y2$	=	Relative displacements, defined by Equation Set (6)
z, z_s	=	as defined by Equation Set (10)
β	=	Friction force - average applied force ratio, as defined by Equation Sets (10)
τ	=	Dimensionless time, as defined in Equation Set (10)
τ_s	=	Dimensionless time required for lading to travel a distance of 'a'
$f(\tau)$	=	Normalized cushion force, as defined by Equation Set (10)
μ	=	Friction coefficient between lading and car floor
ξ	=	$z - z_s$ = Maximum end-wall force dynamic load factors
F_v	=	End-wall force

With reference to Figure 1, Newton's laws of motion, and the preceding considerations, the two sets of equations shown as (1) and (2) are obtained. Eqs. (1) are valid when the lading is not in contact with the wall, and Eqs. (2) are applicable when the lading is in contact with the wall [1,3,5].

$$\left.\begin{array}{l} M_s\ddot{x}_1 = -F_b \\[2mm] F_b = M_B\ddot{x}2 + \dfrac{(\dot{x}_2-\dot{x}_3)}{|\dot{x}_2-\dot{x}_3|}\,\mu M_L g \\[4mm] \dfrac{(\dot{x}_2-\dot{x}_3)}{|\dot{x}_2-\dot{x}_3|}\,\mu M_L g = M_L\ddot{x}_3 \end{array}\right\} \quad \begin{array}{l} \text{for } x_2\ x_3 \leq a \\ \text{and } \dot{x}_2 - \dot{x}_3 \neq 0 \end{array} \qquad (1)$$

and:

$$\left.\begin{array}{l} M_s\ddot{x}_1 = -F_b \\[2mm] F_b = M_B\ddot{x}_2 + \dfrac{(\dot{x}_2-\dot{x}_3)}{|\dot{x}_2-\dot{x}_3|}\,\mu M_L g = k\left[x_2 - (x_3+a)\right] \\[4mm] \dfrac{(\dot{x}_2-\dot{x}_3)}{|\dot{x}_2-\dot{x}_3|}\,\mu M_L g + k\left[x_2 - (x_3+a)\right] = M_L\ddot{x}_3 \end{array}\right\} \quad \begin{array}{l} \text{For } x_2 - x_3 \geq a \\ \text{and } \dot{x}_2 - \dot{x}_3 \neq 0 \end{array} \qquad (2)$$

The initial conditions are:

$$\begin{array}{l} x_1(0) = x_2(0) = x_3\ (0) = 0 \\ \dot{x}_1(0) = V_o;\ \dot{x}_2\ (0) = \dot{x}_3\ (0) = 0 \end{array} \qquad (3)$$

Application of the following transformations to equations (1) and (2):

$$\left.\begin{array}{l} y_1 = x_1 - x_2;\ \dfrac{1}{M_{BS}} = \dfrac{1}{M_s} + \dfrac{1}{M_B} \\[4mm] y_2 = x_2 - x_3;\ \dfrac{1}{M_{BL}} = \dfrac{1}{M_B} + \dfrac{1}{M_L} \end{array}\right\} \qquad (4)$$

They become:

$$\left.\begin{array}{l} \ddot{y}_1 = -\dfrac{F_b}{M_{BS}} + \dfrac{\dot{y}2}{|\dot{y}2|}\,\mu g\ \dfrac{M_L}{M_B} \\[4mm] \ddot{y}2 = \dfrac{F_b}{M_B} - \dfrac{\dot{y}_2}{|\dot{y}2|}\,\dfrac{M_L}{M_{BL}}\,\mu g \end{array}\right\} \quad \begin{array}{l} \text{for } y_2 \leq a \\ \text{and } \dot{y}_2 \neq 0 \end{array} \qquad (5)$$

and:

$$\left.\begin{array}{l} \ddot{y}1 - \dfrac{k}{M_B}\,y2 = -\dfrac{F_b}{M_{BS}} + \dfrac{\dot{y}2}{|\dot{y}2|}\,\mu g\ \dfrac{M_L}{M_B} - \dfrac{k}{M_B}\,a \\[4mm] \ddot{y}2 + \dfrac{k}{M_{BL}}\,y2 = \dfrac{F_b}{M_B} - \dfrac{\dot{y}2}{|\dot{y}2|}\,\dfrac{M_L}{M_{BL}}\,\mu g + \dfrac{k}{M_{BL}}\,a \end{array}\right\} \quad \begin{array}{l} \text{for } y_2 \geq a \\ \text{and } \dot{y}2 \neq 0 \end{array} \quad (6)$$

The corresponding initial conditions are:

$$y_1(0) = y_2(0) = \dot{y}_2(0) = 0$$

$$\dot{y}_1(0) = V_o \tag{7}$$

By assuming a cushion force-time relationship, it is evident from the second equation in Eqs. (6) that it becomes uncoupled from the first equation; hence, it can be solved separately. The first equation, however, depends upon the solution of the second equation. In order to facilitate the solutions of the second equations and increase their domain of applicability, it is convenient to non- dimensionalize them by the introduction of the following transformations and definitions:

$$\left.\begin{aligned} \tau &= \frac{t}{t_1};\quad f(\tau) = \frac{F_b(\tau)}{F_o};\quad T = 2\pi\sqrt{\frac{M_{BL}}{k}} \\ z &= \frac{M_B}{M_{BL}}\left[\frac{ky_2}{F_o}\right];\quad \beta = \frac{\mu g M L}{F_o};\quad n = \frac{t_1}{T} \\ z_s &= \frac{M_B}{M_{BL}}\left[\frac{ka}{Fo}\right] \end{aligned}\right\} \tag{8}$$

Application of Eqs. (8) into the second equations in Eqs. (5) and (6), and recognizing that:

$$\left.\begin{aligned} \frac{d}{dt} &= \frac{d\tau}{dt}\,\frac{d}{d\tau} = \frac{1}{t_1}\,\frac{d}{d\tau} \\ &\text{and:} \\ \frac{d^2}{dt^2} &= \frac{1}{t_1^{\,2}}\,\frac{d^2}{d\tau^2} \end{aligned}\right\} \tag{9}$$

yields:

$$\frac{d^2z}{dt^2} = 4\pi^2n^2f(\tau) - \frac{\frac{dz}{d\tau}}{\left|\frac{dz}{d\tau}\right|}\,4\pi^2n^2\beta\,\frac{M_B}{M_{BL}},\qquad \text{for } \frac{dz}{d\tau} \neq 0,\quad z \leq z_s \tag{10}$$

and:

$$\frac{d^2z}{d\tau^2} = -4\pi^2n^2(z-z_s) + 4\pi^2n^2\left[f(\tau) - \frac{\frac{dz}{d\tau}}{\left|\frac{dz}{d\tau}\right|}\,\beta\,\frac{M_B}{M_{BL}}\right]\quad \text{for}\begin{cases}\frac{dz}{d\tau} \neq 0 \\ z \geq z_s\end{cases} \tag{11}$$

The initial conditions become:

$$z(0) = \dot{z}(0) = 0. \tag{12}$$

3.2 Cushion Force-Time Characteristics [1]

In general, the cushion force, F_b, is a function of $\dot{y}_2$ and y_2, which, in turn, are functions of time. Since this functional relationship is determined by the cushioning unit's metering pin design, the approach used herein is to assume various cushion force-time relationships, with the same impulse period of application and mean impulse force, to calculate the maximum end-wall force and then determine the cushion force-travel characteristic.

When impact tests are performed, the measured quantities are cushion force *vs.* time and displacement *vs.* time; hence, force-travel characteristics can be developed from this information. The maximum end-wall forces for five force-time relationships were calculated, and these relationships are shown in Figure 2.

With reference to Figure 2, the equations for the various selected cushion force-time relations are given [1,4] by:

curve 1: $f(\tau) = u(\tau) - u(\tau-1)$ (13)

curve 2: $f(\tau) = -2(\tau-1)\left[u(\tau) - u(\tau-1)\right]$ (14)

or:

$f(\tau) = 2\left[-\tau u(\tau) + u(\tau) + (t-1)\, u(t-1)\right]$

curve 3: $f(\tau) = 4\tau\left[u(\tau) - u(\tau-.5)\right] - 4(\tau-1)\left[u(\tau-.5) - u(\tau-1)\right]$

or: (15)

$f(\tau) - 4\left[\tau u(\tau) - 2(\tau-.5)\, u(\tau-.5) + (\tau-1)\, u(\tau-1)\right]$

curve 4: $f(\tau) = 2\tau\left[u(\tau) - u(\tau-1)\right]$

or: $f(\tau) = 2\left[u(\tau)\tau - (\tau-1)\, u(\tau-1) - u(\tau-1)\right]$ (16)

curve 5: $f(\tau) = \frac{\pi}{2} \sin\pi\tau \left[u(\tau) - u(\tau-1)\right]$

or: $f(\tau) = \frac{\pi}{2}\left[u(\tau) \sin\pi\tau + u(\tau-1) \sin\pi(\tau-1)\right]$ (17)

where:

$u(\tau)$ is the unit step function, defined as:

$u(\tau) = 0$ for $\tau < 0$ (18)

$u(\tau) = 1$ for $\tau \geq 0$

3.3 Solutions and Results

For the five force-time relationships shown in Figure 2, analytical solutions for the non-dimensional end-wall forces were obtained for zero friction and lading looseness, using Laplace transform techniques [1,4]. Details of the analytical procedure are shown in Appendix A. The maximum end-wall force or dynamic end-wall load factors are compared for each loading case, as functions of lading stiffness, in Figure 3. Note that the rectangular input pulse does not always produce the lowest dynamic load factor for all conditions of lading rigidity; however, it appears to be the best compromise for the higher floor friction and the more flexible lading conditions.

In Appendix B the phase plane-delta method [2,3] was applied to the Case 1 forcing function, and a very simple equation was derived for the maximum end-wall force, for all conditions of lading looseness and floor friction. Although this method can be applied discretely to the other loading conditions, a considerable amount of work would be required for each case of lading stiffness, lading looseness, and floor friction; hence a computer program was developed to solve the maximum lading response for all the cushion forces, lading looseness and floor friction cases. The computer program logistics and flow diagrams are presented in Appendix C, and the diagrams showing maximum non-dimensional end-wall forces versus lading stiffness or period ratio are shown in Figures C2 through C31. For each loading case, a family of curves was generated for six values of floor friction and for six values of lading looseness. The computer results, and the analytical and phase plane results were coincident for those cases solved by the latter two methods.

3.4 Discussion of Results

It is apparent from Figures C2 through C31 that the maximum end-wall force increases as backlash or lading looseness increases, and decreases with increasing floor friction. These results are expected, since increased lading looseness permits the lading to strike the end-wall with an initial velocity, whereas increasing the floor friction removes impact energy prior to and during the lading compression at the end-wall.

In general, when impact testing is performed, various forces are measured as functions of time. In particular, cushion and end-wall forces versus time are measured with load cells, but cushion travel and impact velocity are recorded as functions of time using displacement transducers. From these data, impulse - change-in-momentum checks and cushion force - travel characteristics can be developed for each impact velocity and mass condition.

In order to utilize the non-dimensional maximum lading force curves, it is necessary to know the masses of the lading, struck car body and striking car, as well as the lading flexibility and looseness, and the floor friction. The mean impulsive force can be determined from the momentum change for each impact velocity, and the cushion closure can be approximated from the momentum equations for each stroke and impact velocity, assuming an inelastic impact. Depending on the shape of the force-time curve, which depends upon the cushion characteristics, the maximum end-wall force can be estimated from the appropriate curve.

4.0 CONCLUSIONS

The purpose of the study was to explore the effects that the shape of the force-time relationships has upon the maximum lading end-wall force. Based on the results, the following conclusions can be made:

1. Within certain small values of n, i.e., "soft lading," the shapes of the input pulses were of relatively small significance in determining the magnitudes of the maximum lading end-wall forces.
2. Among the five input pulse shapes investigated, the triangular impulse (Curve 2 in Figure 2) always produced the highest maximum end-wall forces, when the same values of n above the minimum n and β were considered.
3. In general, the maximum lading end-wall force increased as the lading looseness increased, and it decreased as the friction increased, for all of the cases considered (See Figures C2 through C31).
4. The trend of the results remained practically the same, except for changes in the magnitude of the maximum end-wall forces, when friction and lading looseness were taken into account.
5. The rectangular pulse shape produced lower end-wall forces, for intermediate values of lading flexibility, and for all values of lading flexibility with large values of floor friction. Cartoned canned goods fall into this range.
6. Pulse shape 3, 4, and 5 in Figure 2 produced lower end-wall forces than the rectangular pulse, for the more rigid ladings at all values of lading looseness and the lower values of floor friction. Tin plate and lumber, etc., would be lading types that fall into this category.

In general, this analysis showed that the constant force-time pulse does not always yield the minimum-maximum lading end-wall force conditions, for all values of lading flexibility, floor friction, and backlash; however, for the most practical overall conditions, the rectangular pulse represents the best compromise.

5.0 REFERENCES

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6.1 Appendix A
Analytical Solutions Using the Laplace Transform Method

The forcing functions, $f(\tau)$, given by Equations (15) through (19), were used to facilitate the application of the Laplace transform method for obtaining the simpler solutions, This method is applied to the special case of each force characteristic curve, when neither floor friction nor lading looseness are considered. 6.1.1 Case No. 1: Rectangular Pulse Input.

This case assumes Curve 1 of Figure 2, with $z_s = \beta = 0$, or $a = \mu = 0$, i.e., no backlash or floor friction. Combining Eqs. (13) and (11) gives:

$$\frac{d^2z}{d\tau^2} + 4\pi^2 n^2 z = 4\pi^2 n^2 \left[u(\tau) \quad -u(\tau-1)\right] \qquad (19)$$

Transforming and rearranging (21), using the initial condition (14), yields:

$$\bar{z}(s) = \left[\frac{1}{s} - \frac{s}{s^2+4\pi^2 n^2}\right](1-e^{-s}) \qquad (a)$$

where: s is the Laplace tranform variable, and the bar over the variable z denotes the Laplace transformation of the variable $z(\tau)$. Inversion of the above equation back to the time domain gives:

$$z(\tau) = (1-\cos 2\pi n\tau)\ u(\tau) - \left[1-\cos 2\pi n(\tau-1)\right] u(\tau-1) \qquad (b)$$

Eq. (b) can be interpreted as

$$z(\tau) = 1-\cos 2\pi n\tau, \text{ for } 0 \leq \tau \leq 1 \qquad (c)$$

and:

$$z(\tau) = 2\sin\pi n \sin\pi n(2\tau-1), \text{ for } \tau \geq 1 \qquad (d)$$

The maximum value of z for $\tau \leq 1$ and $\tau \geq 1$ is given, respectively, by:

$$z_{max} = 2 \text{ for } \tau \leq 1,\ n \geq \frac{1}{2}$$

and:

$$z_{max} = 2\sin\pi n \text{ for } \tau \geq 1,\ n \leq \frac{1}{2} \qquad (20)$$

6.1.2 Case No. 2: Rapid-Rise Traiangular Pulse Input
Friction and Backlash are zero, and the forcing function for this case is given by Eq. (17). The governing equation becomes:

$$\frac{d^2z}{d\tau^2} + 4\pi^2n^2z = 8\pi^2n^2 \left[-tu(\tau) \quad +u(\tau) \quad +(\tau-1)\ u(\tau-1)\right] \qquad (21)$$

whose transform is:

$$\bar{z}(s) = 2\left\{\frac{1}{s} - \frac{1}{s^2} + \frac{1}{s^2+4\pi^2n^2} - \frac{s}{s^2+4\pi^2n^2} + \left[\frac{1}{s^2} - \frac{1}{s^2+4\pi^2n^2}\right] e^{-s}\right\} \text{(a)}$$

Its inversion is:

$$z(\tau) = 2\left\{\left[1-\tau+ \frac{1}{2\pi n} \sin 2\pi n\tau - \cos 2\pi n\tau\right] u(\tau) + \left[(\tau-1) - \frac{1}{2\pi n} \sin 2\pi n(\tau-1)\right] u(\tau-1)\right\} \qquad \text{(b)}$$

which is interpreted as:

$$z(\tau) = 2\left(1-\tau + \frac{1}{2\pi n} \sin 2\pi n\tau \quad -\cos 2\pi n\tau\right), \text{ for } 0 \le \tau \le 1 \qquad \text{(c)}$$

and:

$$z(\tau) = \left[\frac{1}{\pi n} (1-\cos 2\pi n) \sin 2\pi n\tau \quad + \left[\sin 2\pi n-2\pi n\right] \cos 2\pi n\tau\right],$$

for $\tau \ge 1$ (d)

The maximum value for $\tau \le 1$, and $\tau \ge 1$ is given, respectively, as follows:

$$z_{max} = 2\left[2 - \frac{1}{2\pi n}\left\{\text{or}\ \begin{matrix} \sin^{-1} \dfrac{4\pi n}{4\pi^2n^2+1} \\ \cos^{-1} \dfrac{2}{4\pi^2n^2+1} - 1\end{matrix}\right\}\right] \text{ for } \tau \le 1,\ n \ge .371$$

and:

$$z_{max} = \frac{1}{\pi n}\left[2(1-\cos 2\pi n) + 4\pi n(\pi n-\sin 2\pi n)\right]^{\frac{1}{2}}, \text{ for } \tau \ge 1,\ n < .371$$

which is the same as that of CASE 4 expressed in Eq. (26). The maximum values for these cases are shown as functions of n and compared with one another in Figure A1.

6.1.3 Case No. 3: Symmetrical Triangular Pulse Input
Friction and backlash are zero, and the forcing function is given by Eq. (14). The equation to be solved for in this case is:

$$\frac{d^2z}{d\tau^2} + 4\pi^2n^2z = 4\pi^2n^2 \left\{4\left[\tau u(\tau) \quad -2(\tau-.5)\ u(\tau-.5) \quad + \quad (\tau-1)\ u(\tau-1)\right]\right\} \qquad (23)$$

The Laplace transform of Eq. (25), after some rearranging, becomes:

$$\bar{z}(s) = 4\left[\frac{1}{s^2} - \frac{1}{s^2+4\pi^2n^2}\right] (1-2e^{-.5s}+e^{-s}) \qquad \text{(a)}$$

Inversion of Eq. (a) gives:

$$z(\tau) = 4\left\{\left[\tau - \frac{1}{2\pi n} \sin 2\pi n\tau\right] u(\tau) \quad -2\left[\tau-.5 - \frac{1}{2\pi n} \sin 2\pi n(\tau-.5)\right] u(\tau-.5) \quad + \left[\tau-1 - \frac{1}{2\pi n} \sin 2\pi n \quad (\tau-1)\right] u(\tau-1)\right\} \qquad \text{(b)}$$

which is reducible to:

$$z(\tau) = 4\left(\tau - \frac{1}{2\pi n}\sin 2\pi n\tau\right), \text{ for } 0 \leq \tau \leq .5 \qquad (c)$$

$$z(\tau) = 4\left\{1 - \tau - \frac{1}{2\pi n}\left[\sin 2\pi n\tau - 2\sin 2\pi n(\tau - .5)\right]\right\}, \text{ for } .5 \leq \tau \leq 1 \qquad (d)$$

and:

$$z(\tau) = \frac{4}{\pi n}(1 - \cos\pi n) \quad \sin\pi n(2\tau - 1), \text{ for } \tau \geq 1 \qquad (e)$$

The maximum value is:

$$z_{max} = \frac{4}{\pi n}(1 - \cos\pi n), \text{ for } \tau \geq 1 \qquad (24)$$

6.1.4 Case No. 4: Slow-Rise Triangular Pulse Input

Friction and backlash are zero, and the forcing function is given by Eq. (16). The equation to be solved becomes:

$$\frac{d^2z}{d\tau^2} + 4\pi^2n^2z = 8\pi^2n^2\left[\tau u(\tau) - (\tau - 1)\, u(\tau - 1)\right] \qquad (25)$$

It transforms to:

$$\bar{z}(s) = s\left\{\left[\frac{1}{s}2 - \frac{1}{s^2 + 4\pi^2n^2}\right](1 - e^{-s}) - \left[\frac{1}{s} - \frac{s}{s^2 + 4\pi^2n^2}\right]e^{-s}\right\} \qquad (a)$$

Inversion of Eq. (a) back to the time domain gives:

$$z(\tau) = 2\left\{\left[\tau - \frac{1}{2\pi n}\sin 2\pi n\tau\right]u(\tau) - \left[\tau - \frac{1}{2\pi n}\sin 2\pi n(\tau - 1) - \cos 2\pi n(\tau - 1)\right]u(\tau - 1)\right\} \qquad (b)$$

which is interpreted as:

$$z(\tau) = 2\left[\tau - \frac{1}{2\pi n}\sin 2\pi n\tau\right], \qquad \text{for } 0 \leq \tau \leq 1 \qquad (c)$$

and:

$$z(\tau) = 2\left\{\left[\frac{\cos 2\pi n - 1}{2\pi n} + \sin 2\pi n\right]\sin 2\pi n\tau - \left[\frac{\sin 2\pi n}{2\pi n} - \cos 2\pi n\right]\cos 2\pi n\tau\right\} \qquad \text{for } \tau \geq 1 \qquad (d)$$

The maximum value of z always occurs in the residual amplitude, that is:

$$z_{max} = \frac{1}{\pi n}\left[2(1 - \cos 2\pi n) + 4\pi n(\pi n - \sin 2\pi n)\right]^{\frac{1}{2}} \qquad (26)$$

6.1.5 Case No. 5: Half Sine Wave Pulse Input
Friction and backlash are zero, and the forcing function is given by Eq. (15). The equation to be solved for this case becomes:

$$\frac{d^2z}{d\tau^2} + 4\pi^2n^2z = 4\pi^2n^2 \left[\frac{\pi}{2}\right] \left[u(\tau) \quad \sin\pi\tau \quad + \quad u(\tau-1) \quad \sin\pi \quad (\tau-1)\right] \qquad (27)$$

The equation transforms to:

$$\bar{z}(s) = \frac{2\pi^2n^2}{4n^2-1} \left[\frac{1}{s^2+\pi^2} - \frac{1}{s^2+4\pi^2n^2}\right](1+e^{-s}) \qquad (a)$$

Its inversion gives

$$z(\tau) = \frac{2\pi n^2}{4n^2-1} \left\{\left[\sin\pi\tau - \frac{1}{2n} \sin 2\pi n\tau\right] u(\tau) + \left[\sin\pi(\tau-1) \right.\right. \qquad (b)$$

$$\left.\left. - \frac{1}{2n} \sin 2\pi n(\tau-1)\right] u(\tau-1)\right\}$$

Which is interpreted as:

$$z(\tau) = \frac{2\pi n^2}{4n^2-1}(\sin\pi\tau - \frac{1}{2n} \sin 2\pi n\tau), \text{ for } 0 \leq \tau \leq 1 \qquad (c)$$

and:

$$z(\tau) = - \frac{2\pi n}{4n^2-1} \cos\pi n \quad \sin\pi n \quad (2\tau-1), \text{ for } \tau \geq 1 \qquad (d)$$

The maximum value for $\tau \geq 1$ is given by:

$$z_{max} = \frac{2\pi n}{4n^2-1} \cos\pi n$$

and for $0 < \tau \leq 1$, $n > \frac{1}{2}$ and i, an integer: (28)

$$z_{max} = \frac{\pi n}{4n^2-1} \left[2n \sin \frac{2\pi i}{2n+1} - \sin \frac{4\pi n i}{2n+1}\right]$$

where:

$$0 < \frac{2i}{2n-1} \leq 1 \qquad (28a)$$

The maximum-maximum end-wall force for the five selected pulse shapes, when floor friction and backlash were present, were obtained by solving the differential equations of motion, Equations (10), (11) and (12), on the AAR's Dec 2050 computer system. Details of the logic, in the form of flow diagrams, as well as a brief description of the numerical algorithm used to solve the equations, are shown in Appendix C.

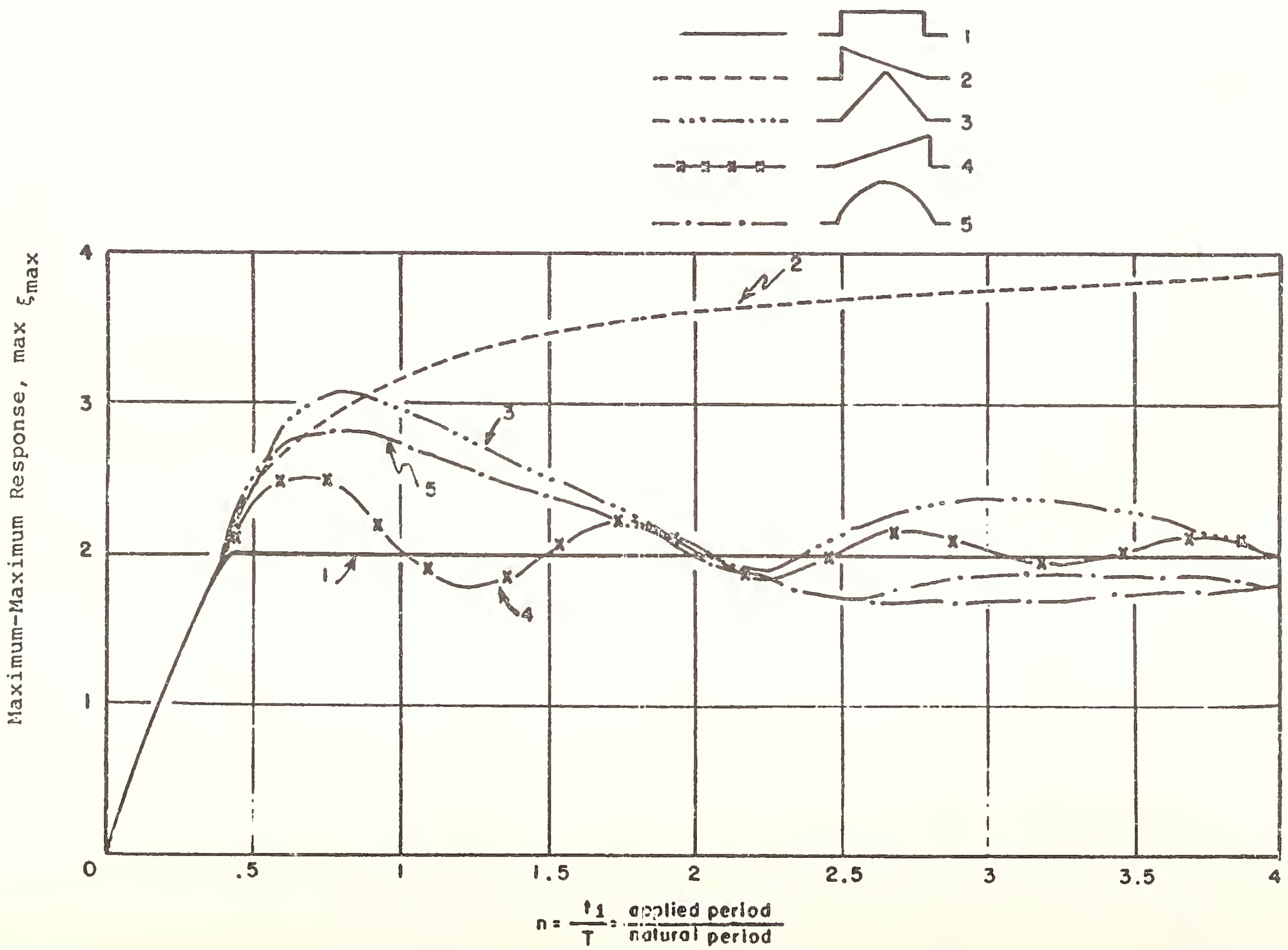


Figure A1. Maximum Maximum Response vs. Period Ratio for the Case of No Lading Friction or Backlash.

6.2 Appendix B

Phase Plane Delta Method for a Rectangular Pulse

The utilization of this technique requires that ξ be plotted versus $\frac{\dot{\xi}}{2\pi n}$. Before using the phase plane method, the free sliding lading equations are solved to produce the initial conditions for this method. Hence:

$$\ddot{\xi} = 4\pi^2 n^2 (1-\beta), \quad \text{for } \xi \leq 0 \tag{1}$$

The initial conditions are:

$$\xi(0) = -z_s \tag{2}$$

$$\dot{\xi}(0) = 0$$

Integrating (1) using (2) gives:

$$\dot{\xi} = 4\pi^2 n^2 (1-\beta)\tau \tag{3}$$

$$\xi = 4\pi^2 n^2 (1-\beta)\frac{\tau^2}{2} - z_s \tag{4}$$

The final condition for (4) requires that $\xi(\tau_s) = 0$, where τ_s is the time that the lading contacts the end-wall. Therefore:

$$4\pi^2 n^2 (1-\beta)\frac{\tau_s^2}{2} - z_s = 0$$

or:

$$\tau_s = \frac{1}{2\pi n}\sqrt{\frac{2z_s}{1-\beta}} \leq 1 \tag{5}$$

The final velocity is obtained by substituting (5) into (3), hence:

$$\frac{\dot{\xi}(\tau_s)}{2\pi n} = \sqrt{2z_s(1-\beta)} \tag{6}$$

$\xi(\tau_s) = 0$ and (6) are the initial conditions for the phase plane method. Figure B1 shows the phase plane scheme, in which

$$\theta_s = \tan^{-1}\sqrt{\frac{2z_s}{1-\beta}} = \tan^{-1} 2\pi n\tau_s$$

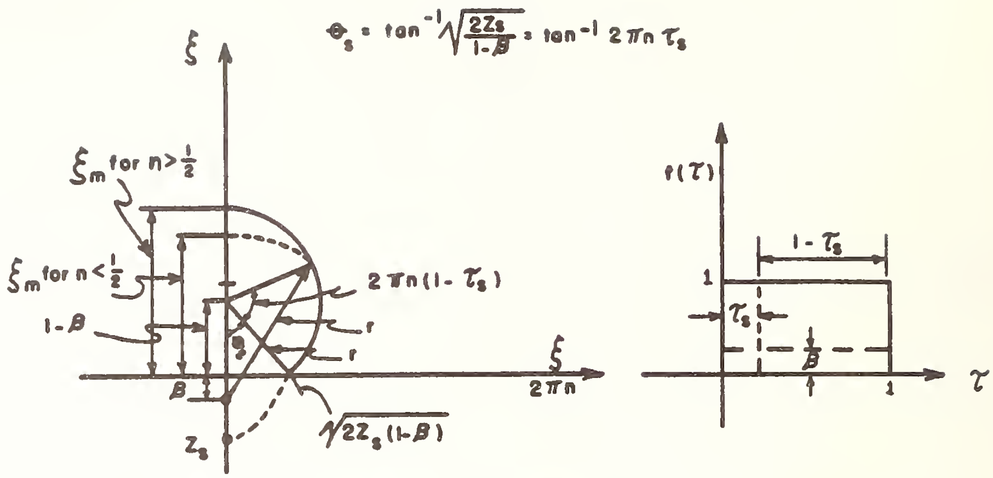


Figure B1. Example of the Phase Plane Delta Method.

The maximum value of ξ for small values of n ($n<\frac{1}{2}$) is equal to $r_2-\beta$. For large values of n, the maximum ξ occurs in the first cycle for $\tau<1$.

For this case:

$$\xi_{max} = r_1+1-\beta \tag{6}$$

But: $$r_1 = \sqrt{(1-\beta)^2+2z_s(1-\beta)} \tag{7}$$

Substituting (7) into (6) gives:

$$\xi_{max} = \sqrt{(1-\beta)^2+2z_s(1-\beta)} \quad +1-\beta,\ n>\tfrac{1}{2} \tag{8}$$

For $n<\frac{1}{2}$, $\xi_{max} = r_2-\beta$, or

$$\xi_{max} = \left\{r_1{}^2+1-2r_1 \cos\left[2\pi n(1-\tau_s)+\theta_s\right]\right\}^{\frac{1}{2}}-\beta$$

$$\xi_{max}= \left\{(1-\beta)(1-\beta+2z_s)+1-2\sqrt{(1-\beta)(1-\beta+2z_s)}\cos\left[2\pi n(1-\tau_s)+\theta_s\right]\right\}^{\frac{1}{2}}-\beta \tag{9}$$

Equation (9) holds for:

$$2\pi n(1-\tau_s) + \tan^{-1}2\pi n\tau_s\leq\pi \tag{10}$$

Table B1 gives values of ξ_{max} for $n>0.5$ and various values of β and Z_s, which were obtained from Equation (8).

Tables B2 through B6 give values of ξ_{max} for $n\leq0.5$, and various values of β and Z_s.

Table B1

ξ_{max} for n >0.5 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	2.000	1.600	1.200	0.800	0.400	0.0
0.2	2.183	1.780	1.375	0.966	0.546	0.0
0.4	2.342	1.930	1.517	1.092	0.647	0.0
0.6	2.483	2.065	1.640	1.200	0.729	0.0
0.8	2.612	2.186	1.750	1.290	0.800	0.0
1.0	2.732	2.300	1.850	1.380	0.863	0.0

Table B2

ξ_{max} for n = 0.1 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	0.618	0.388	0.224	0.116	0.046	0.0
0.2	0.628	0.325	0.088	0.0	0.0	0.0
0.4	0.632	0.256	0.0	0.0	0.0	0.0
0.6	0.654	0.219	0.0	0.0	0.0	0.0
0.8	0.698	0.234	0.0	0.0	0.0	0.0
1.0	0.765	0.301	0.0	0.0	0.0	0.0

Table B3

ξ_{max} for n = 0.2 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	1.176	0.870	0.595	0.355	0.157	0.0
0.2	1.231	0.889	0.565	0.266	0.0	0.0
0.4	1.250	0.867	0.487	0.116	0.0	0.0
0.6	1.256	0.831	0.393	0.0	0.0	0.0
0.8	1.257	0.792	0.296	0.0	0.0	0.0
1.0	1.257	0.756	0.207	0.0	0.0	0.0

Table B4

ξ_{max} for n = 0.3 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	1.618	1.261	0.916	0.586	0.279	0.0
0.2	1.725	1.347	0.971	0.597	0.210	0.0
0.4	1.789	1.383	0.968	0.531	0.043	0.0
0.6	1.830	1.393	0.934	0.430	0.0	0.0
0.8	1.855	1.388	0.883	0.306	0.0	0.0
1.0	1.871	1.372	0.821	0.170	0.0	0.0

Table B5

ξ_{max} for n = 0.4 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	1.902	1.513	1.127	0.744	0.368	0.0
0.2	2.053	1.653	1.250	0.840	0.401	0.0
0.4	2.164	1.746	1.315	0.857	0.316	0.0
0.6	2.250	1.809	1.345	0.831	0.171	0.0
0.8	2.316	1.851	1.351	0.774	0.0	0.0
1.0	2.369	1.878	1.339	0.696	0.0	0.0

Table B6

ξ_{max} for n = 0.5 versus β and Z_s

Z_s \ β	0.0	0.2	0.4	0.6	0.8	1.0
0.0	2.000	1.600	1.200	0.800	0.400	0.0
0.2	2.182	1.778	1.371	0.957	0.520	0.0
0.4	2.334	1.919	1.495	1.050	0.527	0.0
0.6	2.462	2.033	1.586	1.097	0.464	0.0
0.8	2.572	2.125	1.650	1.109	0.352	0.0
1.0	2.666	2.199	1.692	1.095	0.207	0.0

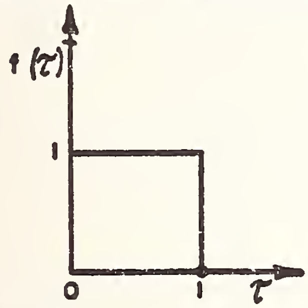
6.3 Appendix C

Computer Solution of Equations for Non-Zero Lading-Floor Friction and Backlash

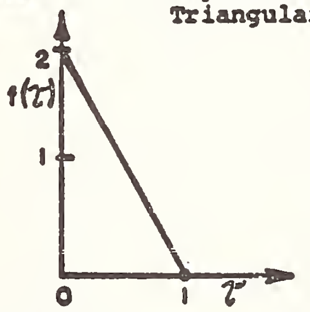
6.3.1 Description

The computer program determines the dynamic response of the lading in the struck freight car to the five different input pulse shapes shown below:

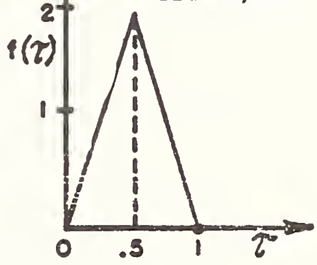
Case No. 1: Rectangular



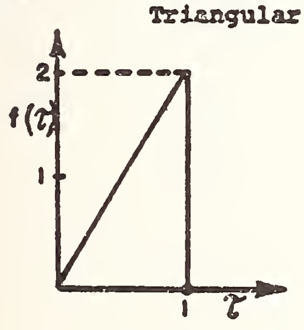
Case No. 2: Rapid-Rise Triangular



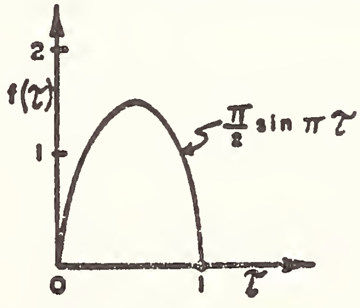
Case No. 3: Symmetrical Triangular



Case No. 4: Slow-Rise Triangular



Case No. 5: Half-Sine Wave



The program uses a Modified Euler numerical integration routine, in determining the maximum load factors for the lading response, for a time simulation period of five seconds. The results show the effect of varying the floor's friction coefficient and the force application period-to-natural period ratio.

6.3.2 Data Flowcharts

Figures C1 a and b show the data flowcharts for the computer solution of equations, involving non-zero lading floor friction and lading backlash.

Figure C1c shows a more detailed data flowchart for the calculations involving coulombic floor friction.

6.3.3 Input Data

The Following data is needed for input to the computer program:

TP	Time Period for Simulation
TI	Time Increment for Numerical Integration
NC(J)	Pulse Shape Code for Cases J=1,5 Code Value = 0, if case is not used = 1, if otherwise
BJ)	Floor Friction Values, J=1, 11
S(J)	Backlash Values, J=1, 11
R(J)	Force Application Period-to-Natural Period Ratio Values, J=1, 20

6.3.4 Output

The following output is printed in the computer program results:

K	Pulse Shape Case
S(J)	Backlash Value
B(J)	Friction Value
R(J)	Force Application Period-to-Natural Period Ratio Value
ZDM	Maximum Value of Lading Response for the Simulation Time Period

The maximum value of lading response, ZDM, is calculated for all combinations of S(J), B(J) and R(J), for each pulse shape case, K.

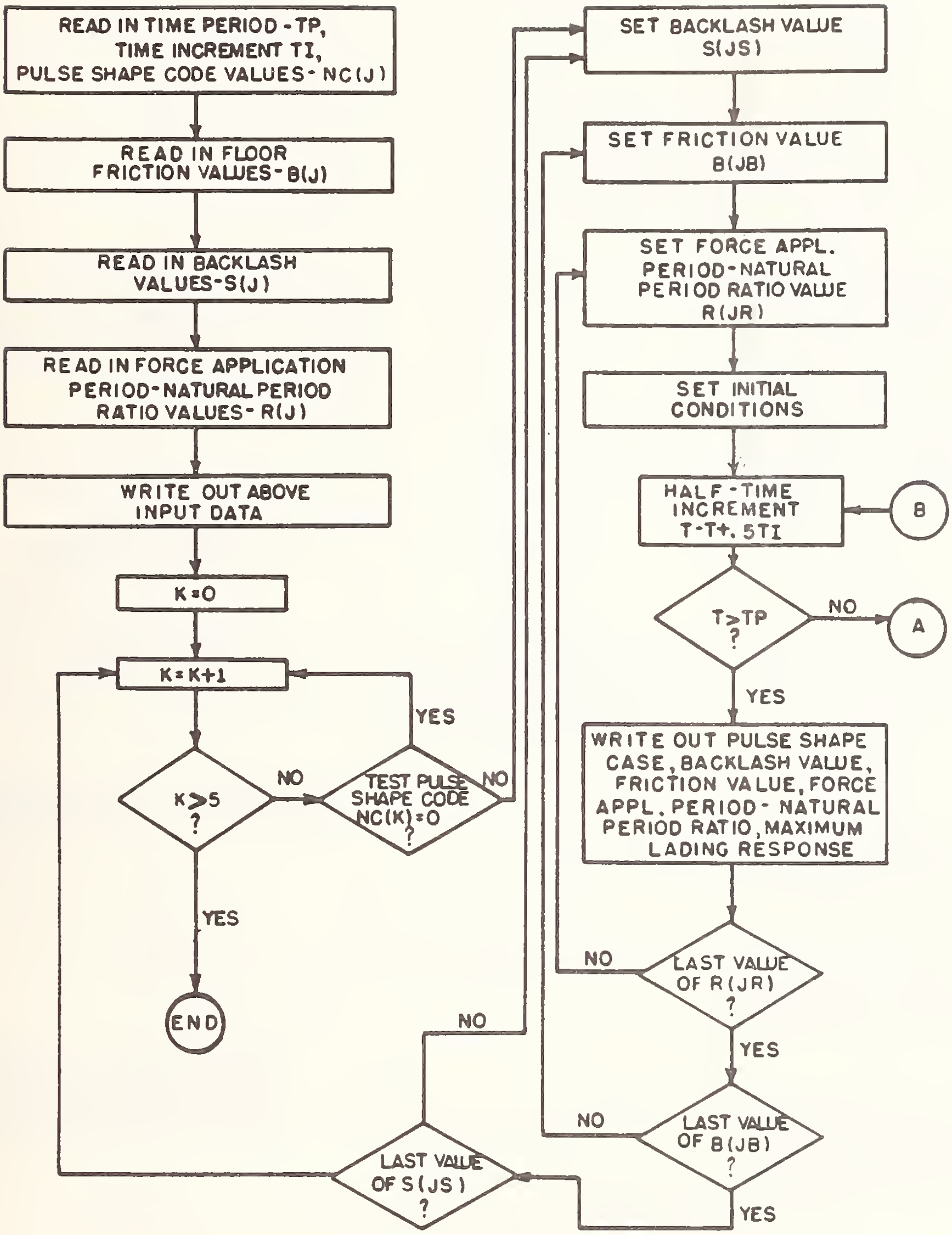


Figure C1a. Data Flowchart for the Computer Solution of Equations for non-Zero Lading Floor Friction and Lading Backlash.

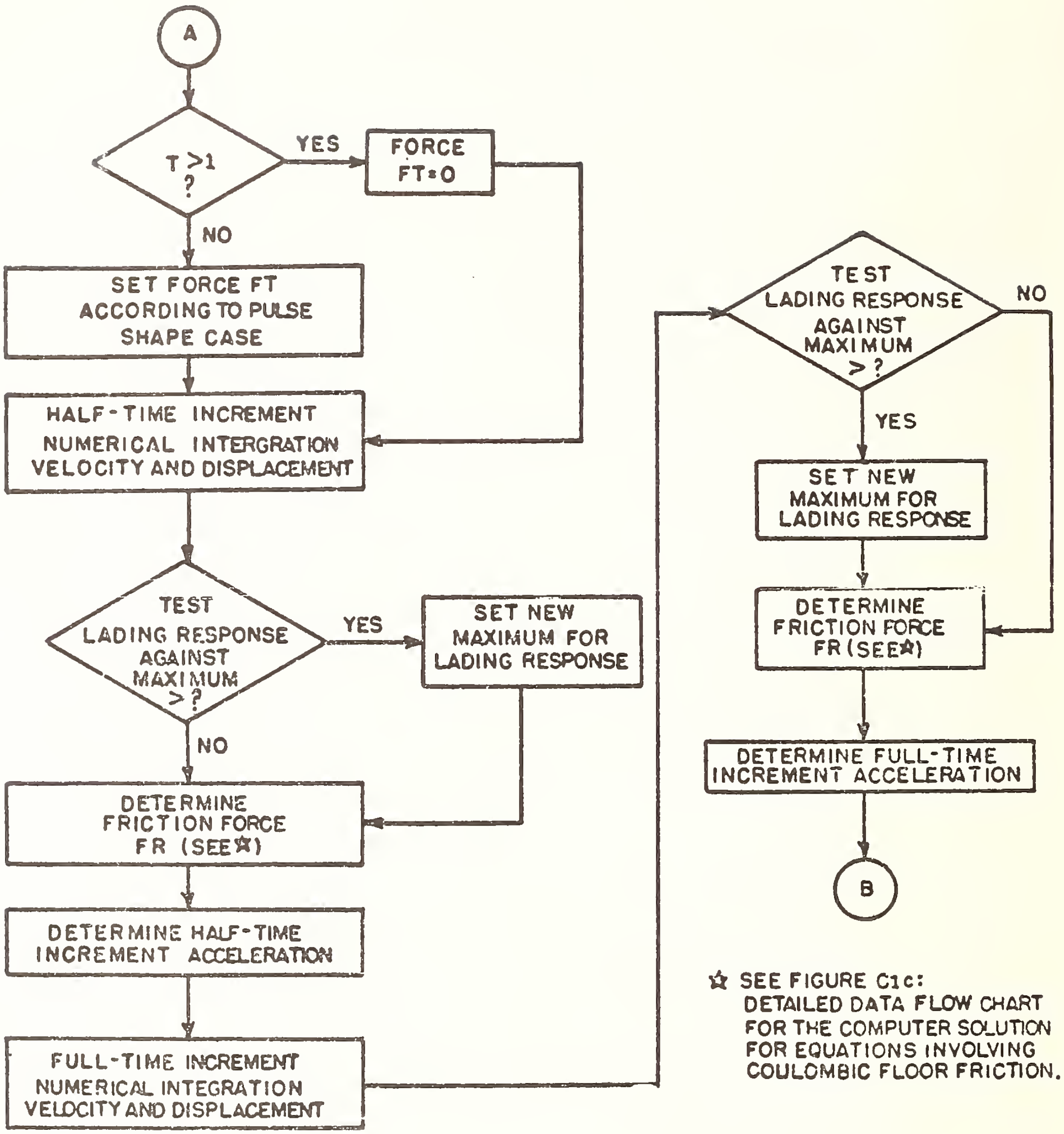


Figure C1b. Data Flowchart for the Computer Solution of Equations for Non-Zero Lading-Floor Friction and Lading Backlash.

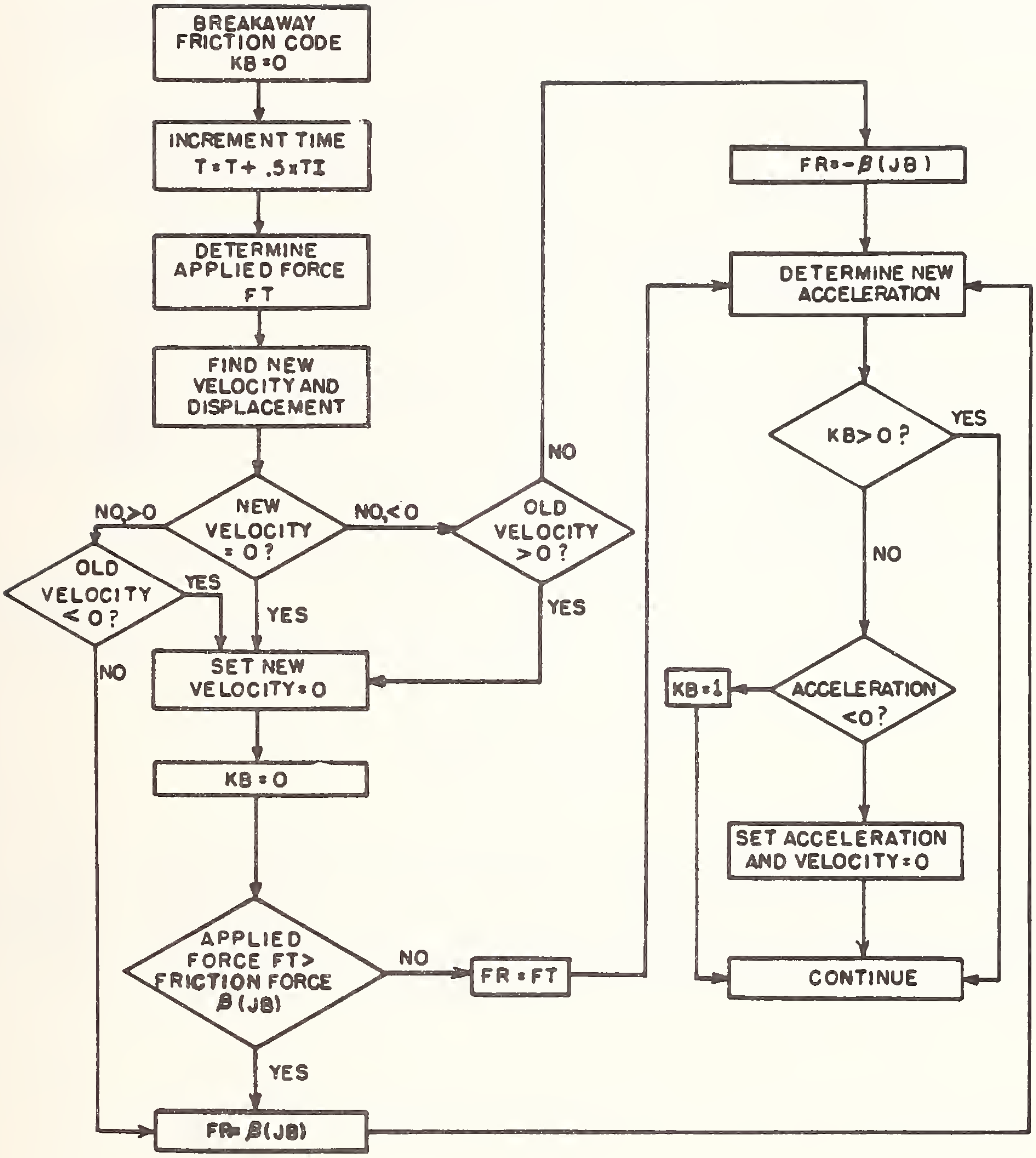


Figure C1c. Detailed Data Flowchart for the Computer Solution for Equations Involving Coulombic Floor Friction.

6.3.5 Case No. 1: Rectangular Pulse Input.
Figures C2 through C7 show the calculated maximum end-wall force ratio vs. lading flexibility ratio curves, for values of both Z_s and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0, respectively.

6.3.6 Case No. 2: Rapid-Rise Triangular Pulse Input.
Figures C8 through C13 show the calculated maximum end-wall force ratio vs. lading flexibility ratio curves, for values of both Z_s and β = 0.0, 0.2, 0.4, 0.6. 0.8 and 1.0, respectively.

6.3.7 Case No. 3: Symmetrical Triangular Pulse Input.
Figures C14 through C19 show the calculated maximum end-wall force ratio vs. lading flexibility ratio curves, for values of both Z_s and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0, respectively

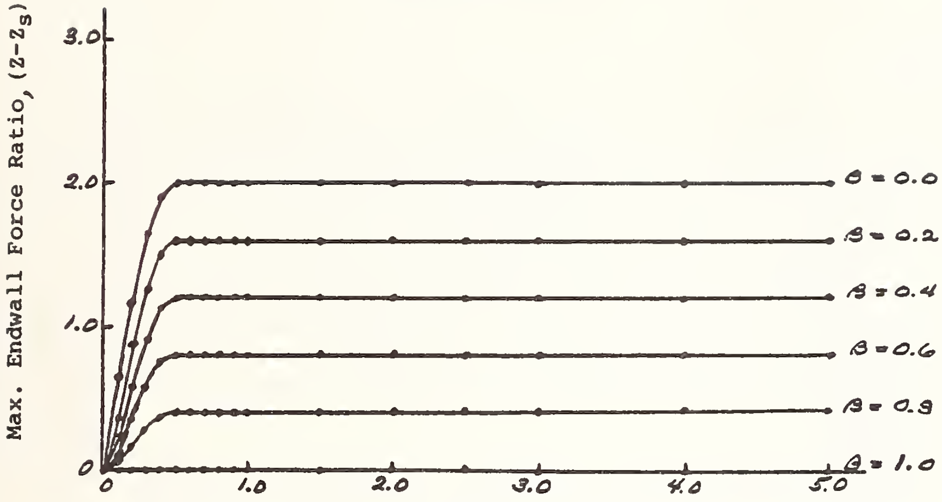
6.3.8 Case No. 4: Slow-Rise Triangular Pulse Input.
Figures C20 through C25 show the calculated maximum end-wall force ratio vs. lading flexibility ratio curves, for values of both Z_s and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0, respectively.

6.3.9 Case No. 5: Half Sine Wave Pulse Input.
Figures C26 through C31 show the calculated maximum end-wall force ratio vs. lading flexibility ratio curves, for values of both Z_s and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0, respectively.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the assistance of Dr. Robert Breese, who technically edited and checked the final report. The authors are also indebted to Mrs. Patricia Collier, for her accurate and neat typing of the text and many complicated mathematical equations, and to Mr. Raymond Trubic, who prepared some of the illustrations used in the Report.

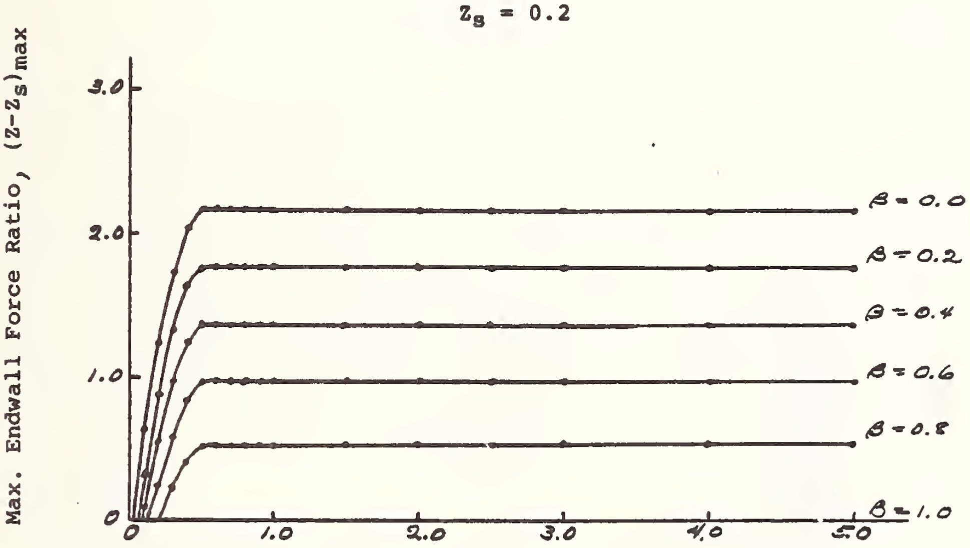
Case No. 1
$Z_s = 0.0$



Flexibility Ratio, n.

Figure C2. Case No. 1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for $Z_s = 0.0$ and $\beta = 0.0$, 0.2, 0.4, 0.6, 0.8 and 1.0.

Case No. 1
$Z_s = 0.2$



Flexibility Ratio, n

Figure C3. Case No. 1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for $Z_s = 0.2$ and $\beta = 0.0$, 0.2, 0.4, 0.6, 0.8 and 1.0.

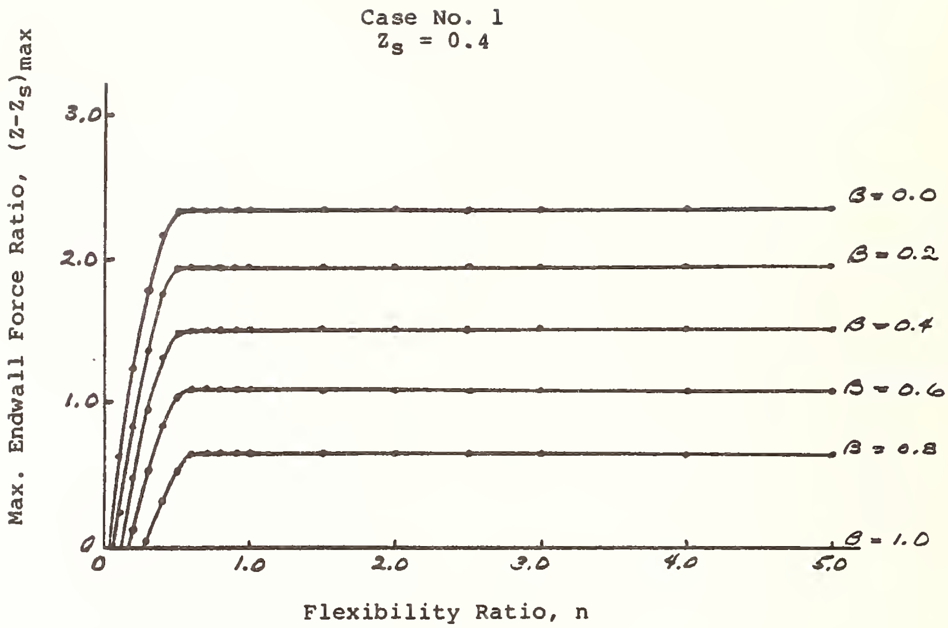


Figure C4. Case No. 1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.4 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

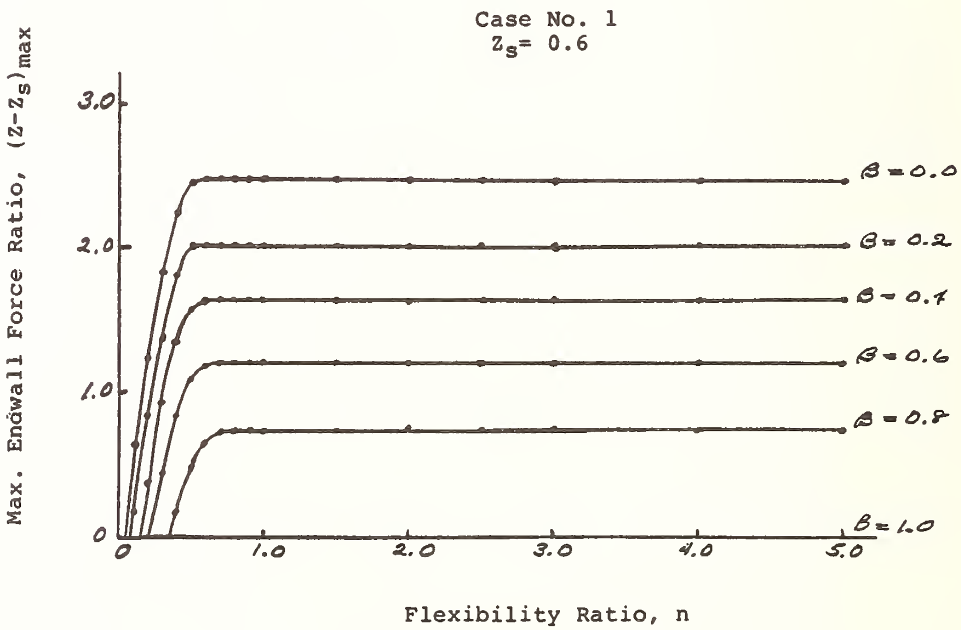


Figure C5. Case No. 1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.6 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

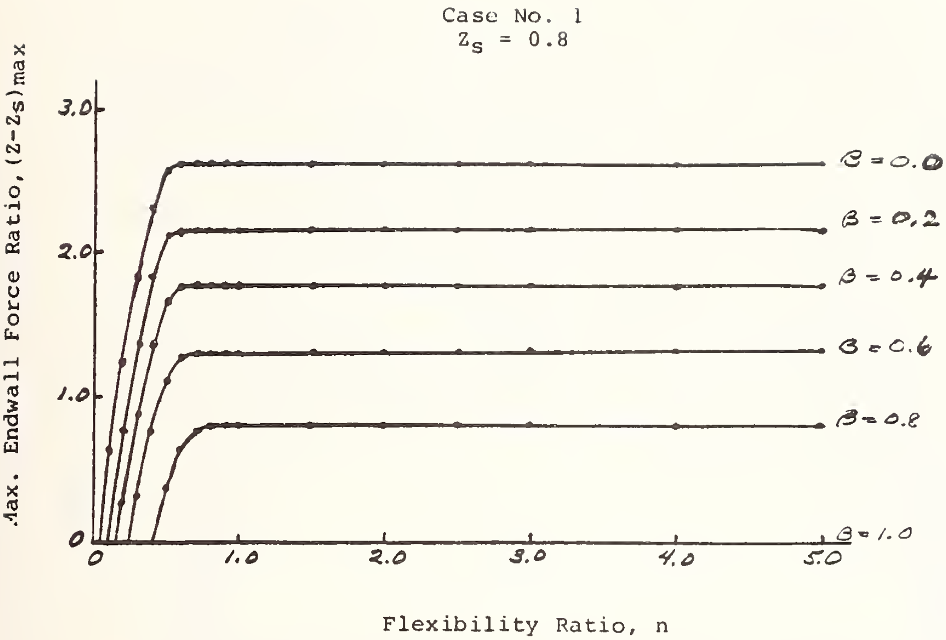


Figure C6. Case No.1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.8 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

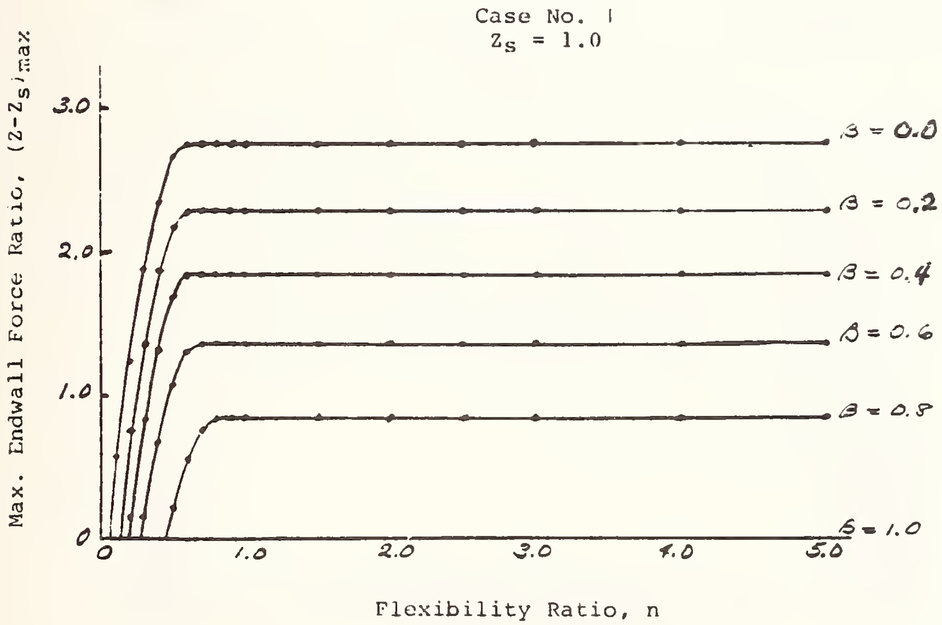


Figure C7. Case No. 1: Rectangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 1.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

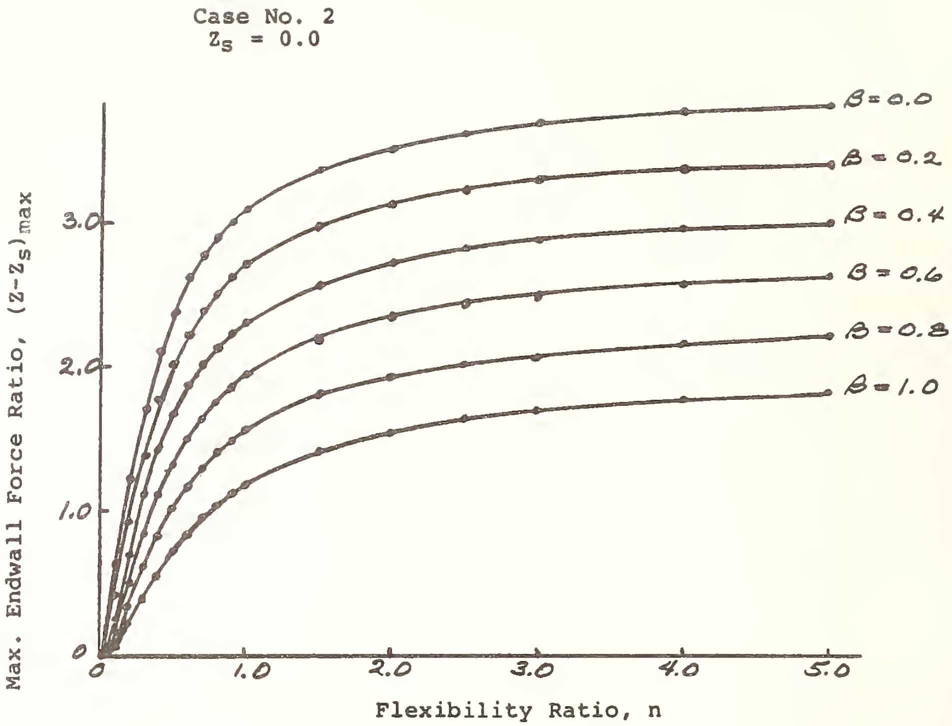


Figure C8. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

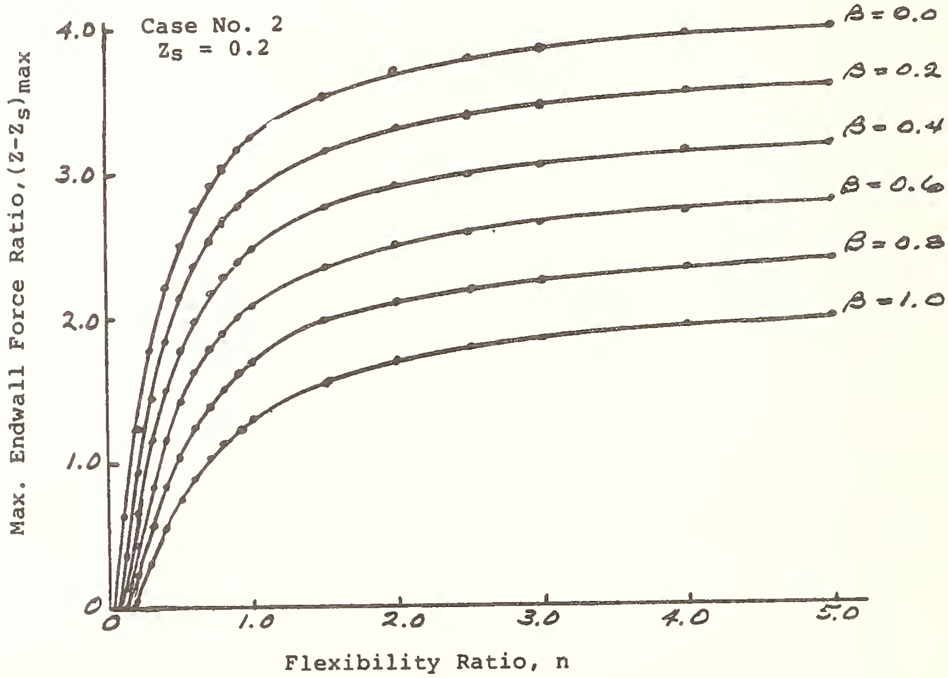


Figure C9. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.2 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

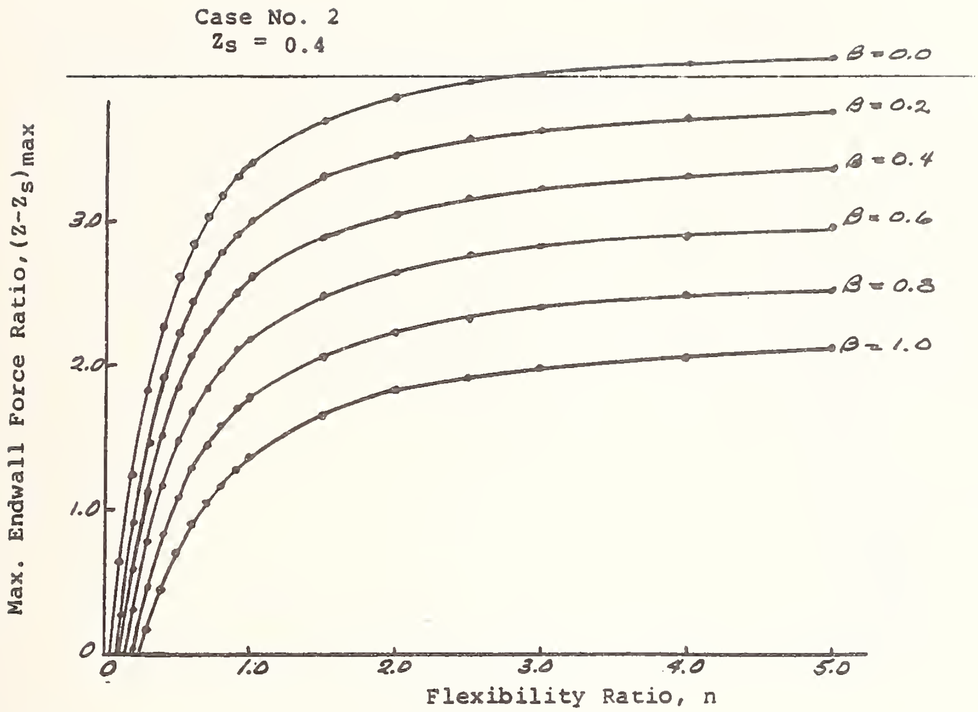


Figure C10. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for $Z_s = 0.4$ and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

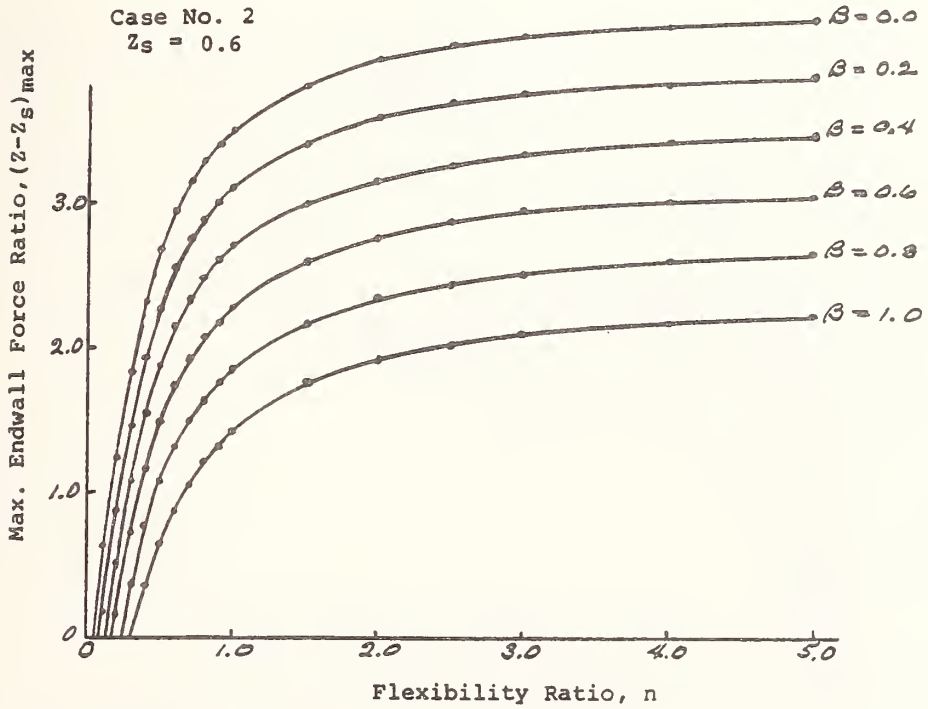


Figure C11. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for $Z_s = 0.6$ and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

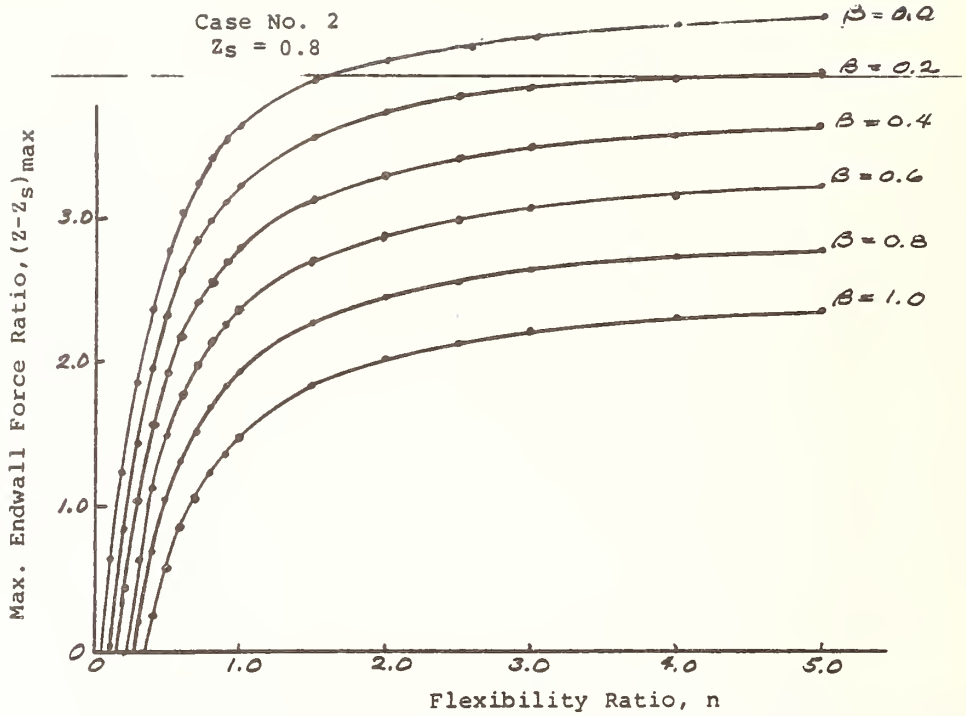


Figure C12. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.8 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

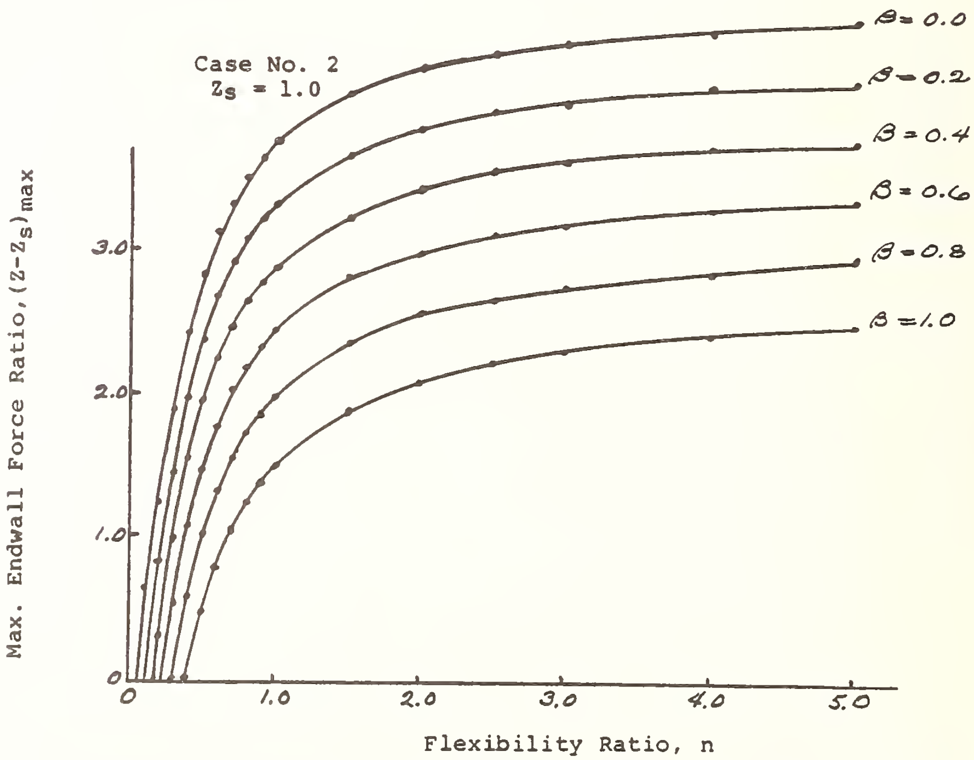


Figure C13. Case No. 2: Rapid-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 1.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

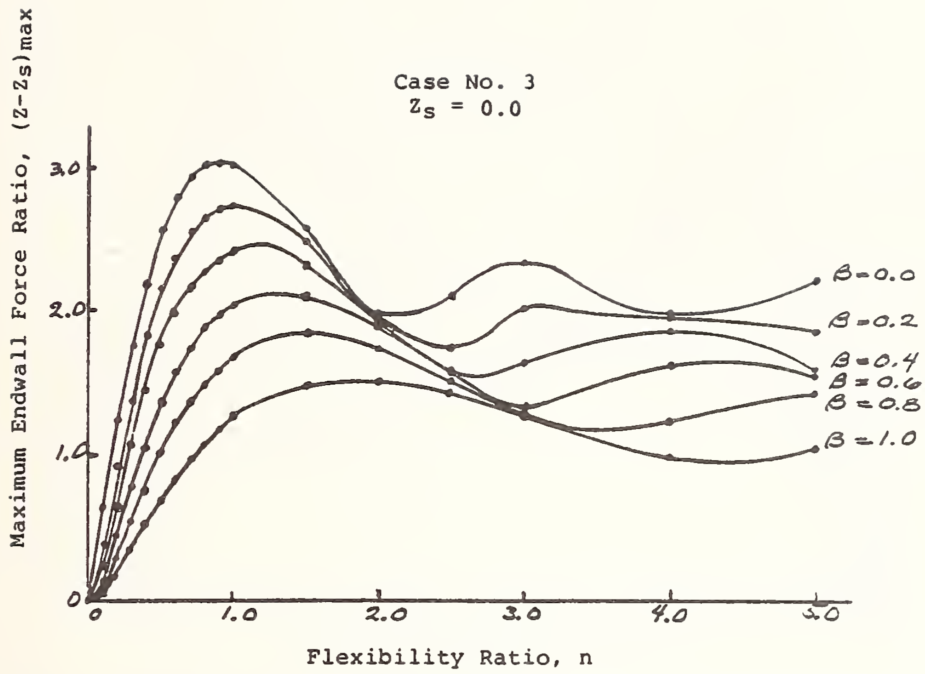


Figure C14. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

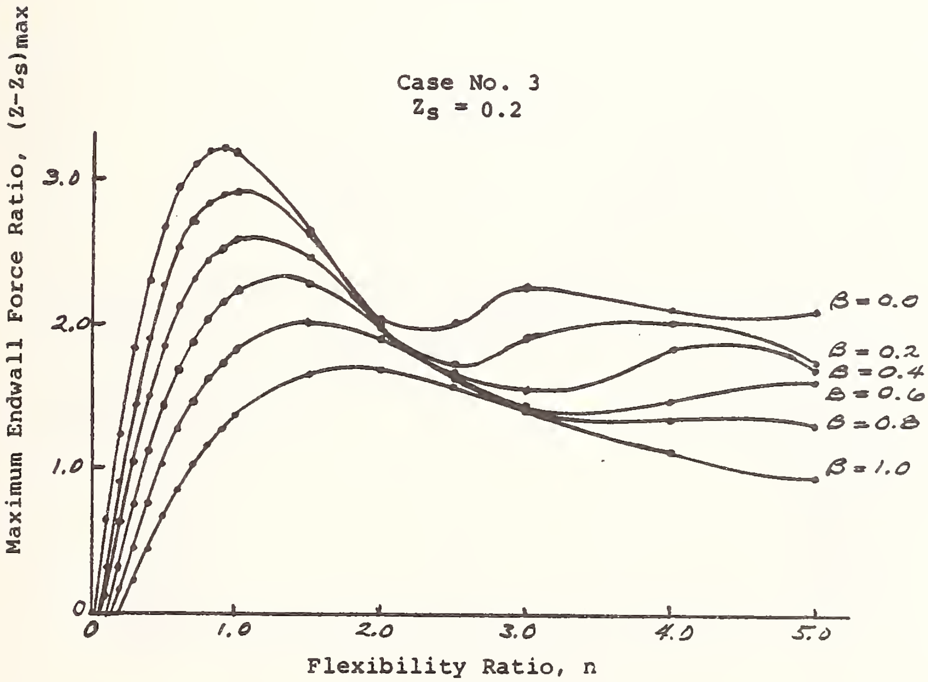


Figure C15. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.2 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

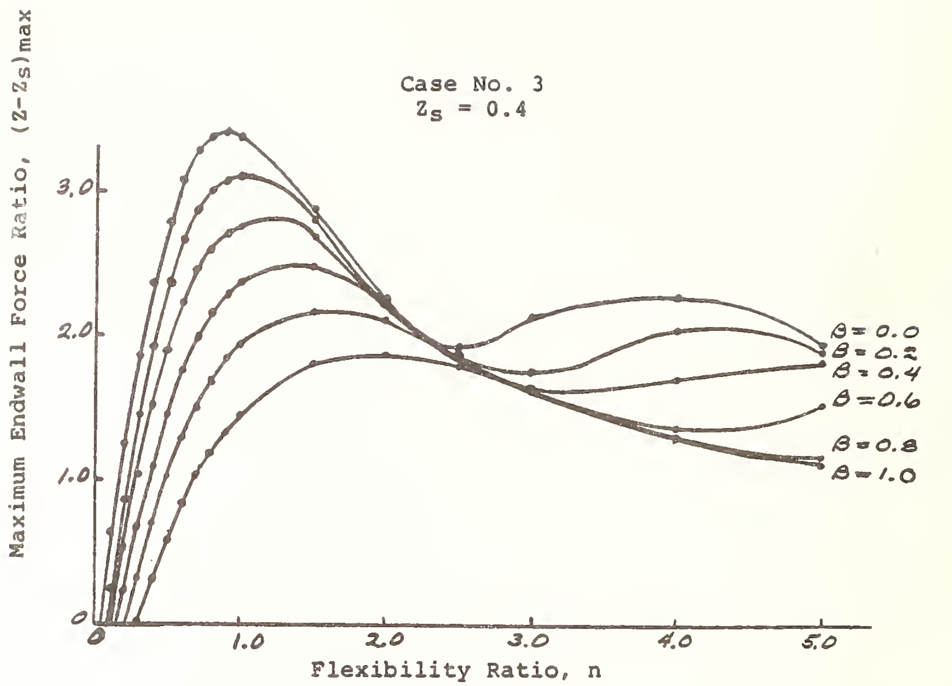


Figure C16. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.4 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

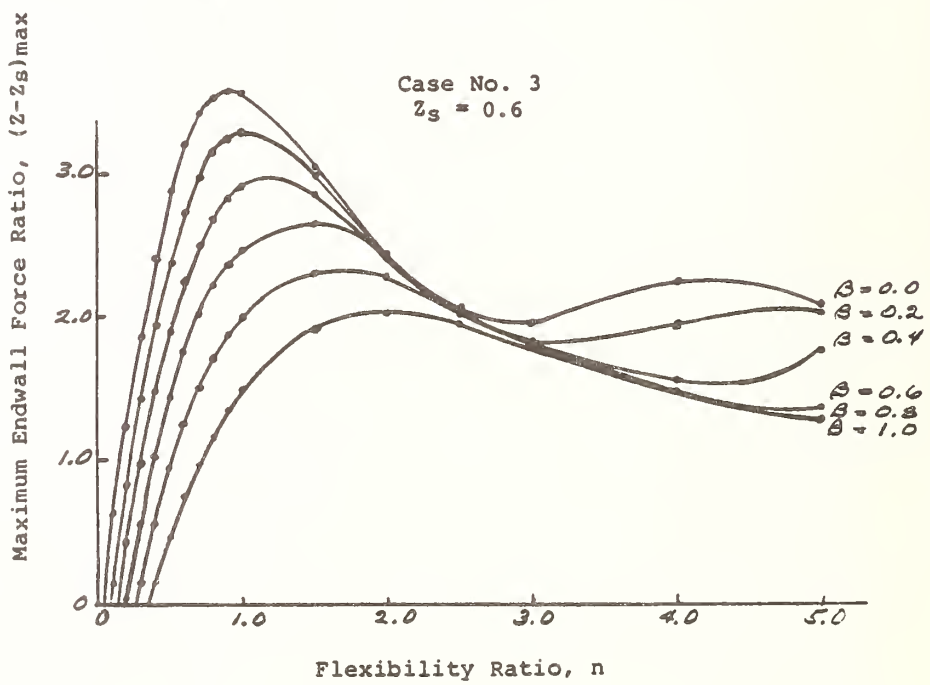


Figure C17. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.6 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

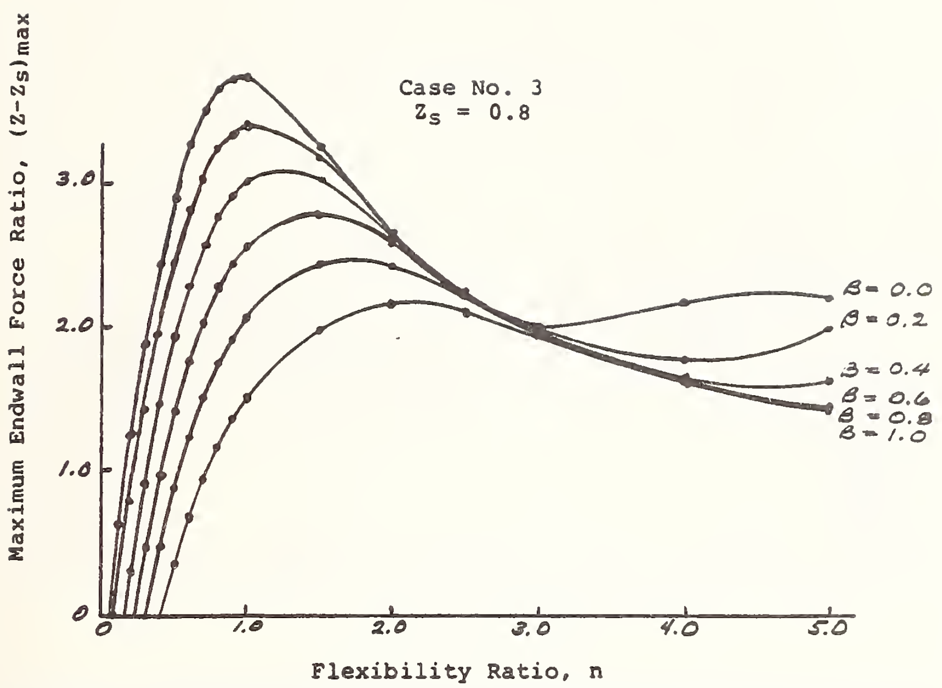


Figure C18. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.8 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

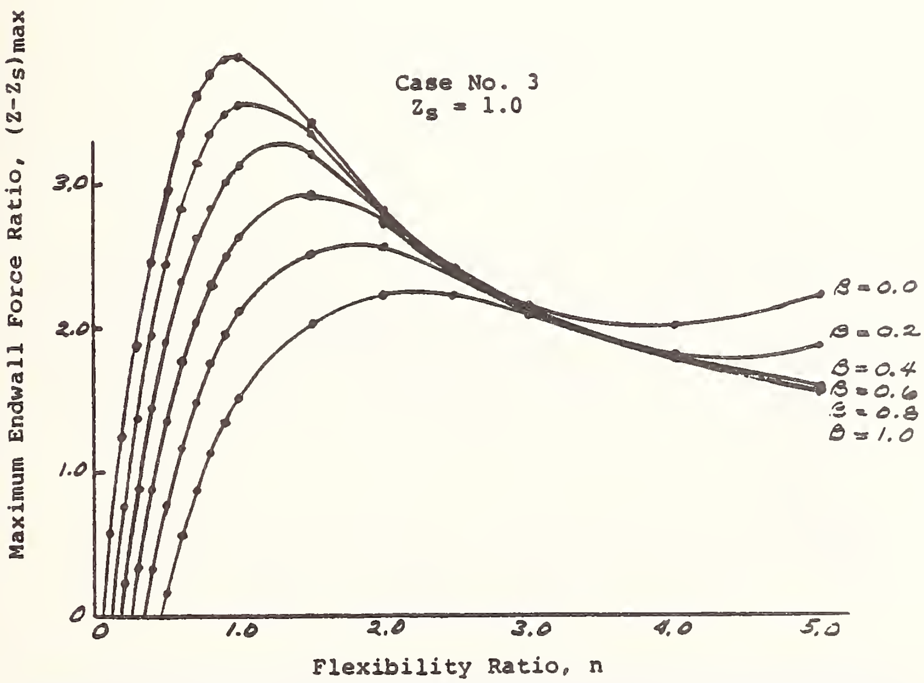


Figure C19. Case No. 3: Symmetrical Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 1.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

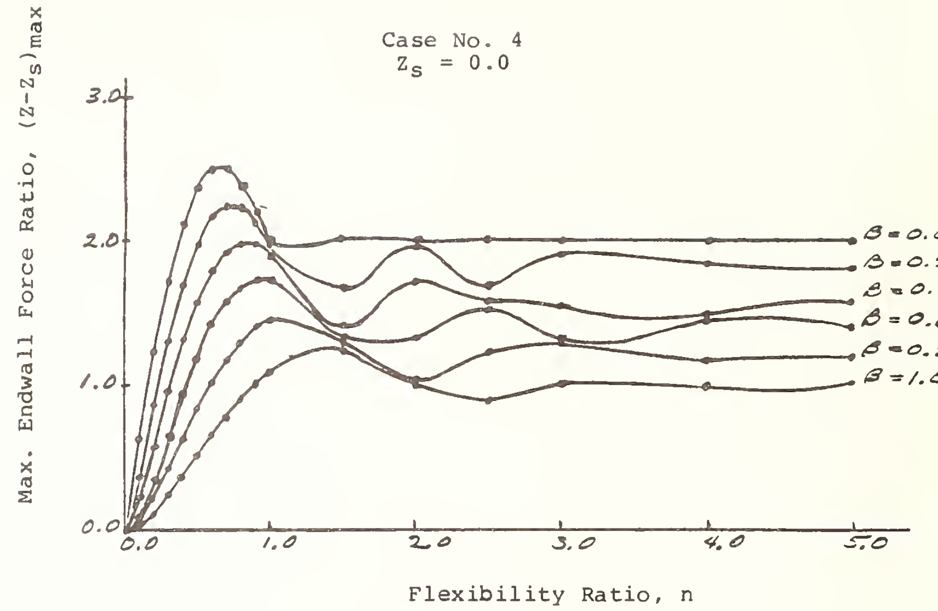


Figure C20. Case No.4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.0 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

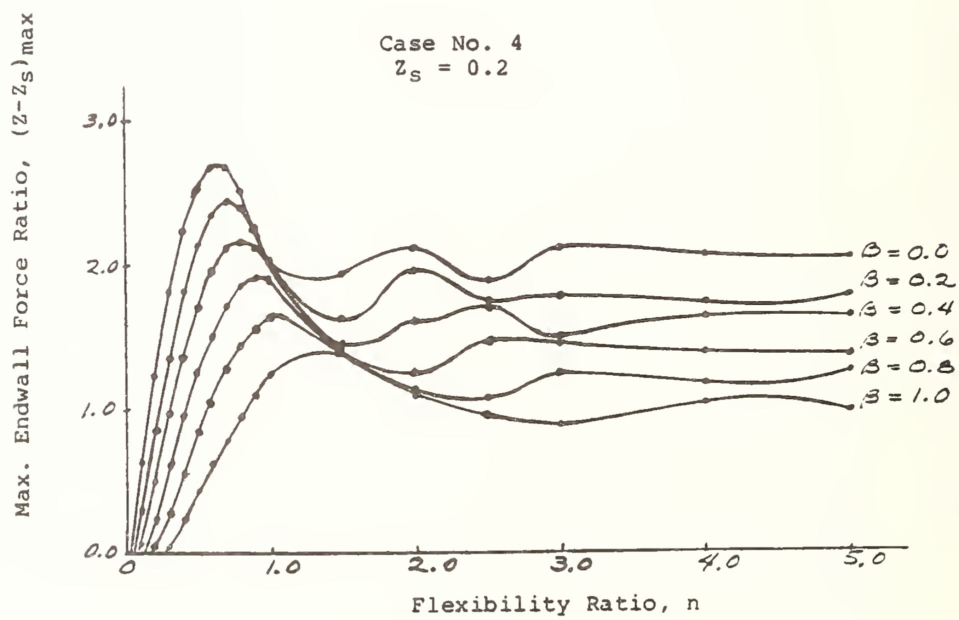


Figure C21. Case No.4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.2 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

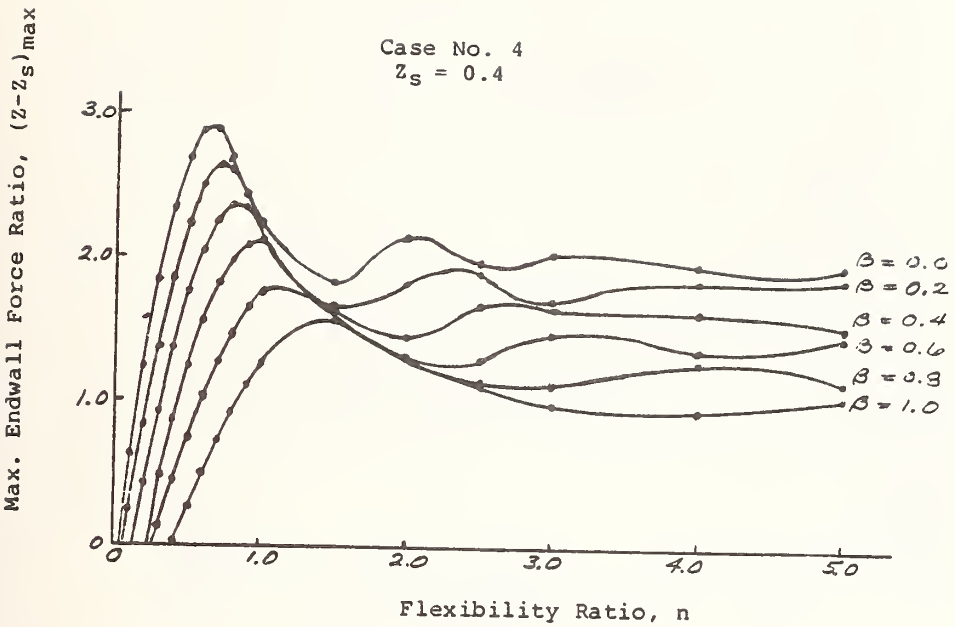


Figure C22. Case No.4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.4 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

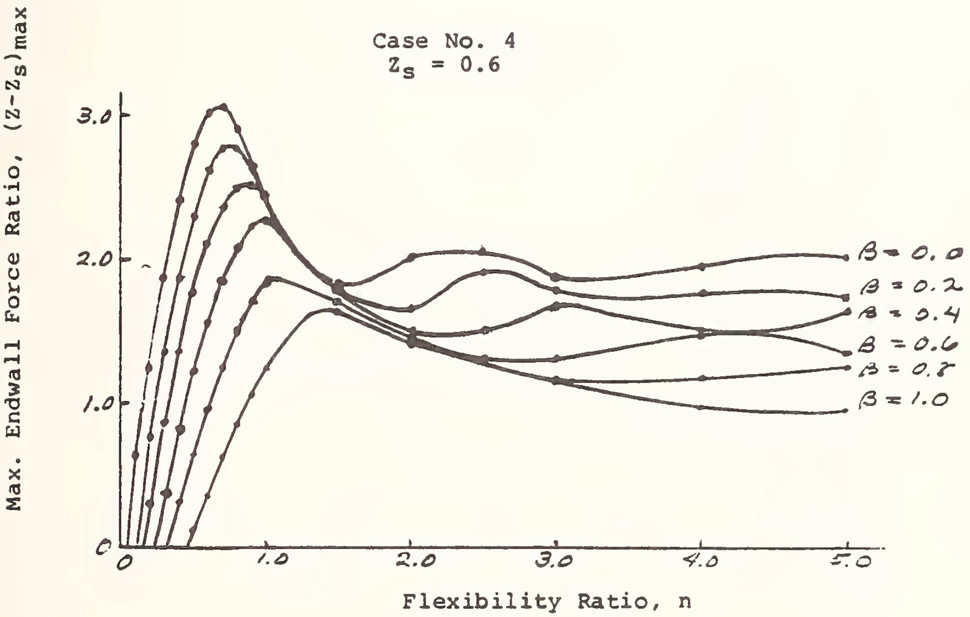


Figure C23. Case No. 4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.6 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

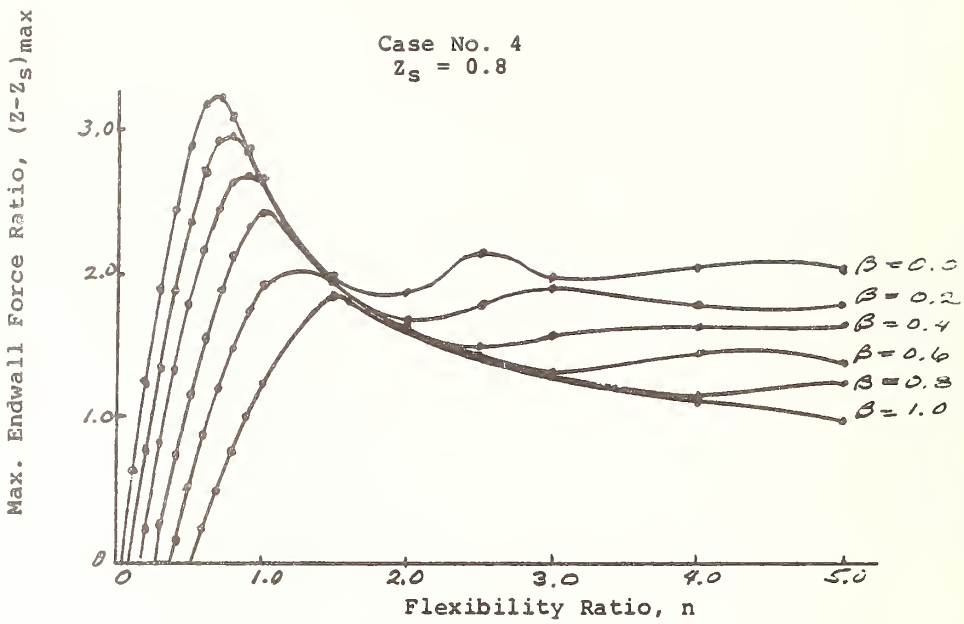


Figure C24 . Case No.4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s =0.8 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

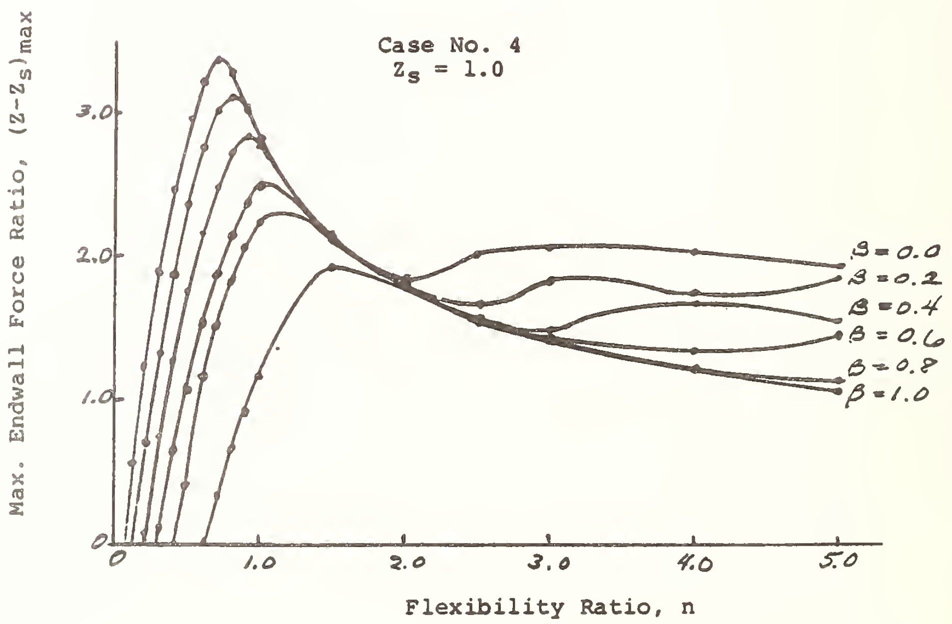


Figure C25. Case No.4: Slow-Rise Triangular Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 1.0 and β=0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

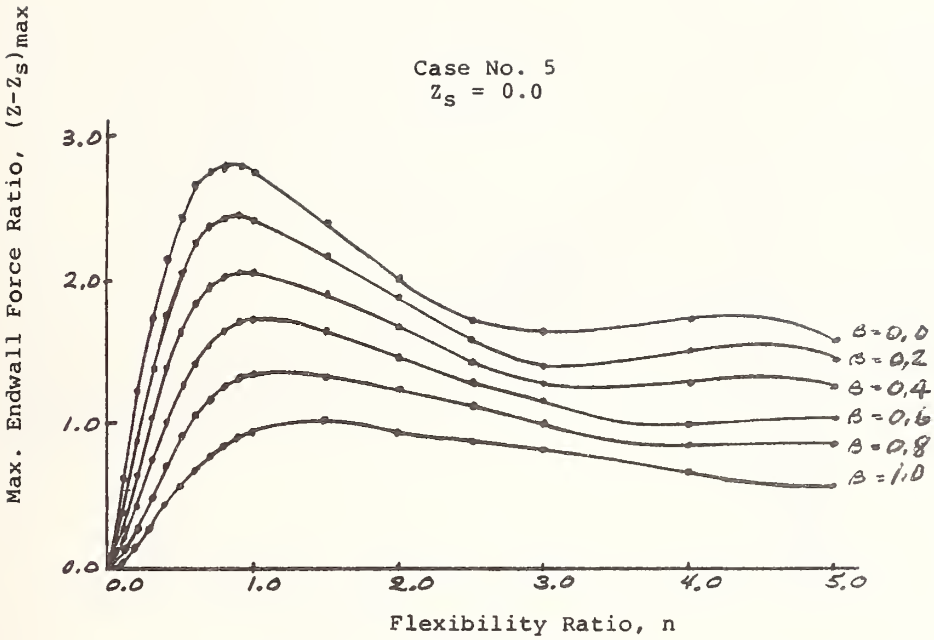


Figure C 26. Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

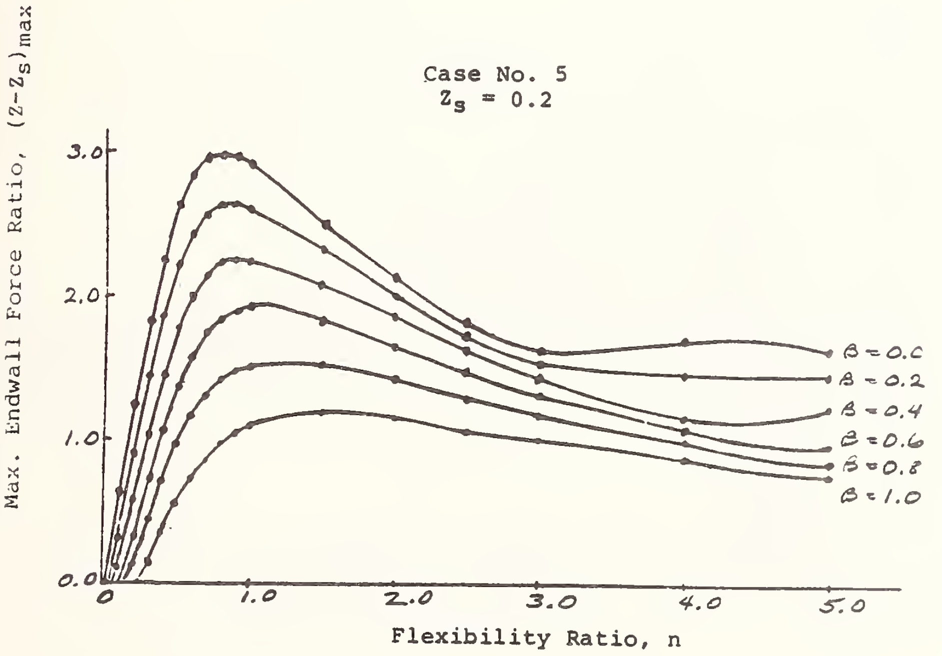


Figure C27 . Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.2 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

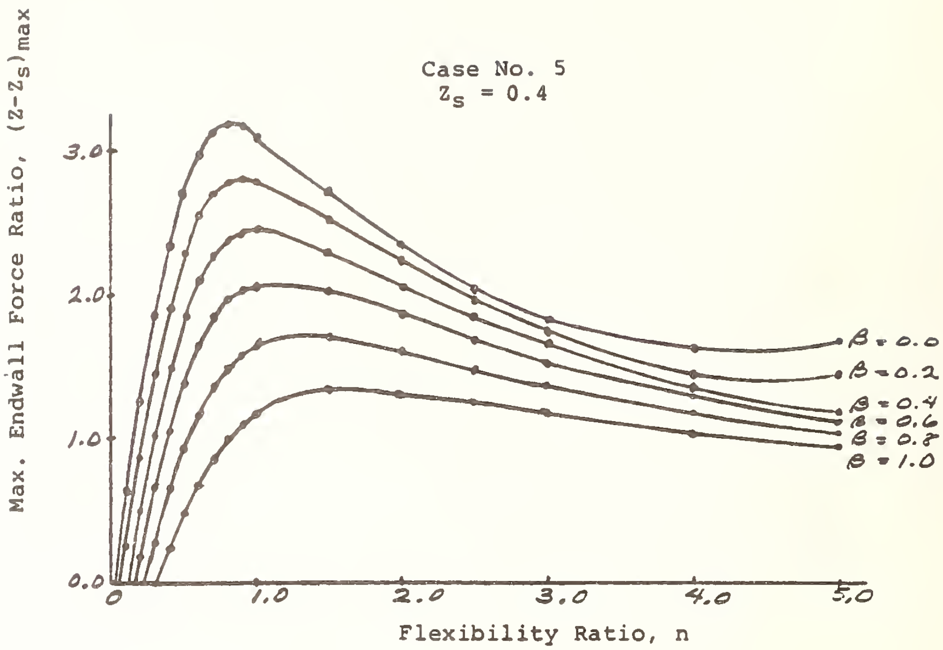


Figure C28. Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for $Z_s = 0.4$ and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

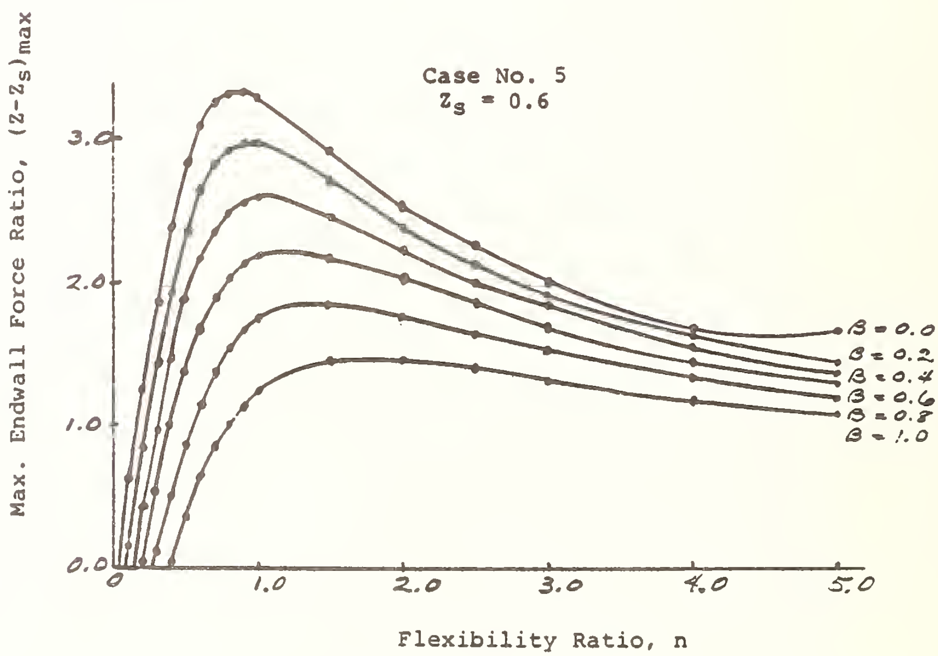


Figure C 29. Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.6 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

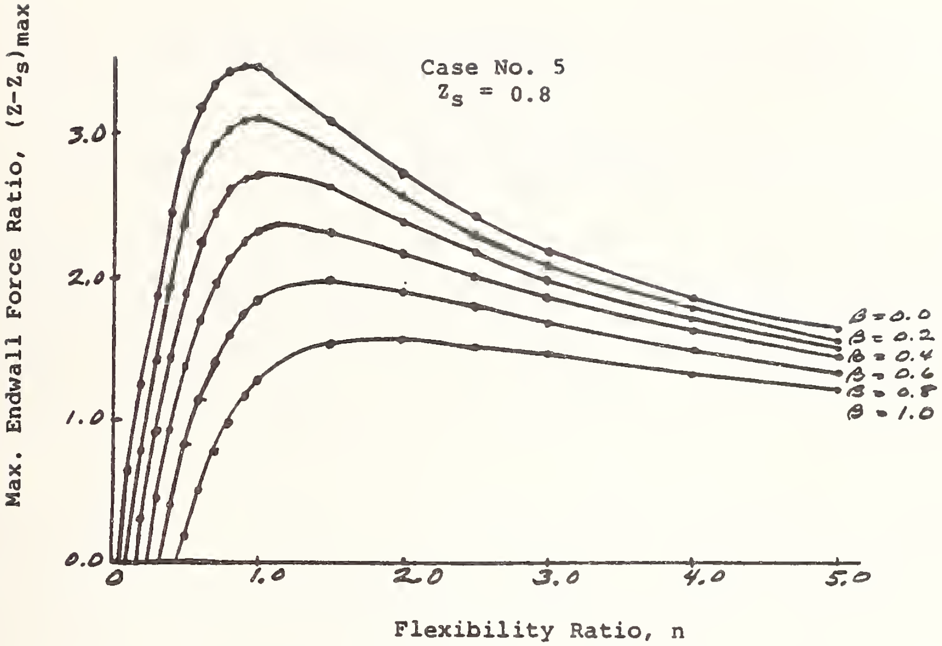


Figure C30. Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 0.8 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

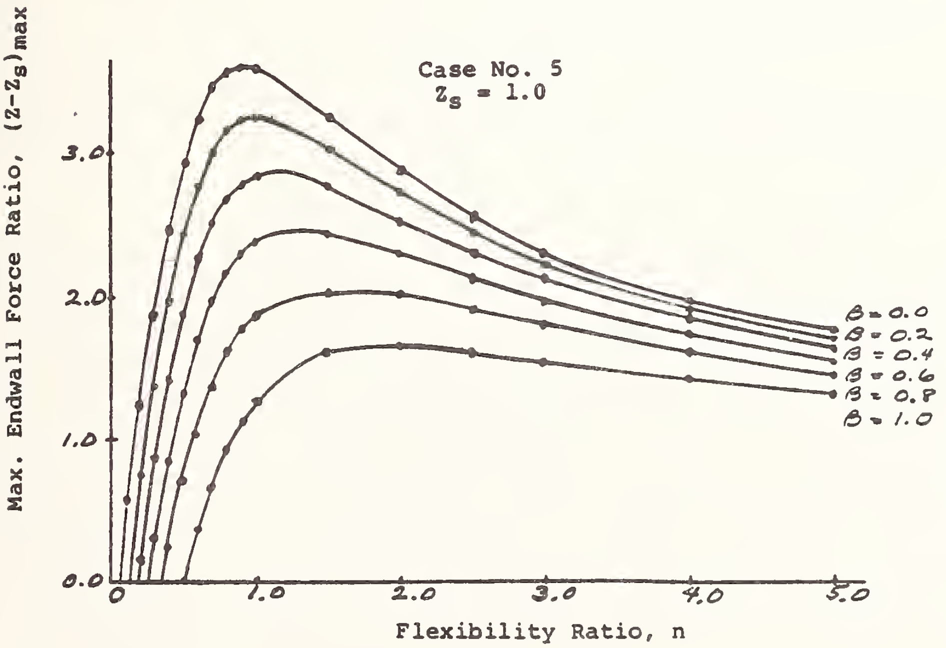


Figure C31. Case No. 5: Half Sine Wave Pulse Input. Maximum End-Wall Force Ratio vs. Lading Flexibility Ratio, for Z_s = 1.0 and β = 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0.

SYSTEM FOR TRAIN ACCIDENT REDUCTION--DOT STAR

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Abstract: The Department of Transportation-System for Train Accident Reduction (DOT STAR) study at the Naval Surface Weapons Center (NSWC) is developing a prototype system to help reduce the number of train accidents. NSWC has taken military technology and applied it to develop an on-train anti-derailment system. This system can sense a local derailment or a hot bearing. Upon sensing these conditions, the system automatically applies emergency braking.

Exploratory development hardware of the journal bearing thermal sensor successfully completed over 100,000 miles of travel. A roller bearing thermal sensor has also been designed. NITINOL is the key component use to sense over-heated bearings. The derailment detector uses the impact of a sensor foot with the rail head to sense a local derailment. Upon generation of a hot box or a local derailment, the sensors initiate a thermal pulse battery. The electrical pulse activates the air-valve which applies the train's brakes. To foster the success of the DOT STAR concept highly reliable, rugged and maintenance free military-developed components are being used. This document provides an overview of the DOT STAR program followed by a more detailed presentation of technical aspects of the sensors.

Key words: Bearing thermal sensor; DOT STAR; journal and roller bearing temperature transients; local derailment sensor; NITINOL; railroad safety system.

DOT STAR Program Initiation: If the Navy applied some of the technology they developed for weapons to help solve the railroad industry's derailment problem, what would result? The answer to this question is the Department of Transportation System for Train Accident Reduction (DOT STAR). The DOT STAR system was developed at the Naval Surface Weapons Center (NSWC) under the sponsorship of the Federal Railway Administration (FRA). This development was undertaken via the Department of Defense's Technology Transfer Program. (Nominally up to 3% of NSWC's manpower is made available to transferring military developed technology to other branches of the federal government, state and local governments, universities and private industry). In this instance, an NSWC employee, knowledgeable of the needs of the railroad industry, conceived of a system using current Navy technology which could solve some aspects of the railroad derailment problem. A proposal was submitted to the FRA, was accepted, and the DOT STAR program began.

Establishing Ground Rules and Objectives: The overall objective of reducing the number of train accidents can be met in a number of ways, some of which are to:

a. Increase the quality and/or quantity of inspections and preventive maintenance. This approach must be evaluated by the railroad industry.

b. Develop wayside monitoring systems which would sense conditions related to imminent derailments. The wayside IR hot bearing sensors, for example, have been in use for a number of years. A good deal of research has already been done in this area.

c. Develop an on-train detection system. This approach is the one selected for DOT STAR. For an on-board system to be acceptable, it should have the following basic features:

- o Independent/self-contained
- o Very little or no maintenance
- o Low cost
- o Very low false alarm rate, and
- o Monitors multiple causes (for increased cost effectiveness)

All of these features clearly tie back to life cycle cost and reliability

DOT STAR System Overview: The DOT STAR system is illustrated in Figure (1). It has a thermal sensor at each wheel to detect an overheated wheel bearing and a local derailment detector (to detect when an axel derails). These are both ordinary causes of major derailments that are train (vs. track) related and can be detected and corrective action taken. The thermal or local derailment sensor electrically sends a signal to a valve which applies emergency braking to the train.

The temperature of the bearing is monitored with the aide of a unique alloy developed by the Metallic Material Research Branch at the Naval Surface Weapons Center. The metal is a nickel titanium alloy called NITINOL. The unique property of NITINOL is that the temperature at which it undergoes its phase change can be controlled well and once it is heated to its phase change temperature, the metal will suddenly revert back to the shape it had when it was last in that higher temperature phase. In the journal bearing thermal sensor, a NITINOL retaining ring holds back a spring loaded firing pin. The NITINOL was straight when it was last in its higher temperature phase, so, when the temperature of the NITINOL retaining ring in the thermal sensor is evevated to the phase change temperature, the ring suddenly straightens out releasing the firing pin. The firing pin strikes the base of a thermal pulse battery to activate it. The battery provides a signal to initiate emergency braking. The thermal sensor for roller bearing cars is configurated differently than the journal bearing system described above, but it also uses NITINOL to sense the temperature.

The thermal pulse battery is the only source of electrical energy in the system. It is a battery developed by the military and is used in many weapons. It is extremely rugged, has an extremely long shelf life, will supply full power immediately at temperature extremes from below -60°F to 160°F and is very reliable. It is the only type of battery known which meets the environmental and functional criteria of DOT STAR.

The local derailment sensor is basically a contract sensor located outboard (on the field side) of each wheel. When the wheel derails inward (on the gage side), it drops and the railhead strikes and depresses the contact sensor which is located just outside the wheel. A trigger mechanism in the contact sensor then fires the thermal battery.

After the thermal battery is fired, the wiring conducts the signal to the air-valve. The air-valve is located in the air clean out system of the railcars air brake lines. The air-valve vents the train's air brake system to apply emergency braking to the train.

Thermal Sensors: The geometry of roller bearings and journal bearings are quite different, but the basic problem of finding a practical location to mount a thermal sensor is the same. A reasonable lag time between a temperature change at the wheel bearing and a corresponding change at the thermal sensor location is essential if the sensor is to avert a derailment. Identifying that location involved two series of field tests as well as laboratory tests and finite element thermal analysis for the roller bearing as well as the journal bearing thermal sensors.

Journal Bearing Thermal Sensor: The first series of field tests in November 1974 revealed that it would be impossible to detect an overheated journal bearing by monitoring the temperature of the journal housing. The second series of field tests conducted in July of 1975 had a sensor as shown in Figure (2). This figure also indicated where the thermal sensors and thermocouples were located. Note the thermal sensor penetrates the journal bearing housing and the sensing head is spring-loaded to remain in contact with the wedge. Figure (3) illustrates the top surface of the bearing wedge follows the temperature of the bearing under normal operating conditions. Figure (5) illustrates how the temperature of the surface of the wedge followed the bearing during a generated hot box. During this series of tests, operative sensors were installed at the right and left wheels. The left bearing sensor actually stopped the train when the sensor reached approximately 240°F — the designed activation temperature). Reference (a) provides additional information on these journal bearing thermal sensor tests.

The journal bearing thermal sensors were installed on four ore cars at the DM&IR Railway in Minnesota and were left to ride around for five years of normal operation. They saw well over 100,000 miles. They were all removed from the cars and tested in the laboratory. Each sensor activated its thermal battery properly and the battery provided the required power to initiate the air valve.

Roller Bearing Thermal Sensors: As with the journal bearing thermal sensor, finding a location that is practical and provides a good response time is equally important for roller bearing thermal sensors. To identify that location, a series of tests were conducted at the Transporation Test Center in Pueblo, Colorado, during December 1977 and July 1978.

The first step in the testing was to determine the operating temperature of the roller bearing at various loading conditions, velocities and ambient temperatures. It was assumed the 240^{o}F chosen for the journal bearing activation temperature would also be good for the roller bearing sensor. Figure (6) illustrates how a roller bearing is mounted on a sideframe. The air blowing around and cooling the outer surface provied to be a major factor in selecting a sensor location point. The roller bearing and adapter are further illustrated in Figure (7). This figure shows the heat indicator holes designed into the adapter but they are seldom used. The base of these holes, if they were drilled to the proper size and tapped, were determined to be a possible location for the thermal sensor, which would not be as influenced by the convective cooling.

During the tests in July 1978, thermocouple #1 was imbedded in the saddle of the adapter. It was in contact with the race of the roller bearing directly above the outer bearing, which was run in a non-greased condition to induce bearing seisure. The temperature sensed by the thermocouple #1 was considered to be the operating temperature of the bearing. There were a total of 14 thermocouples installed to obtain data to develop the heat transfer model for the roller bearing system and evaluate sensor locations.

Figure (8) illustrated the temperature of thermocouples during the attempt to generate a derailment due to roller bearing seisure. Thermocouple #11 was located at the outer surface of the adapter where sensor #1 is attached. Thermocouple #8 was located in thermal sensor #3 which was located in a heat indicator hole. As can be seen from the figure, sensor #3 triggered in a timely manner and in general, thermocouple #8 followed thermocouple #1 quite well. Thermocouple #11 responded to the temperature rise in the bearing nearly as rapidly as thermocouple #8 but at a lower rise rate so it only reached 180^{o}F during the time the bearing was seized. Thus for sensor #1 to activate in a timely manner it should have a NITINOL element which activated around 180^{o}F. Generating and maintaining a seisure of a roller bearing is not a simple problem, especially if you do not wish to cause an axel failure to occur at high speed. See reference (b) for an explanation of the unusual velocity profile and other details of these two test series and the finite element analysis which follows.

A heat transfer analysis of the roller bearings was performed using a computer code designed by the Army to evaluate head conduction in rocket motor nozzles. A portion of the model used is illustrated in

Figure (9). The modeling of the adapter has been omitted for clarity. The heat generation rate and convective heat transfer coefficients were selected using temperature data generated during testing. After the model was developed, it was run for various velocities, ambient temperatures, etc. The model indicated that the sensor located in the heat indicator hole would reach 240°F at the time the surface of the roller bearing reached 300°F. The analysis predicted a 300°F bearing operating temperature would only result from conditions beyond worst case for normal train operation anywhere in the world. TIMKEN Roller Bearing Company engineers verified that they would not want roller bearing to operate at 300°F for any sustained period. Therefore, 240°F appears to be a good activation temperature for a sensor located in the heat indicator hole.

At the time of the study using a thermalsensor very similar to the journal bearing sensor and locating that sensor in the heat indicator hole seemed like the best approach. During the design phase, as attempts were made to address the AAR regulations and the vulnerability of the sensor in the railroad environment, the face of the adaptor was considered a much more practical location for the sensor.

The roller bearing system finally designed is shown in Figure (10). The NITINOL wire in this design shortens when it reaches its phase change temperature (approximately 180°F). The NITINOL pulls a cable which trips a lever in the derailment sensor to fire its battery. (This saves the cost of a separate battery for the thermal sensor). Laboratory tests have been conducted on this new design, but no field tests have been conducted to verify it.

Local Derailment Detector: The original approach for a local derailment detector was to develop a seismic device that would trigger when the wheel dropped from the railhead to the road bed or in the shocks from riding over the railroad ties. It was determined, however, that the on-track shock environment (including coupling shocks, etc.) varies sufficiently between winter and summer with velocity and with various track conditions that a non-sophisticated seismic type which would reliably sense a local derailment but not give false alarms would be difficult to design. Early seismic type sensor work including test data is included in reference (c).

The present local derailment sensor uses a contact sensor. A contact foot is located 3-1/4 inches above the railhead just outside of the wheel. During a local derailment the sensor adjacent to the wheel which derails on the gage side, is activated by impact with the railhead as the wheel drops. See Figure (11). The sensor is designed to function only on vertical loads in excess of 7,000 pounds. A 7,000 pound load will shear the pins allowing the foot to move upward into the sensor body. When the foot is pushed far enough upward, a trigger mechanism is released and a thermal battery activated. The sensors are located so as to be partially shielded by the side frames.

During recent tests at the DM&IR Railway, the local derailment detector gave a false alarm at a grade crossing where the freezing conditions had heaved the asphalt up as much as 3-3/4 inches above the railhead. This raised asphalt (somewhat like a ramp) struck the underside of the contact foot and activated the sensor and stopped the train. The location of the local derailment detector can be changed to reduce the probability of this recurring. Subsequent to the impacts with the grade crossing, two other derailment detectors have been lost. Damage done at the grade crossing is suspected as the cause.

Air-Valve: After a thermal battery is fired either by the thermal sensor or local derailment sensor, it sends a pulse of electricity to the air-valve. The air-valve is located in a little extension added to the air clean-out assembly. See Figure (12). The air-valve is powered by a pyrotechnic gas generator. The gas generator pushes a piston which punctures a diaphragm to allow the pressure in the brake lines to vent. An orifice can be added to the air-valve to limit the vent rate to that of standard braking. Emergency braking is the vent rate used in the test hardware.

NSWC has a great deal of experience developing pyrotechnic compounds and devices. This device is completely enclosed, containing even the gases generated. The pyrotechnic has the standard one watt, one amp no fire safety feature to prevent activation due to the discharge of static electricity. The conduit and junction boxes protecting the wiring from the railroad environment inherently provides the required shielding against electromagnetic radiation. This shielding prevents currents (and the resulting braking) from being induced by transmissions from local radio stations, radar, etc.

Concluding Remarks: NSWC, through the sponsorship of the FRA, has done a lot of work developing and proving the technology for building the DOT STAR system. A detailed description of some of the testing (including a lot of data) can be found in references (a), (b), and (c). The design of the thermal battery and the environmentally protected wiring system for DOT STAR has been omitted to meet the presentation time limitation.

REFERENCES

a. FRA/ORD-80/76, PB 81-127458, Thermal Sensing Unit Test for Railroad Car Journal Bearings, October 1980

b. FRA/ORD-80/43, PB 80-193758, Railroad Car Roller Bearing Temperature Measurement and Analysis, April 1980

c. FRA/ORD-80/75, PB 81-127466, G-Sensing Derailment Detector Final Report, October 1980

JOURNAL BEARING APPLICATION

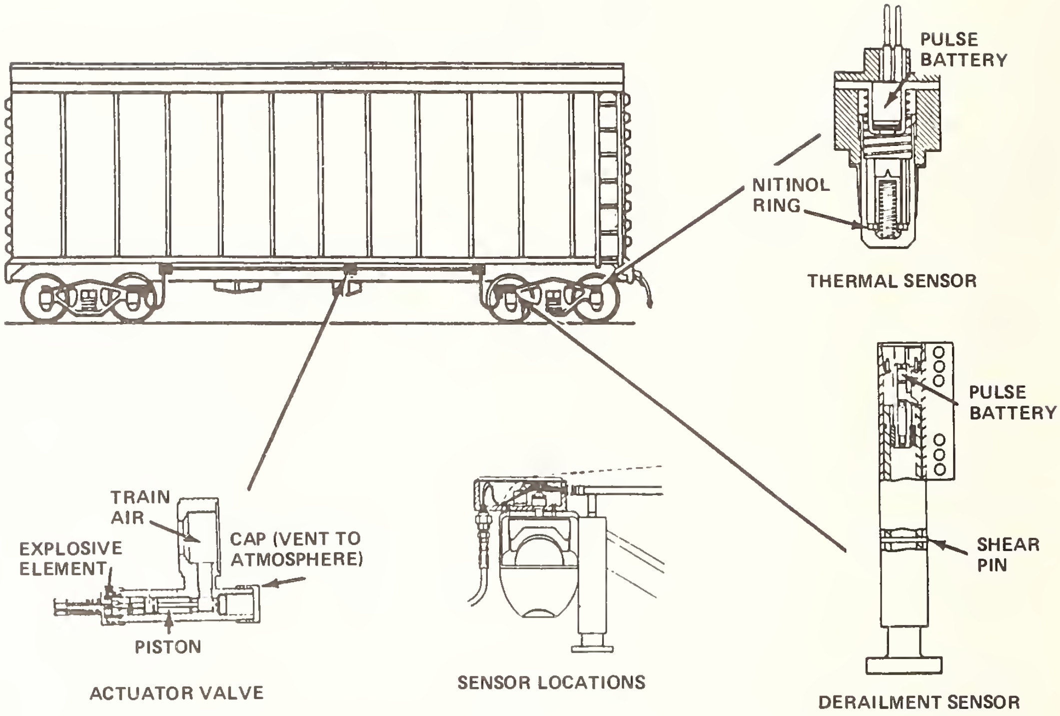

FIGURE 1 D.O.T. SYSTEM FOR TRAIN ACCIDENT REDUCTION (S.T.A.R.)

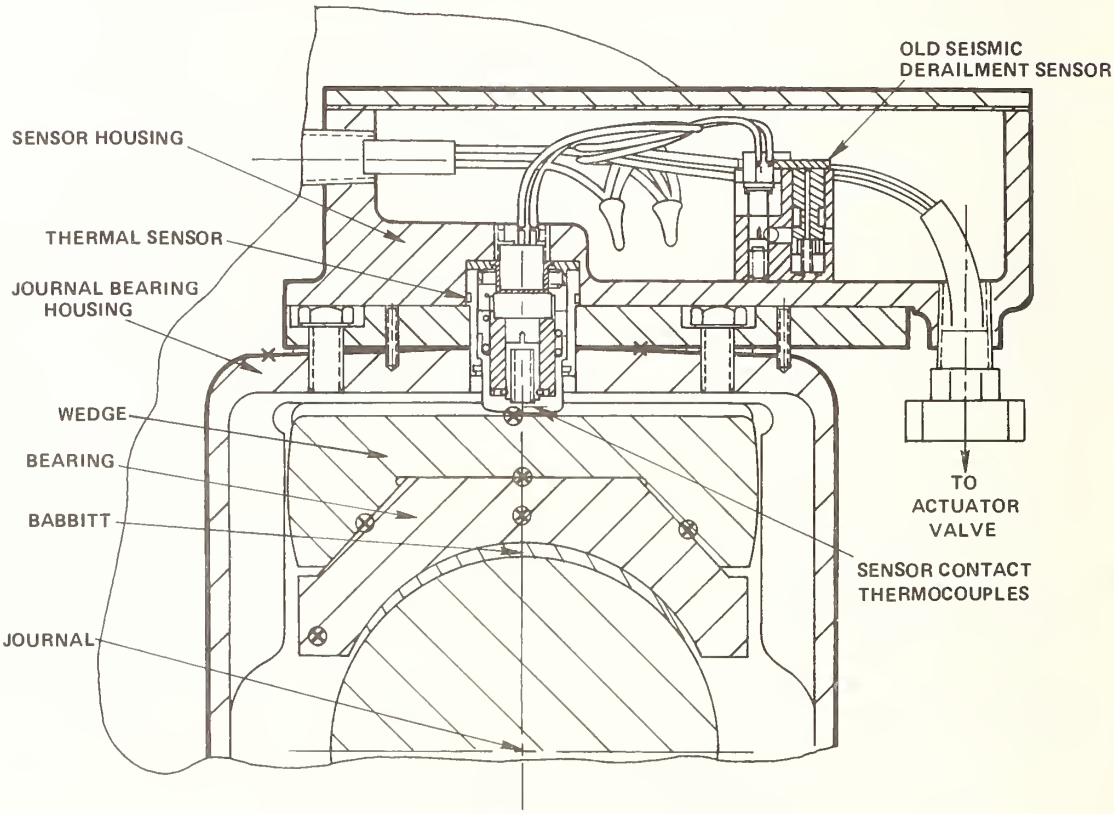

FIGURE 2 SENSOR ASSEMBLY ON JOURNAL HOUSING

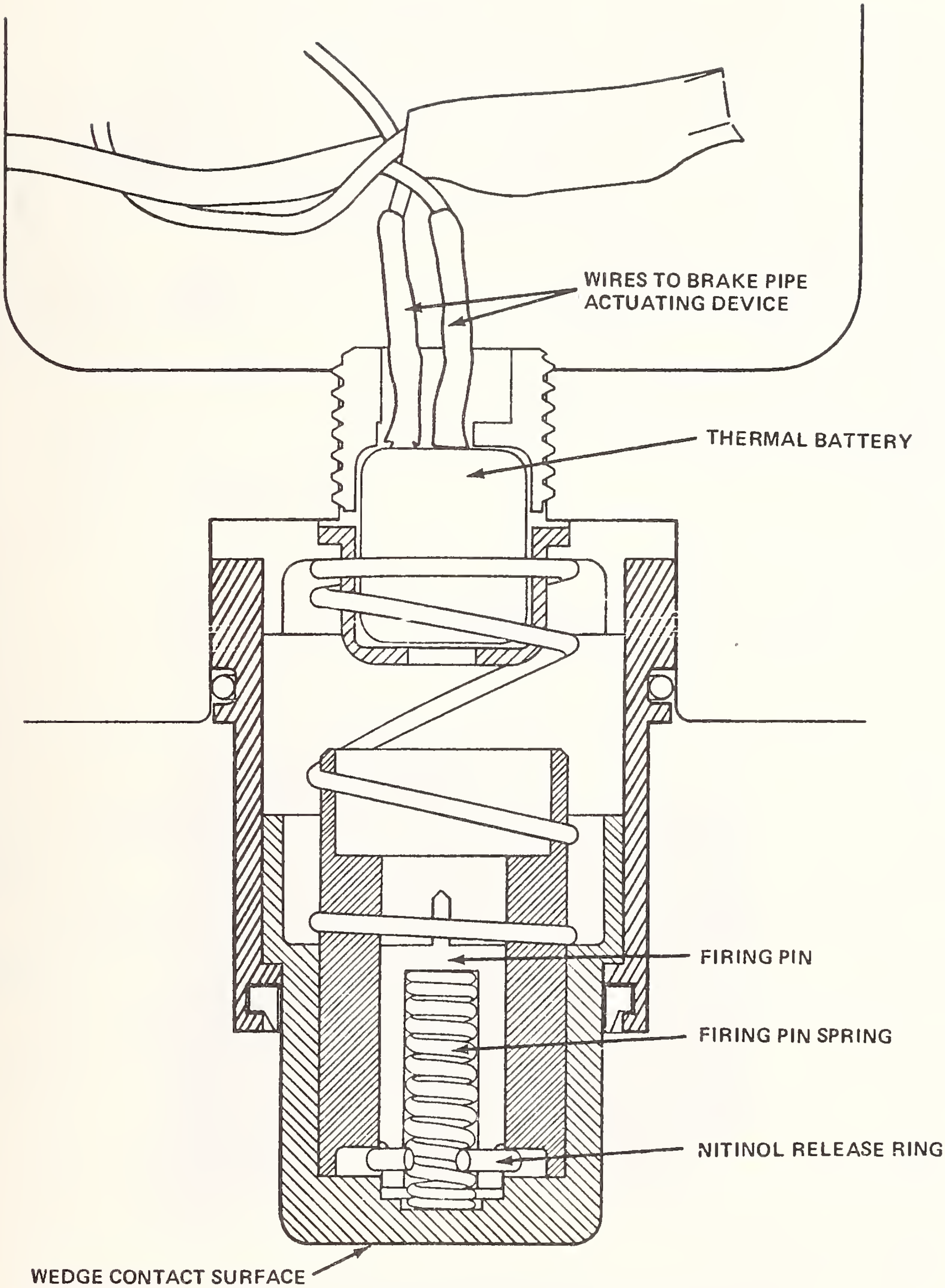

FIGURE 3 JOURNAL BEARING THERMAL SENSOR

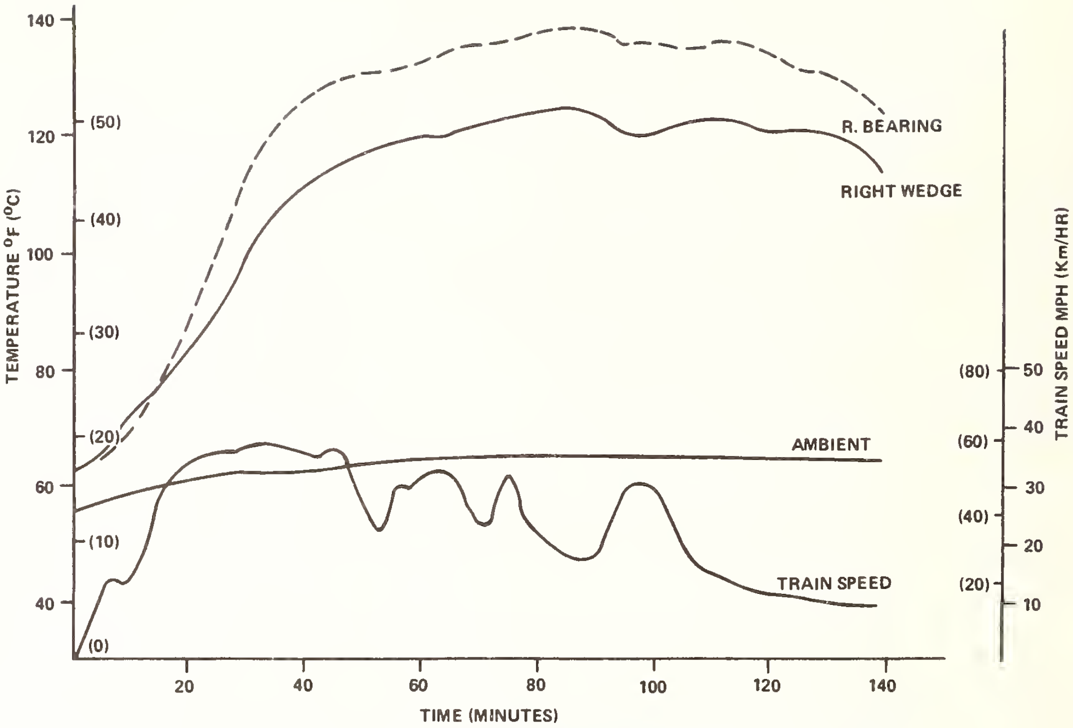

FIGURE 4 OVER-THE-ROAD, TEMPERATURE MAXIMUMS, FULL CAR, JULY 1975, RIGHT

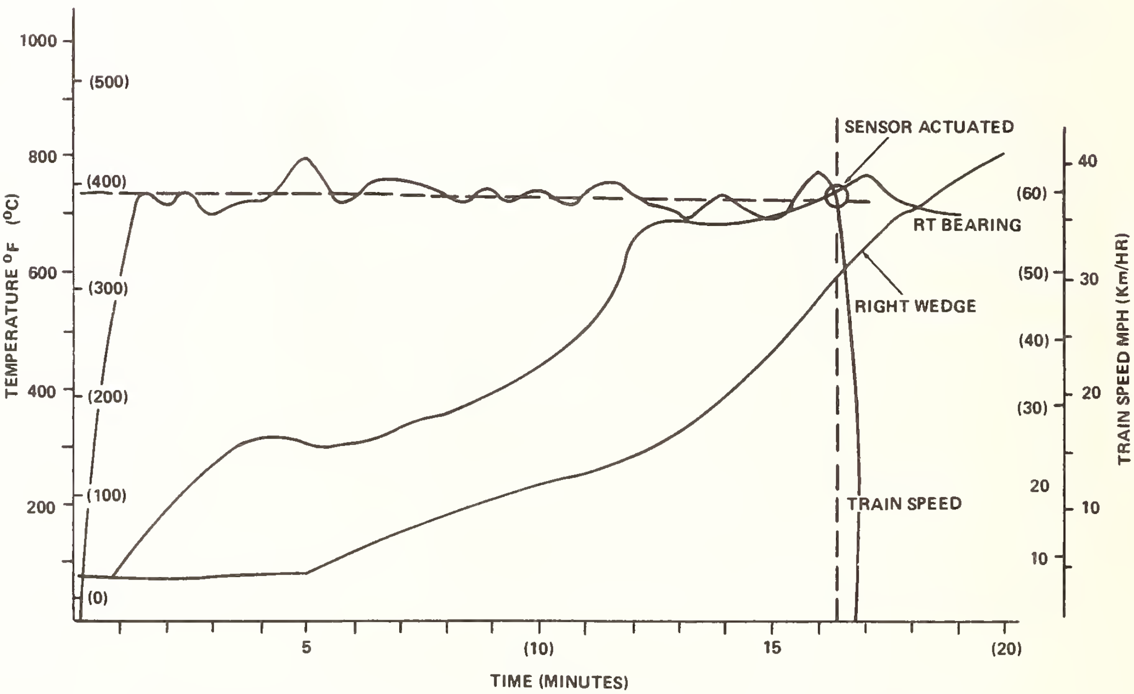

FIGURE 5 HOT BOX TEMPERATURES AND TRAIN SPEED VS TIME, JULY 1975, RIGHT BEARING

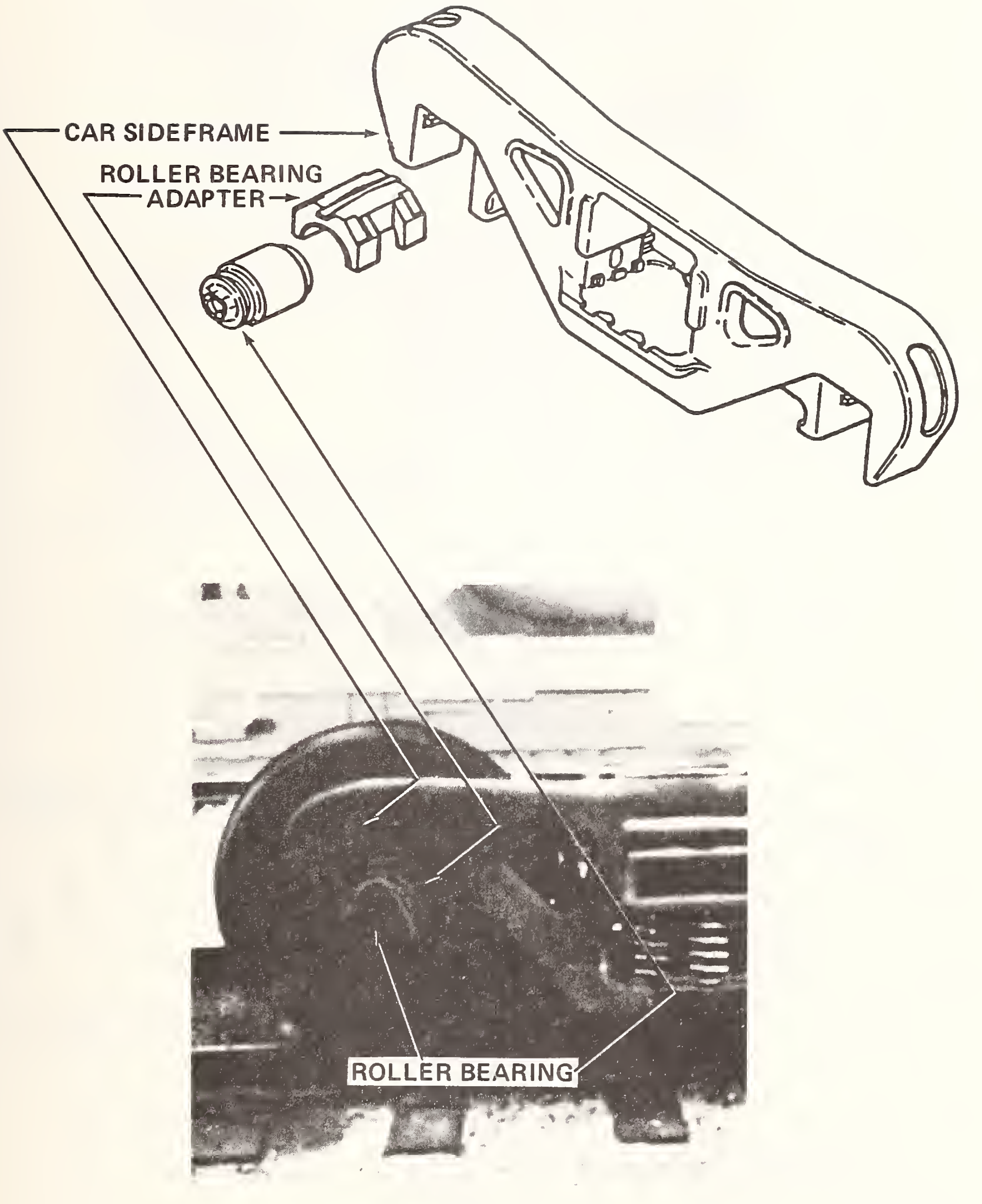

FIGURE 6 ROLLER BEARING AND BEARING ADAPTER

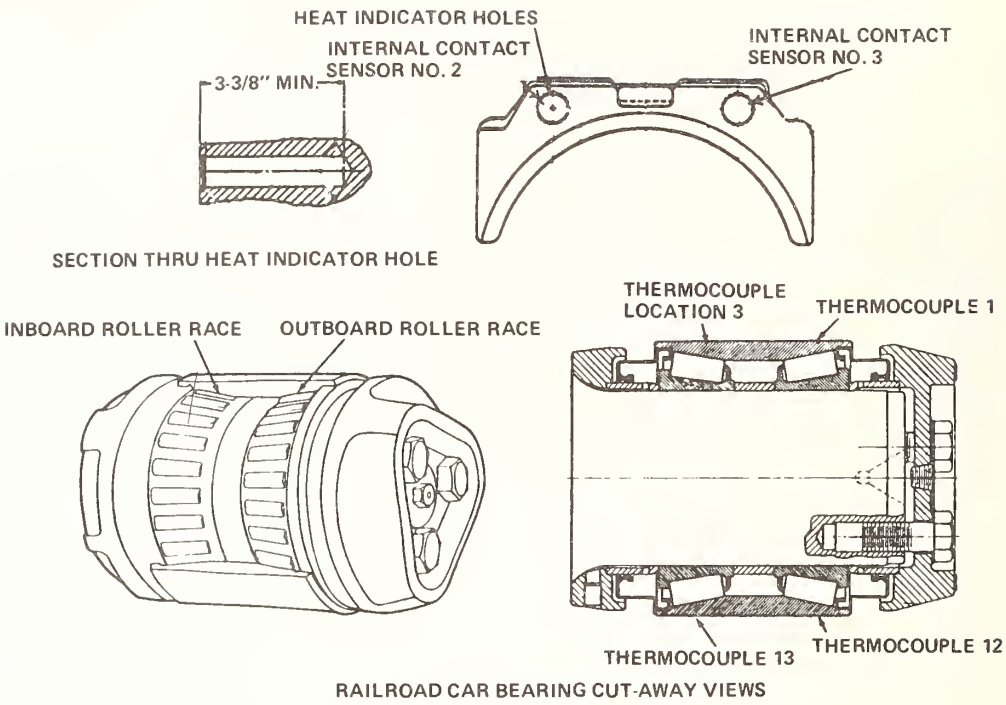

FIGURE 7 HOLES IN BEARING ADAPTER FOR INTERNAL CONTACT THERMAL SENSORS

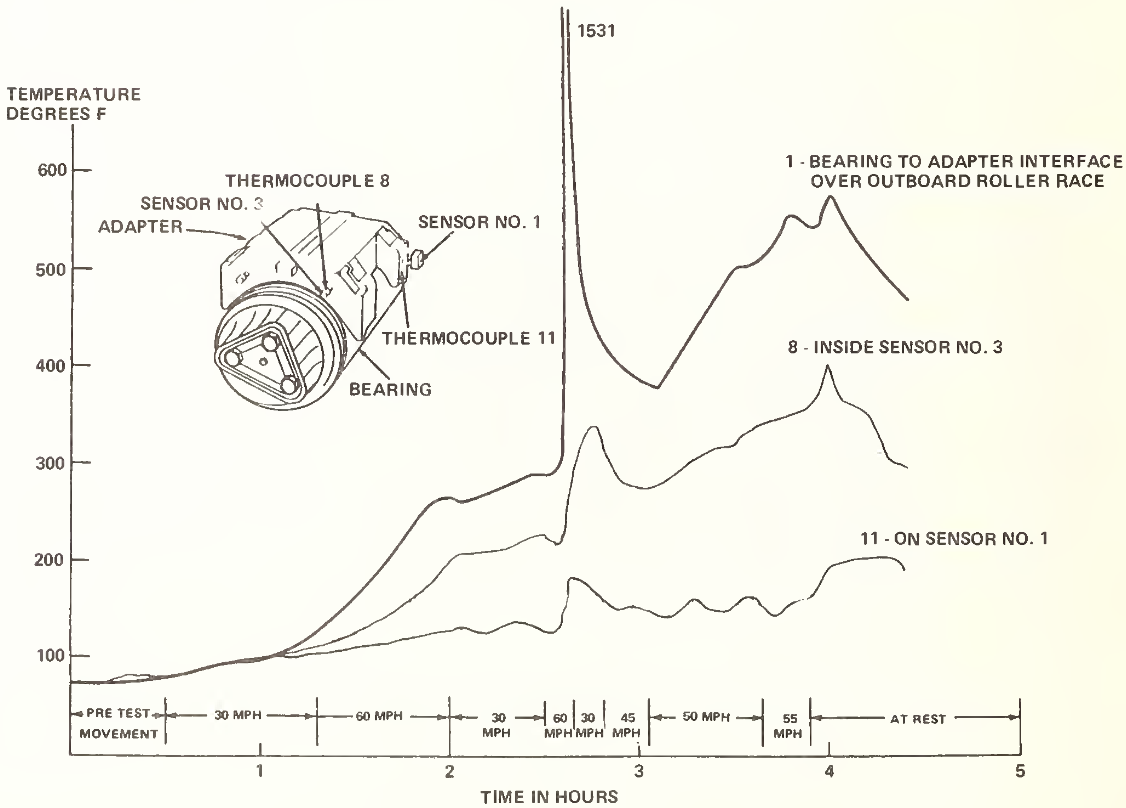

FIGURE 8 OVERHEATING RAILROAD CAR BEARING TEMPERATURES THERMOCOUPLES

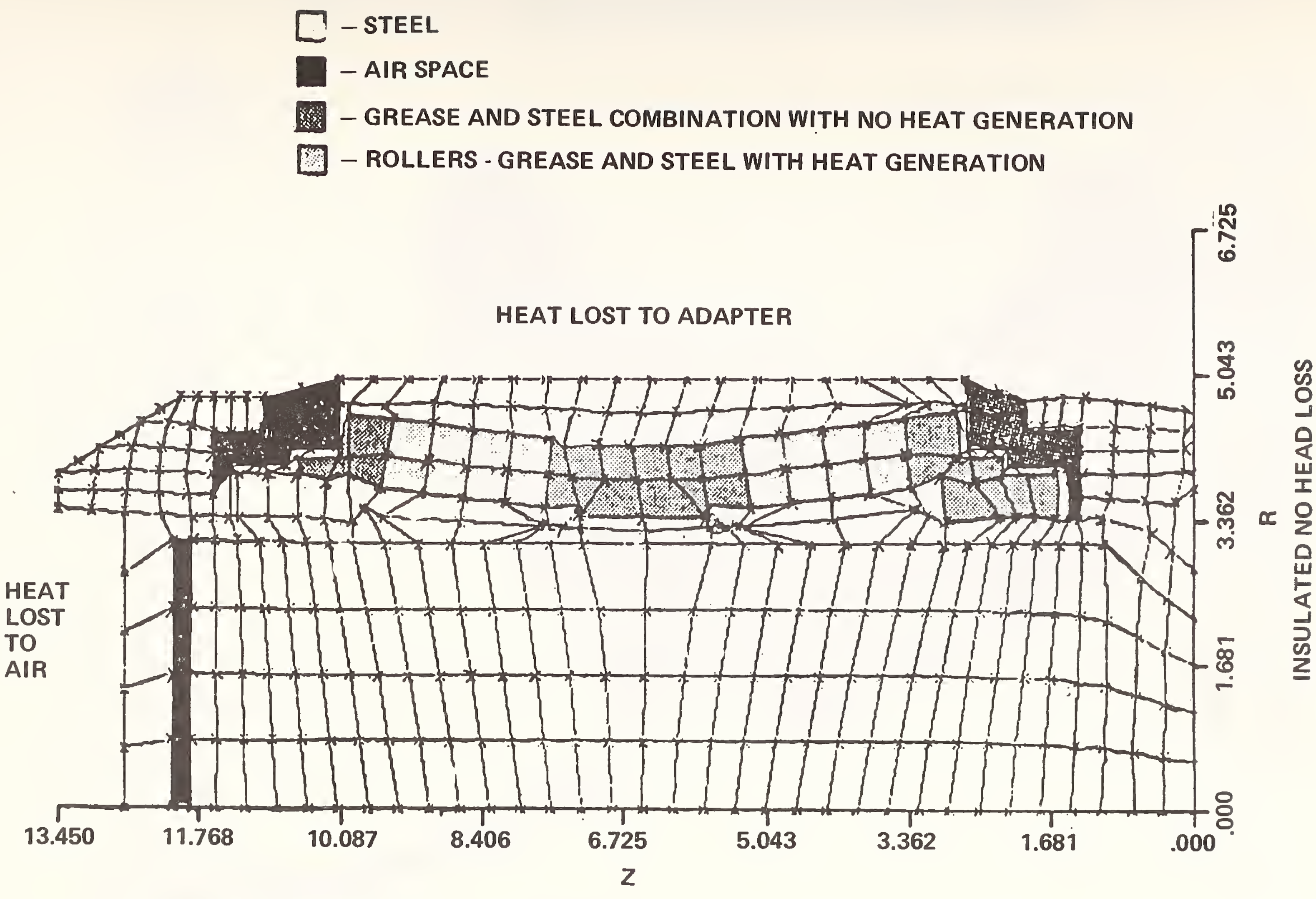

FIGURE 9 BEARING NODAL POINTS AND MATERIAL ASSUMPTIONS

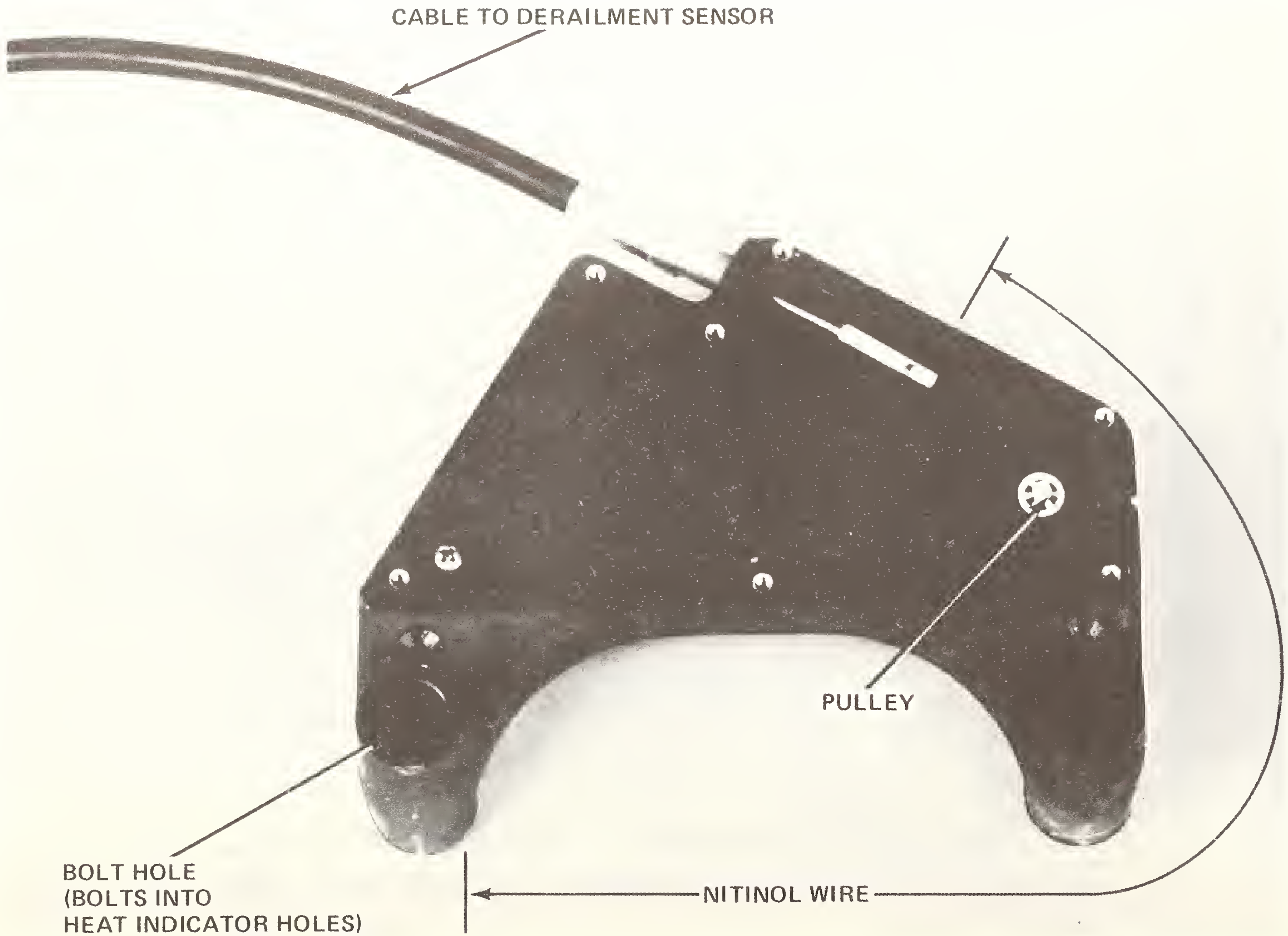

FIGURE 10 ROLLER BEARING THERMAL SENSOR

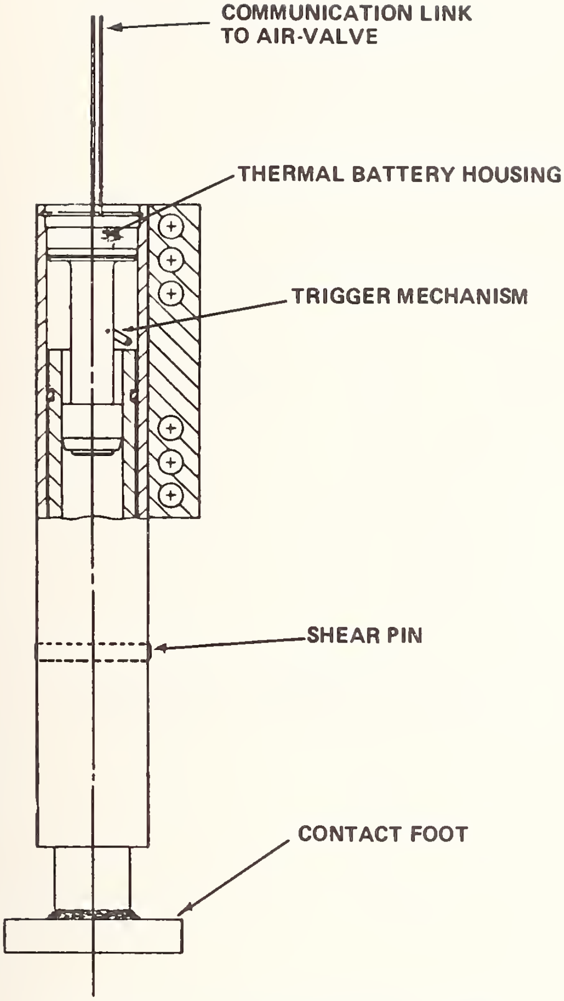

FIGURE 11 CONTACT DERAILMENT SENSOR

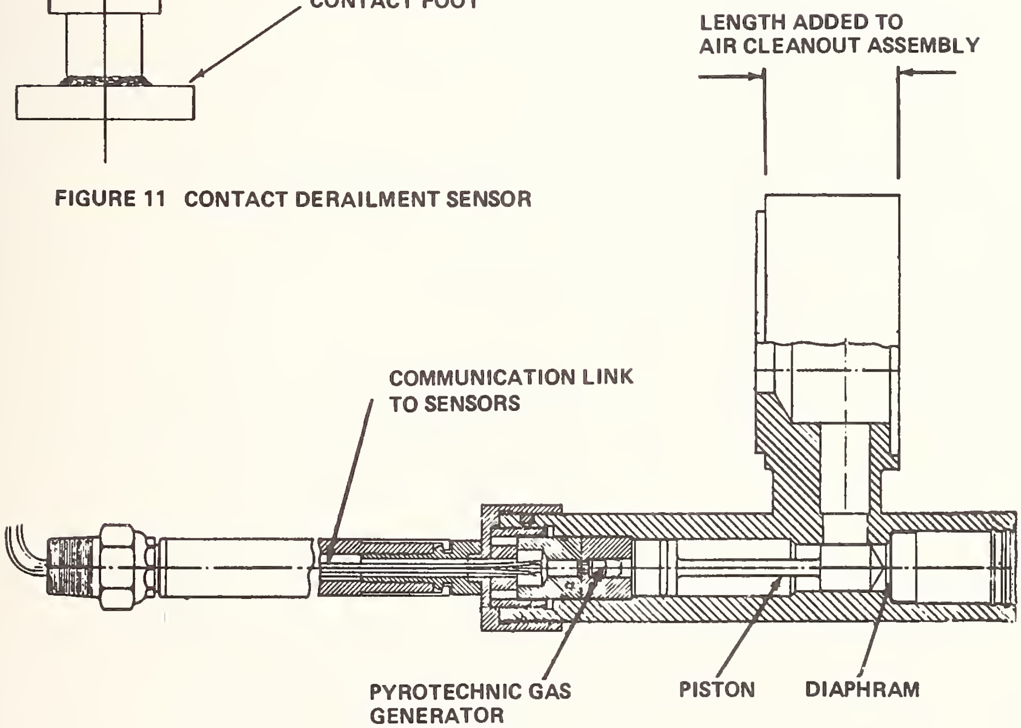

FIGURE 12 AIR VALVE

SESSION III

RESEARCH IN THE RAILROAD INDUSTRY (continued)

CHAIRMAN: HENRY HAHN
ARTECH CORPORATION

FREIGHT CAR RESPONSE ANALYSIS AND TEST EVALUATION MODEL (FRATE)

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Abstract: This paper gives an overview of the computer model, FRATE, which simulates the dynamic responses of a 70-ton boxcar to various track conditions. The FRATE model can, using over-the-road input motions, define the response displacements, accelerations, and loads of the freight car. A recent test utilized an actual 70-ton boxcar on the Vibration Test Unit at the Transportation Test Center, Pueblo, Colorado, to collect data for validating this model. The preliminary results will be presented here.

Key words: Freight car dynamic response; freight car model; hunting simulation; dynamic loads.

The dynamic response of rail vehicles influences safety and cost in the operation of railroads, because it affects the wear of rail car components, rails, track, and roadbed. This extreme dynamic response of a rail vehicle can cause or be a contributing cause of derailments as well as a cause of lading damage. Accordingly, the capability to analytically study vehicle dynamic response can be very useful. Further, if a computer model is used for an analytical study it requires a computer program that is convenient from a user's view point, is economical to use, has demonstrated accuracy and can be modified to simulate rail car configuration or design changes.

The Federal Railroad Administration (FRA) has been investigating the use of dynamic models to determine the responses of rail vehicles to various track conditions. The MITRE Corporation under contract to FRA has developed the FRATE simulation model. FRATE is the acronym for Freight Car Response Analysis and Test Evaluation. In 1976, one version of the model, a trailer-on-flatcar (TOFC), underwent validation testing using the Vertical Shaker System at the Transportation Test Center (TTC) in Pueblo, Colorado. Since then, the model has been modified to analyze boxcar response characteristics. Validation of the boxcar version is currently being conducted.

A recent test used a high cube, 70-ton boxcar to obtain data to validate the latest version of the FRATE model. Testing was conducted in two phases. First a characterization test was performed on each truck to measure its stiffness and damping properties. Secondly, the complete boxcar was tested on the Vibration Test Unit (VTU) at TTC. Four boxcar configurations were tested on the VTU: the boxcar loaded with cans in corrugated containers with and without friction snubbers in the trucks; the empty boxcar without snubbers; and the boxcar loaded with cans in stretch wrap, with friction snubbers. Resonant frequencies were found for the boxcar suspension system, the boxcar body torsion, the boxcar body bending, and the cargo in lateral and vertical motions. Determinations were made of the responses of the vehicle system to two track-profile simulations and a body-hunting simulation. Two shipping materials were used to ascertain if there was a difference in input from the cargo (resonances) to the boxcar during transit. Michigan State University is analyzing the cargo data, and results should be available in January 1982. Validation of the boxcar version of FRATE should be completed by July 1982.

Stationary rail vehicles weighing up to 160 tons can be subjected to controlled vertical and lateral vibration inputs at the wheels on the VTU. These vibrations create the dynamic effects of perturbed track. Excitation is supplied by 12 hydraulic actuators: 8 vertical (one at each wheel) and 4 lateral (one at each axle). The shaker system is computer controlled. The data acquision system has a recording and processing capability with on-line and off-line data display. Currently, the VTU is undergoing modifications to permit easier and less time-consuming preparations for tests.

FRATE Model

The FRATE computer program calculates the dynamic response of freight cars to track inputs. The basic program is nonlinear, written in FORTRAN (currently for Control Data Corporation computers), with solution in the time domain by numerical integration methods. The two versions are FRATXI, for the analysis of boxcars, and FRATFI, for the analysis of trailer-on-flatcar (TOFC). The program permits the user to perform time history analyses; various track profile variations can be simulated and the resulting freight car responses are calculated

The FRATE program can be used to study the dynamics of various freight car configurations and track geometry conditions. Loads on both track and freight can be determined and safety margins can be evaluated, thus providing a means for defining and controlling operational safety margins. The FRATE program can also be used to evaluate innovations to freight cars.

The computer program FRATE is a synthesis of models and analyses by M. Healy, P. Ahlbeck, and P. Abbott. The truck model, unique in this MITRE version, contains the following additional features.

1. Truck suspension in roll has been changed from bilinear to trilinear, whereby the center plate lift is simulated.

2. Friction snubbers and hydraulic snubbers have been added.

3. A lading model has been added.

4. The wheel and rail lateral stiffness have been made bilateral to simulate conditions with and without flange contact.

In the FRATXI version, the boxcar has a rigid body mounted on two trucks and contains rigid and compliant lading. The combination boxcar and lading are symmetrical along the longitudinal and lateral axes. The center of gravity of the combined boxcar and lading is located midway between the trucks and is on the longitudinal centerline. The boxcar structure in FRATXI is a six-sided hollow box with all but the bottom of equal thickness. The lading is divided into five parts, one rigid and four spring mounted.

Validation Methods

In the validation of the model, there are two categories of comparisons which can be made: one is to make a direct comparison of the experimental and analytical response to a known in put at the rail; the other is to compare the dynamic characteristics of resonant frequencies, deflection shapes at resonance, and amplification at resonance. The test methods used are generally referred to as response testing and modal testing. Each comparison has its advantages. The response comparison gives a more direct measurement of the predictive accuracy of the computer model. However, to be done properly, a wide range of possible track variations must be input and responses over a representative portion of the freight car must be compared. The problem is compounded by the nonlinear characteristics of the freight car; for example, it would be necessary to see the effects of the amplitude of the rail variation and the effects of freight car loading.

Comparison of modal characteristics is more tractable in that there are five or six resonances that need to be compared in the typical freight car. These are low center roll, high center roll, yaw, vertical bounce and pitch. In some freight cars, it may also be necessary to measure the additional resonances of body torsion and body bending. To identify and measure the characteristics of these resonances in the laboratory is a reasonable project. Evaluation of the effects of nonlinearities can also be accomplished.

Validation through comparison of modal characteristics also has the advantage of providing clues as to the causes and cures for differences between analytic and experimental results.

Modal testing of large complex structures has been used for many years for the validation of math models. Following are the four basic excitation methods which have been used.

1. Single-point sine using frequency sweeps to identify modes and dewlls to measure modal characteristics

2. Multiple-point sine, sweep and dwell

3. Single-point random input

4. Transient input

All of these testing methods are based on linear undamped systems and become ineffective with damped or nonlinear systems. Of the above four methods, the multi-point sine, sweep/dwell test has probably had the widest use, has proven reliability and is least susceptible to modal test problems due to damping and nonlinear properties.

There are computer aided techniques that have been used with single-point random and transient inputs with outstanding success as to the number of modes identified, accuracy and cost effectiveness. However, the successful applications have all been with linear, lightly damped systems. To attempt to use these methods for a freight car vibration test is likely to result in an open ended development program.

In the validation of FRATXI, both the vibration characteristics and the dynamic response parameters with be used. The vibration criteria will determine whether the general, overall dynamic characteristics of the vehicle have been accurately simulated. This is important since one function of the validated computer program-simulation could be to investigate the effects of modifying parts within the vehicle. Examples are: side bearing changes, or the addition of or changes to hydraulic snubbers, or changes in the spring rate of the truck primary suspension system. It is necessary that the physical part and its mathematical counterpart be clearly identifiable and properly modeled so that not only is the basic model valid but also the accuracy of the simulation is retained with moderate changes to components. The dynamic response criteria will provide a more direct measure of the accuracy of the analysis.

The dynamic characteristics of the boxcar model to be used in the validation include resonant frequencies, deflection shape at resonance, damping associated with each resonance, and the dynamic response at selected points on the boxcar.

The vibration testing was used to obtain definitions of the vibration characteristics of the test configuration that could be used for comparison with the analytical model. Resonance testing and response testing were performed.

Pretest analyses were performed duplicating the planned test. These analyses served two purposes. First, they were a guide in the performance of the testing: that is how to test and what to expect in results. Second, the pretest analysis results were used in the initial comparison between test and analysis results in the validation process. This initial comparison was made while the test was in progress and as test results become available. This was an aid in knowing that all resonances have been identified and that sufficient data had been obtained for the validation process.

The resonant frequencies determined in the pretest analysis are listed in Table 1. The frequency tolerances shown are the difference between test results and analysis results which are not to be exceeded.

Preliminary Test Results

Both trucks of the test boxcar were tested under a variety of test conditions to show the effects of friction snubbers and freight car gross weight. The three types of tests performed included: vertical loadings to obtain vertical spring rates and snubber forces; roll loadings to obtain roll spring/snubber values; and lateral loadings to obtain the lateral spring/snubber values. The frequency of load applications was 0.1 to 0.2 Hertz, slow enough not to introduce any dynamic effects.

From plots generated, it was determined that the truck spring rate was 21,000 pounds per inch per side. The snubber friction force ranged from 2,250 to 5,250 pounds, as the vertical load varied from empty to 75-tons. These numbers were very close to the expected results.

The test resonant frequencies of primary interest were expected to fall in the 0.5 to 10.0 Hertz range, with those of secondary interest out to 35 Hz. It was also expected that maximum responses in the 0.5 - 35 Hz range would not exceed 2.0g (0-peak).

Two characteristics of the acceleration data made measurement difficult. First, the low frequencies -- 0.5 Hz g levels -- were on the order of 0.5g's. Second, higher frequency transients were superimposed to on the low frequency excitations.

A preliminary analysis of the vibration test data has been completed. See Table 2.

Configuration 1A is the boxcar loaded with cans in cartons and with active friction snubbers. Configuration 1B is loaded the same as 1A

TABLE I

VALIDATION CRITERIA - VIBRATION CHARACTERISTICS

Resonance Descriptions	Frequency (pretest analysis) Hertz	Freq. Tolerances
low center roll	0.7*	± 15%
high center roll	1.6	± 10%
yaw	1.7	± 10%
vertical bounce	2.2	± 10%
pitch	2.9	± 10%
carbody torsion	---	---
carbody bending	---	---
lading - lateral	5.0	± 10%
- vertical	9.5	± 10%

*Where resonances are non-linear with respect to amplitudes, the analysis will duplicate the non-linear effect.

TABLE 2

TEST FREQUENCY SUMMARY - 70 TON BOXCAR VIBRATION

MODE	PREDICTION	CONFIG. 1A	CONFIG. 1B	CONFIG. 2	CONFIG. 3
1st Roll	0.7	.62-.95(1)	.7 (.67 decay)	.82-.95	.68, .88
2nd Roll	1.6	N.A.	2.6	2.9-3.1	3.8
Yaw	1.7	N.A.	1.7	2.6	2.4
Bounce	2.2	2.6	2.05	3.8	2.4
Pitch	2.9	3.75	2.77	4.34	3.8
Body Torsion	---	12.6	12.4	13.5	12.7
Body Bending	---	16	--	17.6	15.8 17.2
Lading Lat.	5.0	1.9-2.6(1)	3.0	--	2.3-2.4
Lading Vert.	9.5	7.5-8.1, 17-18	8.3	--	8-8.5

(1) amplitude dependent.

N.A. - not available (snubbers remained locked).

with inactive friction snubbers. Configuration 2 is an empty boxcar with inactive friction snubbers. Configuration 3 is the boxcar loaded with cans in stretch wrap, with active friction snubbers.

In comparing the predicted values with the measured values, it is seen that the first roll mode is close for all configurations. For the second roll mode this is not the case. Part of the problem may be that the model predictions were not made for an empty boxcar (Configuration 2); but this does not explain the variation between predicted and configuration 3 values.

For most all modes there is very little difference between the values for configuration 1A and 3 which seems to indicate that the packing material for the cans had no significant effect on the input of the cargo to the boxcar.

The yaw and pitch modes show a difference between the boxcar loaded with and without snubbers. This indicates that the effect of friction snubbers is to raise the resonant frequency in the yaw and pitch modes.

The final analysis should be completed by mid - 1982.

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THE ACCIDENT PERFORMANCE OF TANK CAR SAFEGUARDS

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The mission of the National Transportation Safety Board is to improve transportation safety. This is done primarily by determining the probable cause of accidents through direct investigations and public hearings, and secondarily through staff review and analysis of accident information, through evaluations of operations, effectiveness, and performance of other agencies, through special studies and safety investigations, and through published recommendations and reports.

BACKGROUND

Until the late 1950's liquefied petroleum gases and other hazardous materials were transported by rail in 11,000-gallon nominal capacity tank cars, now known as DOT Specification 105A tank cars. In 1958, however, specification changes were allowed which permitted the use of DOT 112A/114A tank cars having a nominal capacity of 33,000 gallons.

Between 1968 and 1979 DOT 112/114A tank cars were involved in numerous serious accidents. In many of the accidents, couplers disengaged and overrode each other, puncturing the "head," or ends of tank cars. Punctured tank cars carrying liquefied petroleum gas (LPG) exploded and fire erupted, resulting in death, injury and extensive property damage. In some cases, the intense heat from blazing tank cars caused other tank cars to erupt in "Boiling Liquid-Expanding Vapor Explosions" (BLEVE's). Sometimes, sections of steel tank, weighing several tons, were propelled distances up to one-half mile. Punctures of tank cars containing other hazardous chemicals resulted in releases of toxic gases which were carried over surrounding areas by wind currents.

In 1969, following a disastrous accident at Laurel, Mississippi, the Safety Board first called attention to the serious problem of head punctures of tank cars by overriding couplers. As a result of that accident, the Safety Board made recommendations which addressed coupler design and called for a program to develop technical improvements to hazardous materials tank cars. Following the Laurel accident, the Association of American Railroads, in cooperation with the Railway Progress Institute, formed a tank car research committee which developed a research and test project. As a result, in 1971 the committee recommended installation of tank car headshields and in 1972 and 1973, recommended installation of "shelf couplers" to prevent overriding.

In 1974 the Department of Transportation issued regulations requiring that all DOT 112/114 hazardous material tank cars be equipped with headshields by December 1977. The regulations were challenged in court by some shippers, tank car owners and lessors, and though the challenge was unsuccessful, the retrofit program was delayed.

Federal agencies and the railroad and tank car industries were slow to act to improve the situation and after a series of catastrophic tank car accidents in late 1977 and early 1978, the Safety Board held a public hearing in April 1978 into derailments and their consequences for the carriage of hazardous materials by rail. As a result of the hearing, the Safety Board recommended that the Federal Railroad Administration accelerate Federal rulemaking actions to require the installation of head shields and shelf couplers on U.S. Department of Transportation specifications 112A and 114A cars by December 25, 1978. The Board also recommended that thermal protection be installed on tank cars carrying volatile products as soon as possible. The recommendations resulted in an accelerated safety retrofit program for 112A and 114A tank cars by the DOT. The Safety Board's effort to improve the safety of hazardous materials tank cars was later expanded to include DOT specification 105A tank cars in a recommendation that resulted from the Board's investigation of an accident at Youngstown, Florida, on February 26, 1978. The Safety Board, concerned about the DOT's response to its earlier recommendation, recommended in a September 1979 progress report that the shelf coupler retrofit for 105A tank cars be completed by December 25, 1980.

As a result of these Safety Board efforts to improve hazardous materials tank car safety, it was desirable that the Board review the effects of shelf couplers, head shields, and thermal protection under actual accident conditions in comparision to tank cars without this protection to determine the impact of the new measures and if there were areas where further safety improvements were required. Therefore, the Safety Board examined the sequence of events in the derailment of a chemical train near Paxton, Texas, on September 8, 1979.

This special investigation examined the behavior of each derailed tank car that was equipped with shelf couplers, head shields, and thermal coating and compared this behavior with that of cars not so equipped. This report compared the actual safety results of the new safeguards with the stated objectives for which they were mandated.

The reconstruction of a railroad accident by developing the sequence of events affecting each car before, during, and after the accident is not a commonly used procedure, but it is essential to revealing the point at which the events exceeded the normal and safe operating limits. In this special investigation, the analysis was limited to an examination of the sequence of events from the start to the end of the derailment--a period of about 80 seconds--and of the ensuing fire and subsequent events. The Safety Board's task was difficult since the reconstruction was not undertaken until the wreckage has been cleared. It was possible only because of the participation and contributions of the FRA, the Association of American Railroads, the Railway Progress Institute, the Southern Pacific Transportation Company, and the tank car owners and shippers whose cars were involved in the accident. The Safety Board analysis is based on the reconstruction of events--the first such analysis of a chemical freight train accident in the railroad industry.

THE ACCIDENT

About 4:22 p.m., c.d.t., on September 8, 1979, 33 cars and 2 locomotive units of Southern Pacific Transportation Company freight train (chemical) Extra 1291 East SRASK derailed in a lightly populated area near Milepost 183, about 3 miles east of Paxton, Texas. The train consisted of 53 cars and 3 locomotive units. The train had descended a 1.6-percent grade, with locomotive braking in the dynamic mode, and was beginning to ascend a 1.6 percent grade at about 30 mph when the cars derailed on a 2° 20' curve on a single track. The track was located on a 15-foot-high fill embankment.

Couplers

The intent of existing tank car regulations (49 CFR 179.105) is to prevent vertical separation through the use of shelf couplers, to prevent end punctures by the use of head shields, and to reduce the potential for catastrophic effects of fire by requiring thermal coating.

Couplers on tank cars have been the subject of Federal regulations since 1971. Since that date, the FRA has required that tank cars built or rebuilt be equipped with F-type couplers. The E-type coupler, when coupled to another E-type coupler or to an F-type coupler, would be expected to disengage along the vertical plane during compression (run-in or buff forces). Vertical disengagement of E-type couplers would not be expected when they are coupled to E t/b-type or F t/b-type shelf couplers. F-type couplers are restrained along the vertical plane when coupled to other F-type couplers, to E t/b-type shelf couplers, and to F t/b-type shelf couplers. Cars equipped with E t/b-type and F t/b-type shelf couplers also would be expected to remain coupled.

Title 49 CFR 179.105-6 requires coupler vertical restraint systems to lessen the possibility of tank punctures by constraining vertical disengagements of couplers. During the initial run-in, when compressive forces are greatest, cars with shelf couplers should remain coupled, and vertical separation (which can result in head puncture) should not occur.

Four 112 tank cars in the derailment were equipped with E t/b-type shelf couplers. Examination of the heads of the cars for coupler penetration and dents showed that all four of the 112 tank cars remained coupled and undamaged during the initial run-in; vertical separation did not occur during this phase of the collision. However, five of the eight couplers disconnected later in the accident sequence.

The five 105 tank cars involved in the derailment were equipped with E- and F-type couplers. Title 49 CFR 179.105-6 does not currently require top-and-bottom shelf couplers on this type of equipment. Five of the 10 couplers should have provided protection by remaining coupled. Examination of the derailment data shows that the cars equipped with top-and-bottom or F- to F-type couplers did not separate along the vertical plane, whereas four of the five couplers that were expected to separate vertically did; one coupler could not be evaluated because portions of the tank head were destroyed by fire.

The 15 specification 111 tank cars involved in the derailment were all equipped with E- or F-type couplers, except for car No. 26 which had an F t/b-type shelf coupler on its trailing end. It was expected that 8 car ends would remain coupled, and that 22 car ends would uncouple under compressive forces. The 8 car ends expected to remain coupled did remain coupled during the initial collision; 21 of the 22 car ends expected to uncouple did uncouple during the initial collision.

By comparing the coupler performance of the cars which were equipped with protective couplers with those which were not so equipped, the advantages of top-and-bottom shelf couplers or F- to F-type couplers become apparent. These data indicate that all of the car ends with protective couplers stayed together during the initial run-in. On these cars, there were no punctures or dents attributable to overriding coupler strikes on tank heads. In contrast, of the 27 car ends connected with couplers in which vertical separation would be expected, 96 percent did, in fact uncouple. None of the cars joined by protective couplers suffered a head puncture attributable to coupler strike. There were 4 head punctures and 9 heads dented in the 27 ends without protective couplers. Clearly, protective couplers provided significant protection against end penetration during initial run-in.

Head Shields

A major area of concern to the Safety Board has been the protection of tank heads from puncture. Tank heads are punctured when couplers disengage vertically and strike the tank head or when the tank head strikes or is struck by other objects during a collision. Shielding the tank head is achieved by attaching a metal plate to the car or tank structure. This shielding is required by 29 CFR 179.105.5.

The intent of the regulatory requirements for tank head puncture resistance is to provide tank cars with the capability of withstanding specified impacts without loss of lading.

In this accident, the 26 car ends that uncoupled during the run-in were susceptible to end coupler strike. There were 4 punctures and 9 dents or about 1.5 punctures and 4 strikes for every 10 car ends involved. Only four cars (8 ends) in the derailment were equipped with head shields, and none of these cars uncoupled, so the utility of the head shields as protection against coupler strikes was not tested in the initial run-in. Tank car No. 30 was the only car of this group exposed to head damage as a result of subsequent strikes during the derailment dynamics, and it resisted probable head puncture.

The 26 specifications 105 and 111 car ends that uncoupled in the initial phase of the accident, as noted before, resulted in 4 head punctures and 9 heads dented. In other terms, 7 unprotected cars in 10 suffered a head strike and 3 in 10 were punctured. If head shields and shelf couplers had been in place, the likelihood of punctures would have been reduced. The nine dented heads on unprotected cars clearly demonstrated a potential for penetration.

Thermal Protection

The intention of the requirements of 49 CFR 179.105-4 for thermal protection is to prevent the release of any of the car's contents (except release through the safety valve) when subjected to (1) a pool fire for 100 minutes, and (2) a torch fire for 30 minutes.

In this derailment, two 112 cars with thermal protection, Nos. 7 and 30, were exposed to fire. There was no apparent release from car No. 30. A large burn hole developed in car No. 7, through which the cargo was released, but the car did not rupture violently. The cause of the failure of car No. 7 is not known.

Some 111 and 105 cars are insulated to reduce the effects of solar heating. Although this insulation was not intended to protect these cars in a fire environment (as are installations of thermal protection), tests made by the FRA have shown this insulation protects against a 2,200° F fire for 20 minutes. In contrast, the tests showed that a bare tank will survive the same environment for only 4 minutes.

Twelve of the 24 cars involved in this accident were neither punctured during the run-in nor in the collisions that occurred after the cars left the tracks. Six of these cars were in a fire environment that permitted an evaluation of the survivability of these cars. All three of the exposed cars, Nos. 9, 13, and 30, had some form of thermal protection or insulation; one of the 3 cars that failed, No. 8, had neither and ruptured violently. The other two cars that failed, Nos. 6 and 14, had 4 inches of fiberglass insulation. Car No. 6 was breached by fire impingement, or may have had an undetermined head puncture. Car No. 14 ruptured violently. Although not conclusive, the evidence indicates that thermal protection and insulation delayed violent ruptures.

Car Damage and Loss of Product

The reconstruction of the events of this accident demonstrated that although the requirements of 49 CFR 179.105 reduce the potential for punctures during run-in and decrease the likelihood of BLEVEs, car damage and loss of product is still a problem. The investigation showed that (a) 18, or 82 percent, of the breaches resulted from collisions that occurred after the cars left the tracks, (b) 78 percent of the breaches resulted from damage to the top fittings or bottom outlets, and (c) large tears in the sides of tank cars resulted from collisions with boxcars or other tank cars. While all of these causes of product loss may not be controllable at this time, the release of product from 12 tank cars (50 percent) through top fittings or lower outlet fittings can be precluded by corrective action. Safety benefits can be expected through design protection of the lower outlet leg area. The technological feasibility of protection to the fitting is proven. It has been established that "skids," protective structures around the valve, provide protection to lower outlet valves. The outlet (lower) valves can also be protected by recessing them above the shear line of the tank structure.

One tank car manufacturer now offers tank cars equipped with an "internal ball valve" that is located inside of the tank. A customer of this car manufacturer orders this valve on all new tank cars. While top fittings have not received comparable attention, the same technology could be applicable to top fitting designs. Protection needs to be applied expeditiously to the existing fleet of hazardous materials tank cars to limit the potential for product loss.

RECOMMENDATIONS
OF THE
NATIONAL TRANSPORTATION SAFETY BOARD

As a result of the special investigation, the National Transportation Safety Board recommended that the U. S. Department of Transportation:

Extend the requirements for top-and-bottom shelf couplers to all tank cars used to transport hazardous materials.

Extend the requirements of 49 CFR 179.105, including those for head shields and thermal protection, to include new and rebuilt specification 105 tank cars.

Examine specialty products and Class A poisons which are shipped in Type 111 tank cars to determine if the toxicity hazard is sufficient to justify the protection afforded by 49 CFR 179.105.

Take immediate steps to cause the modification of both new and existing tank cars so that damage to the top fittings and bottom outlet valves is minimized in train accidents.

Cause data to be collected on tank car derailment behavior to identify breach mechanisms, analyze these mechanisms, identify control methods, and incorporate findings in new car construction.

Conduct tests of tank cars in freight train derailments to determine if the severity of collision damage can be reduced by tank car placement in trains. Identify and test countermeasures.

Require crashworthiness testing of new hazardous materials tank car designs to resist breaching in the derailment environment.

RAIL TANK CAR IMPACTS

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Abstract: This paper describes the Federal Railroad Administration's research efforts to mitigate the consequences of impacts to rail tank cars.

Key words: Metallurgy; puncture prevention; structural integrity; non-destructive testing.

INTRODUCTION

In recent years there has been a number of railroad accidents involving hazardous material tank cars. Many of these cars were Department of Transportation specification 112 or 114 tank cars, and the initial research and regulatory efforts of the Federal Railroad Administration (FRA) were directed primarily at those specification tank cars. However, since tank cars of other specifications (e.g. 105 and 111) have also been involved in railroad accidents, current FRA research and regulatory efforts concern these other specification tank cars.

In 1970, FRA initiated a research program to reduce the frequency and severity of hazardous material accidents on railroads. For the two areas of research involving tank car impacts, FRA is trying to reduce the frequency of the release of hazardous materials in impact situations and developing techniques for assessing the structural integrity of rail cars subjected to impacts. This paper describes the past and present research efforts in these two areas.

HAZARDOUS MATERIAL RELEASE PREVENTION

The release of hazardous materials in railroad accidents can produce catastrophic consequences. In several incidents a tank car carrying a flammable material has been punctured, fire engulfed adjacent cars, and these adjacent cars ruptured violently causing deaths, injuries, and property damage (1-9). There have been other incidents where a tank car carrying toxic material was punctured and the escaping vapor killed and caused injuries (10-14). In still other incidents, a tank car carrying a flammable material was punctured, the resultant vapor cloud drifted downwind, was ignited, and led to deaths, injuries, and property damage (3, 15-18).

Since 1970, FRA has had an interagency agreement with the National Bureau of Standards concerning the metallurgy of tank car steels. This research has disclosed a wide variation in the fracture toughness of tank car steels. Current research is aimed at developing guidelines for cost effective improvements in tank car steels.

Although improvements in tank car steels would increase the puncture resistance of new cars, they would not solve the problem of puncture resistance for existing cars which have a useful life of 40 years. Accordingly, the FRA has sponsored research and analysis that could be incorporated in existing cars.

An analysis of the FRA accident data from 1969-1974 for specification 112 and 114 tank cars has shown that approximately 84 percent of the tank punctures occurred at the head ends of the cars and 16 percent in the shell (19). As presented in Figure 1, additional analysis indicates that most of these head punctures occurred in the lower half of the heads (20). Therefore, by providing puncture resistance only at the lower half of the tank heads (approximately 3.8% of the total tank car surface area), a significant reduction in the releases of hazardous material could be achieved. Furthermore, many head punctures occur because the couplers of adjacent cars become uncoupled, and one coupler acts as a "hammer" and punctures an adjacent car. If coupler disengagement could be reduced, a further decrease in the release of hazardous materials could be achieved.

In light of the above discussion, FRA, the Association of American Railroads (AAR), and the Railway Progress Institute (RPI) conducted research (in many instances, jointly) to develop cost beneficial methods of head protection and coupler vertical restraint systems. Jointly sponsored field tests at the Transportation Test Center showed that although neither head shields nor shelf couplers alone were effective in all accidents, the combination of the two safety devices was effective in all tested accident situations (21, 22).

In 1980, under a contract to FRA, Biotechnology undertook the development of an analytical methodology for predicting the vulnerability of tank cars to railroad impacts. It is concerned with tank car material properties, head shield material properties, tank car/head shield design characteristics, lading being transported, ambient conditions, and projectile characteristics. This study is scheduled to be completed in 1982.

STRUCTURAL INTEGRITY ASSESSMENT

Even if a tank car does not rupture during a railroad accident, its structural integrity may be so impaired that it will rupture unexpectedly at some later time. In at least two instances the tank cars did not rupture at the time of the railroad accident, but ruptured several days later causing deaths, injuries, and property damage (23, 24). The apparent cause was the increase in vapor pressure of the cargo in the tank car due to an increase in ambient temperature; consequently, the burst strength of the damaged tank cars was exceeded. Other explanations have also been advanced for these type of accidents (25, 26).

The AAR Bureau of explosives (BOE) has long recognized the problems posed by damaged tank cars and has adopted guidelines for assessing tank car structural integrity based solely on visual inspection. The BOE considers cracks in the base metal, no matter how small, are sufficient justification for unloading the tank as soon as possible. According to the BOE, the following conditions are sufficient cause for a tank car to be unloaded without moving it:

1. A crack combined with a dent or gouge.

2. A dent combined with a gouge.

3. A dent that crosses a tank seam weld.

4. For tank cars built before 1966, a dent with a radius of curvature less than 4 inches.

5. For tank cars built after 1966 (presumably made of AAR M-128 steel), a dent with a radius of curvature less than 2 inches.

In 1979, FRA and the U.S. Air Force initiated joint research on post accident procedures for hazardous material accidents. One aspect is the assessment of the structural integrity of those tank cars involved in railroad accidents. The study concluded that although there is a sound technological basis for the BOE visual guidelines, often tank car defects will not be visible. For example, insulated tanks cannot easily be inspected for defects. Defects on the inside or underside of a tank are not observable. Also environmental conditions, such as darkness, rain, and dirt on surface, etc., make visual detection of defects difficult. In addition, there may be a danger to personnel who are performing these time consuming visual inspections on the damaged tank cars. The joint study has identified acoustic emission techniques as a promising method for assessing tank car structural integrity. These techniques permit remote, continuous monitoring of the tank car once the detector has been attached. Magnetic detectors can be rapidly attached to the tank car surface.

CONCLUSION

In summary, FRA has sponsored research to reduce the frequency of releases of hazardous materials during impacts and to assess the structural integrity of tank cars subjected to impacts. Recent accident data indicate that this research has resulted in a significant decrease in the number of hazardous material releases (27).

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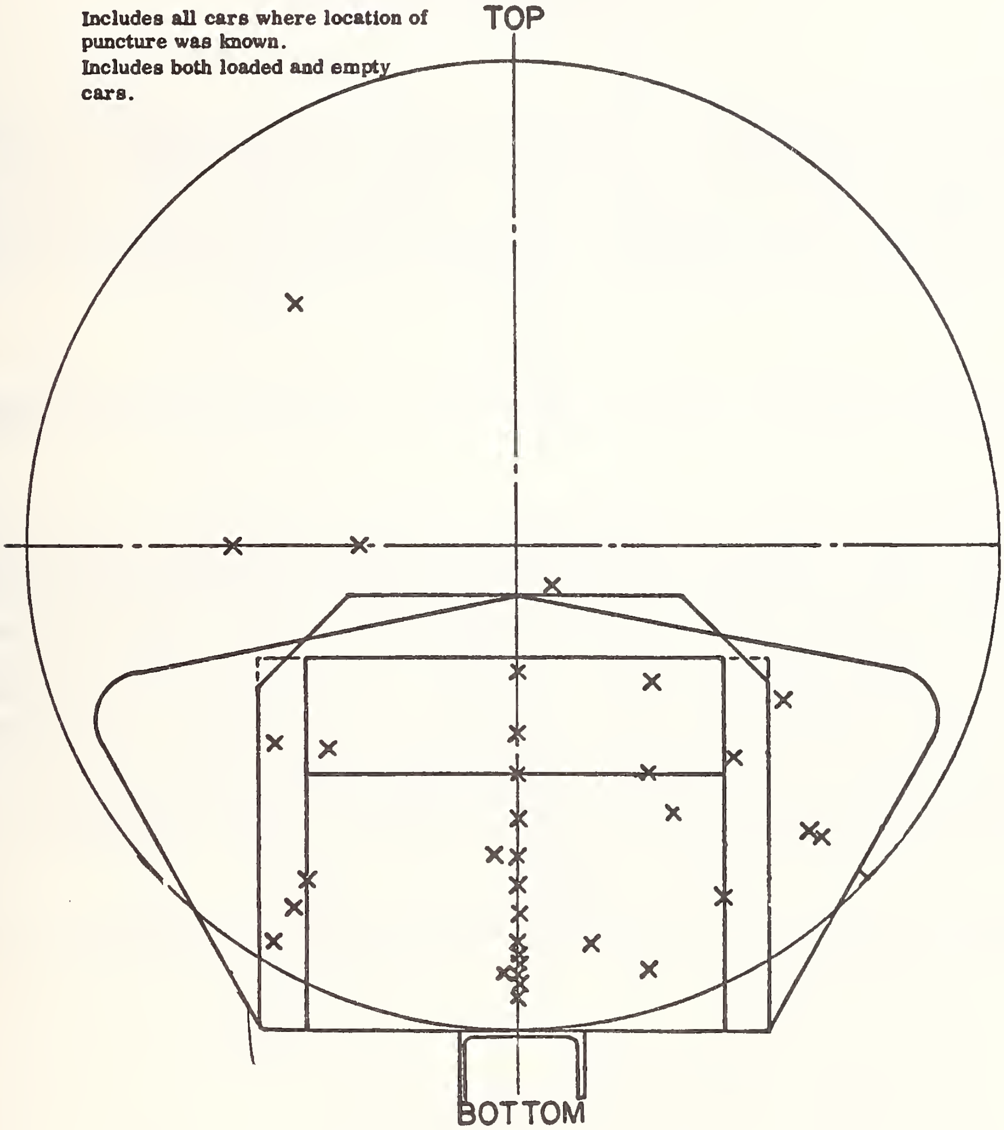
FIGURE 1

Head Punctures: Pressure Cars (DOT 112A340W, 114A340W) Generally With Stub Sills.
(X marks location of puncture.)



A LOCOMOTIVE EVALUATOR—NEW RESEARCH CAPABILITY THROUGH SIMULATION

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Abstract: The Locomotive Evaluator, presently being developed by the Federal Railroad Administration, is reviewed in terms of the basic difference in approach between a "research evaluator" and a "training simulator." The new research capabilities which will be possible with an advanced, real-time train action model combined with human input in a realistic environment are discussed along with the potential for improvements in both safety and efficiency.

Key words: Railroad safety; simulation; train handling; locomotive engineer; training.

New equipment, larger cars, and heavier trains make the job of the locomotive engineer more complex at a time when many experienced engineers are nearing retirement age. As a consequence, an increasingly complex job will have to be handled by newer engineers who must learn the job in a far shorter time than the engineers they will be replacing. The need for newer and better train-handling methods and the need to develop more effective training programs have resulted in the development of plans for a locomotive research evaluator. The locomotive research evaluator will provide a test subject with a realistic simulation of the visual, audio, and motion cues that the engineer would experience when actually operating a locomotive. This requires visual, audio, and motion subsystems that can provide real-time actions dependent on the action of the engineer.

The evaluator differs from training simulators because it will have the added capability of being a research facility. This capability is a result of:

- increased realism
- high-resolution video system
- ability to provide real-time train simulation
- simulation based on train characteristics and test subject reactions

The evaluator to be discussed has been essentially designed but only partially fabricated. As shown in Figure 1, a simulated locomotive cab will be mounted on a motion base. The engineer in the cab will receive visual cues of the chosen territory, at a rate dependent on locomotive speed.

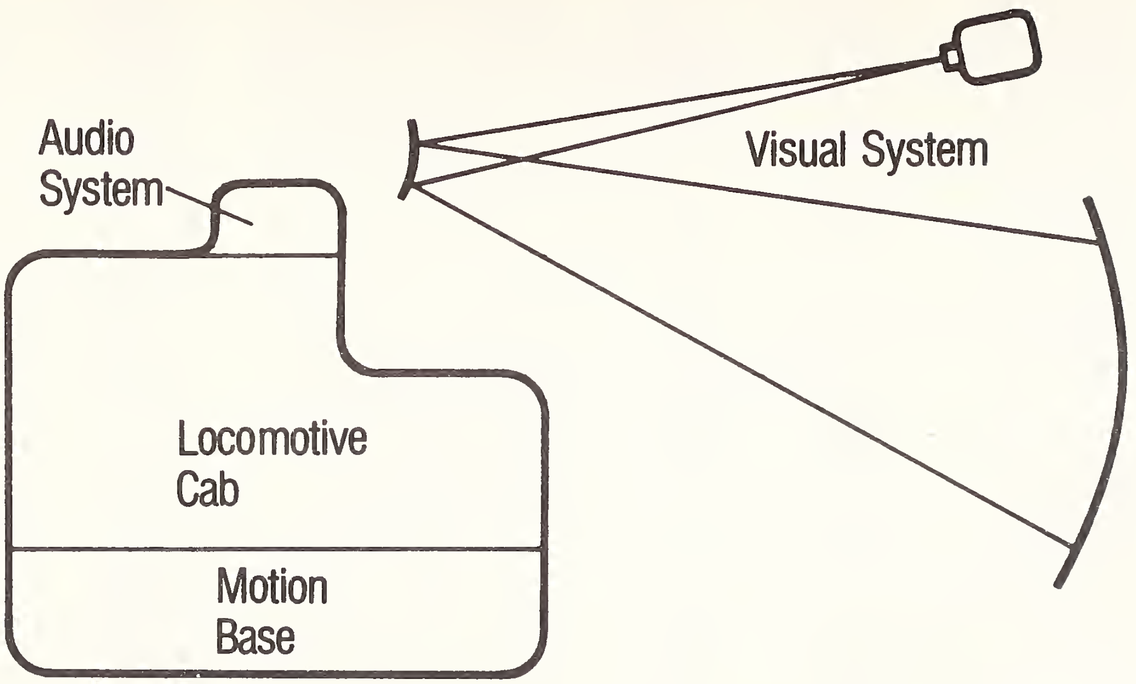

FIGURE 1. Physical layout of subsystems in Locomotive Evaluator

The evaluator will be capable of using films available for other training simulators but will also be equipped to use specially generated films, which will result in a wider field of view and a higher resolution. The projector runs at a constant shutter speed with variations in the rate at which film is pulled down dependent on the simulated locomotive speed.

The audio cues received by the engineer is part of a system with a memory, including digital reproductions of various sounds such as the engine, bell, horn, and brake pipe venting. These sounds will be combined using volumes and frequencies that take into account the speed of the train, as well as any dopplers shifts. A radio speaker will be placed in the cab to give train orders to the engineer.

Since train dynamics will be modeled in real time, the motion base is a key element in the evaluator. The motion base requirements call for the 5 degrees of motions shown in Figure 2; longitudinal, lateral, vertical, roll, and pitch.[1/] Yaw is not thought to be critical as long as the lateral motions are present.

The locomotive cab will be as realistic as possible; an actual cab was modified to reduce its weight. Controls in the cab will be computer operated to give the engineer such data as speed and brake pipe pressure. In the evaluator system, the cab controls will also be the entry point for the engineer. The test subject will have access to the same controls as in an actual locomotive run.

As shown in Figure 3, all the described subsystems will be itegrated in the evaluator system. The computer will control the system, but overrides will be possible by the operator. The planned experiment will be put into the computer, and data such as track class and grade will be entered into the computer memory. The test subject will be in the cab, just as an engineer would be in a real train, and the operator will be at the master controls to provide any desired inputs. Two projects will give the operator the option of introducing variables into an experiment. For example, the operator can set the scene to enter a siding rather than to remain on the main line--if the siding has been photographed and in the film library.

Having briefly described the evaluator system, I would like to discuss its varied uses. The idea of train simulation is not new, but the goals of previous simulators differ from the goals of this locomotive evaluator. Although training has been the primary goal in the past, the goals for the evaluator also include research.

To appreciate the need and potential use for a locomotive evaluator, we must recognize that the locomotive engineer is a key element in the resultant train forces. For example, for a 75-car train, each coupler would be in either draft or buff, and an adverse combination of coupler forces could lead to a derailment.

1/ The current design includes six degrees of freedom.

Simulator Motion Modes Required

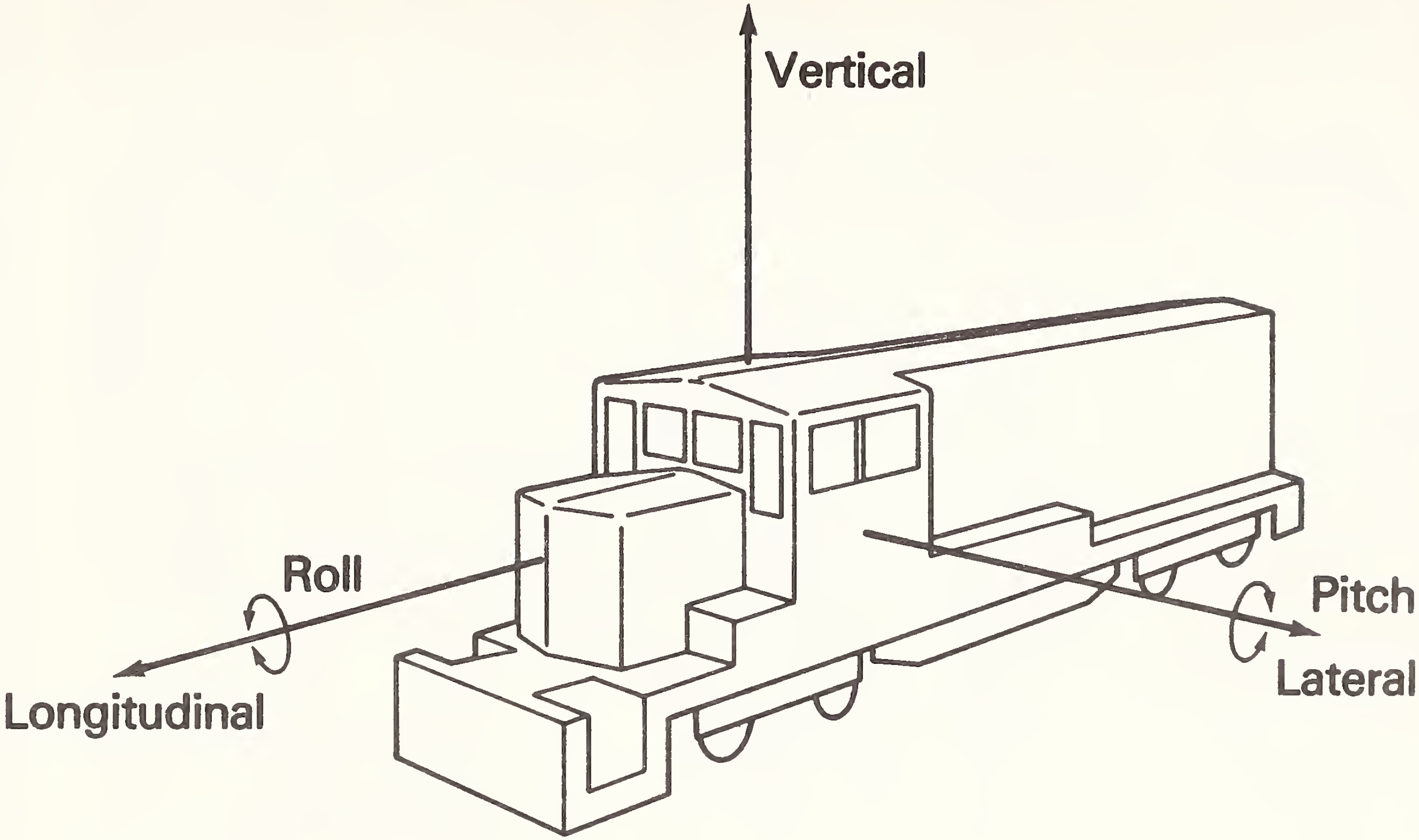

FIGURE 2. Motion modes required for Locomotive Evaluator

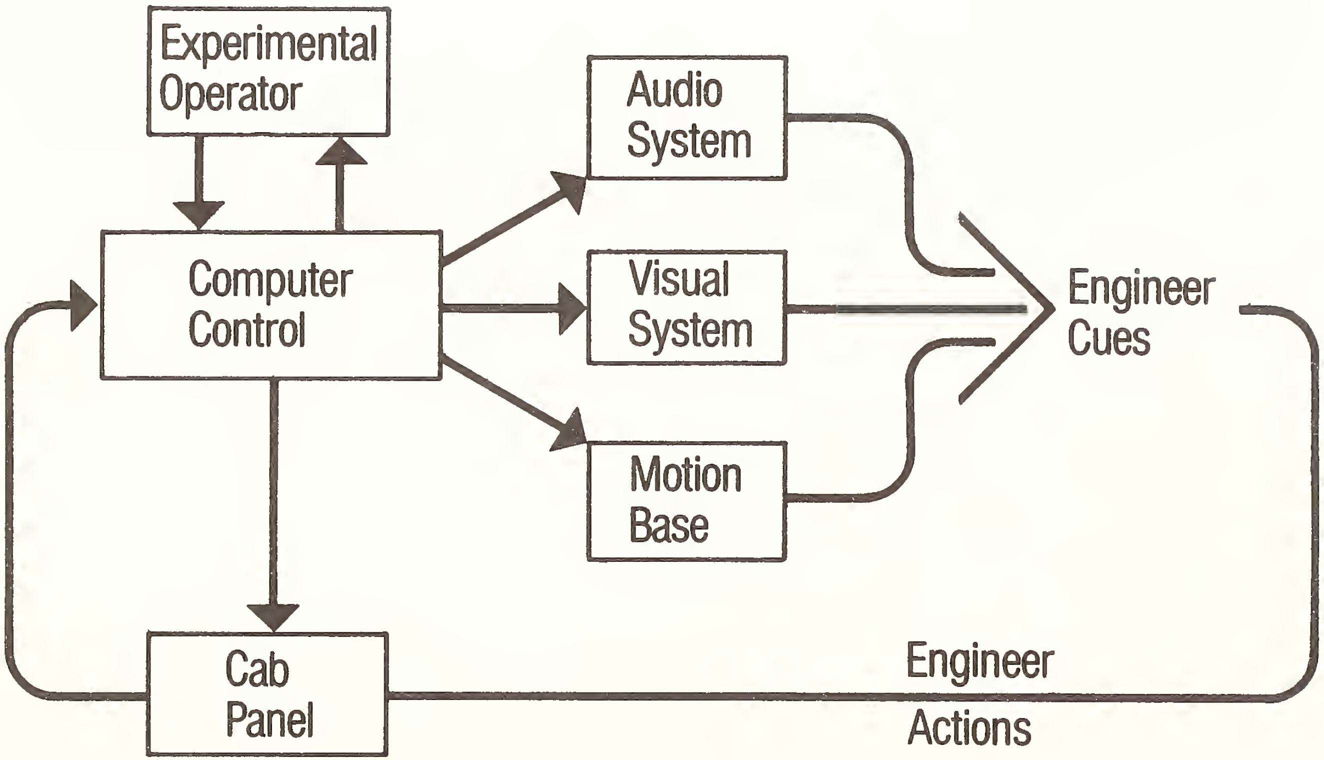

FIGURE 3. Interrelationship of subsystems in Locomotive Evaluator

In 1980, 2,323 accidents were attributable to human factors.[2/] While not all these accidents could have been avoided by the evaluator, the following examples show some areas where the evaluator might solve the problems.

Improper use of automatic brake	20 accidents
Improved use of dynamic brake	6 accidents
Employee falling asleep	4 accidents
Buffing or slack action excessive	307 accidents
Lateral drawbar force on curve excessive	92 accidents

These accidents could be addressed through training, new train-handling techniques, or new train-handling aides. These factors are interrelated to some extent since, for example, development of train-handling techniques are useful only if they provide the engineer with information relating to a train-handling technique. At this time, each will be considered separately.

There are two ways in which the evaluator can be used in training. First, the evaluator can be used as a training simulator by running engineers through the evaluator to teach new recruits basic locomotive handling, to teach experienced engineers new train handling techniques or to provide refresher courses. The second involves the use of evaluator as a research tool. The evaluator can be employed to develop and test new training programs that may include classroom training, over-the-road training, or a training simulator. Using the evaluator in this manner can aid present efforts to develop model engineer-training programs.

As for train-handling techniques, a computer train dynamics model can predict train action based on predetermined engineer actions, but only a simulator with a real-time train dynamics model can include the engineer in the loop, giving real-time feedback cues based on the subject's own actions and the resultant train dynamics. This provides input from the engineer that is not available from a computer model and provides the controlled environment not available in over-the-road testing.

The third area is train-handling aids. Here, the evaluator can help answer the question of what information the engineer needs to handle the train. Too much information can be as detrimental as too little information. If a system were developed to provide the engineer with coupler force for each coupler in the train, the engineer would not be able to comprehend the information, and, as a result, the data would be ignored. On the other hand, if a system could indicate whether couplers at certain key locations arein draft or buff, the lesser amount of the data could be of far greater use to the engineer. The determination of the

2/ Accident/Incident Bulletin No. 149, Calendar Year 1980, Federal Railroad Administration, June 1981.

optimum amount of information for the engineer will be a key function of the evaluator.

Besides its use in training, train-handling techniques, and train-handling aides, the evaluator can be used in accident reconstruction. In many instances when an accident occurs, the exact chain of events leading to the accident is uncertain. Key inputs, such as the exact time the brake was applied or whether the intrain forces had reached high levels in unfavorable terrain may lead to ambiguities. Often, the results are several scenarios that "may" have led to the accident. The evaluator can be used to recreate each of the scenarios, and a determination can then be made of the feasibility of each and its consequences.

To summarize, the evaluator, a system for both research and training, combines the features of training simulators, computer train dynamics models, and over-the-road operations. Such a facility can aid in preparing locomotive engineers to handle trains with an increasing number of heavy cars and locomotives with increasing amounts of sophisticated, new equipment.

I look forward to a day when we can discuss the evaluator in terms of results rather than plans.

SESSION IV

DAMAGE PREVENTION IN THE RAILROAD INDUSTRY

CHAIRMAN: BRAD JOHNSTONE
PULLMAN STANDARD

DESIGNING FOR DAMAGE PREVENTION IN THE RAILROAD TANK CAR INDUSTRY

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Abstract: This paper describes a cooperative industry project between the railroads and the major tank car companies that has had the objective of improving the safety of tank cars in accidents. The project was established in 1970 following a sudden rash of catastrophic accidents. Organized to study all aspects of tank car design, the project sought to find cost effective changes that would reduce the probability of tank car punctures and fire induced ruptures in the accident environment. Work was divided into 18 technical phases, a major one of which was the collection of accident data to develop a solid base upon which to evaluate potential "solutions". In many areas, work was carried out cooperatively with the Federal Railroad Administration which also had ongoing research with the same objectives.

This paper describes the various phase activities and the major results of the research. One has been the retrofitting of 22,000 liquified compressed gas tank cars with special (shelf) couplers to reduce the severity of jackknifing and the potential for head punctures. Another has been the application of head puncture protection (head shields) on the same cars and the application of fire retardant insulation to 19,000 of the cars. Means also were developed for protecting bottom fittings from being torn off when cars slide along the ground after being detrucked. The project still continues with major effort comprising accident data collection and evaluation of the need to improve the safety of other type cars.

Key words: Damage prevention; hazardous materials transportation; railroad tank cars; safety research.

I. INTRODUCTION

In the transportation industry, damage to cargo can be very costly, as can damage to the transport vehicles themselves. In the case of railroad tank cars, their cargo, or "lading" cannot be damaged, but it can be lost. This leads to a dollar loss, but more importantly, in some cases it can create a serious hazard. This hazard can be to the environment, to people, or to both.

This paper describes a major program which was undertaken to reduce the hazards of tank car transportation by reducing the probability of lading release in accidents.

II. THE PROBLEM DEFINITION

To accurately define the problem, a great deal of information was required on tank car accidents. While accident scenarios are infinite in variety, the most hazardous can be categorized into three general types:

(1) A tank car is punctured, flammable lading is released, fire ensues, and, in a chain reaction sequence, other cars rupture violently from thermal overheating and segments of the cars are hurled (rocketed) considerable distances. Rocketing can expand the danger zone from the accident site to a radius of several thousand yards.

(2) A tank car is punctured without ensuing fire, but a hazard is created by the toxic or contaminating nature of the lading which is released.

(3) A tank car carrying liquified flammable compressed gas is punctured, lading is released without ignition, the area is flooded with flammable vapor, the vapor reaches a source of ignition, and a sudden concussive fire results.

Mechanically induced punctures are the initiating cause in each case, and they constitute the basic problem for investigation. Were they totally preventable, there would be no other problem to solve. Because no practical solution will prevent all punctures, the fire induced rupture is also a matter for study. A solution which improves puncture resistance reduces the probability of rupture (and rocketing), but a solution which shields a tank car against fire even further reduces these hazards. In other words, the effectiveness of a puncture shield and a thermal shield are additive. Both puncture shields and thermal shields were developed under this program. Methods of reducing the probability of rocketing, given a thermal rupture, were also investigated, but no practical solutions were found.

III. ORGANIZING TO ATTACK THE PROBLEM

Figure 1 from ref. {44}[1/] illustrates an accident peak which occurred in 1969. There is no explanation for it, and it has to be considered a statistical freak. Some people predicted that the 1969 trend would continue upward, an obviously intolerable situation. Railroads,

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[1/] Numbers in brackets refer to references listed at the end of this paper.

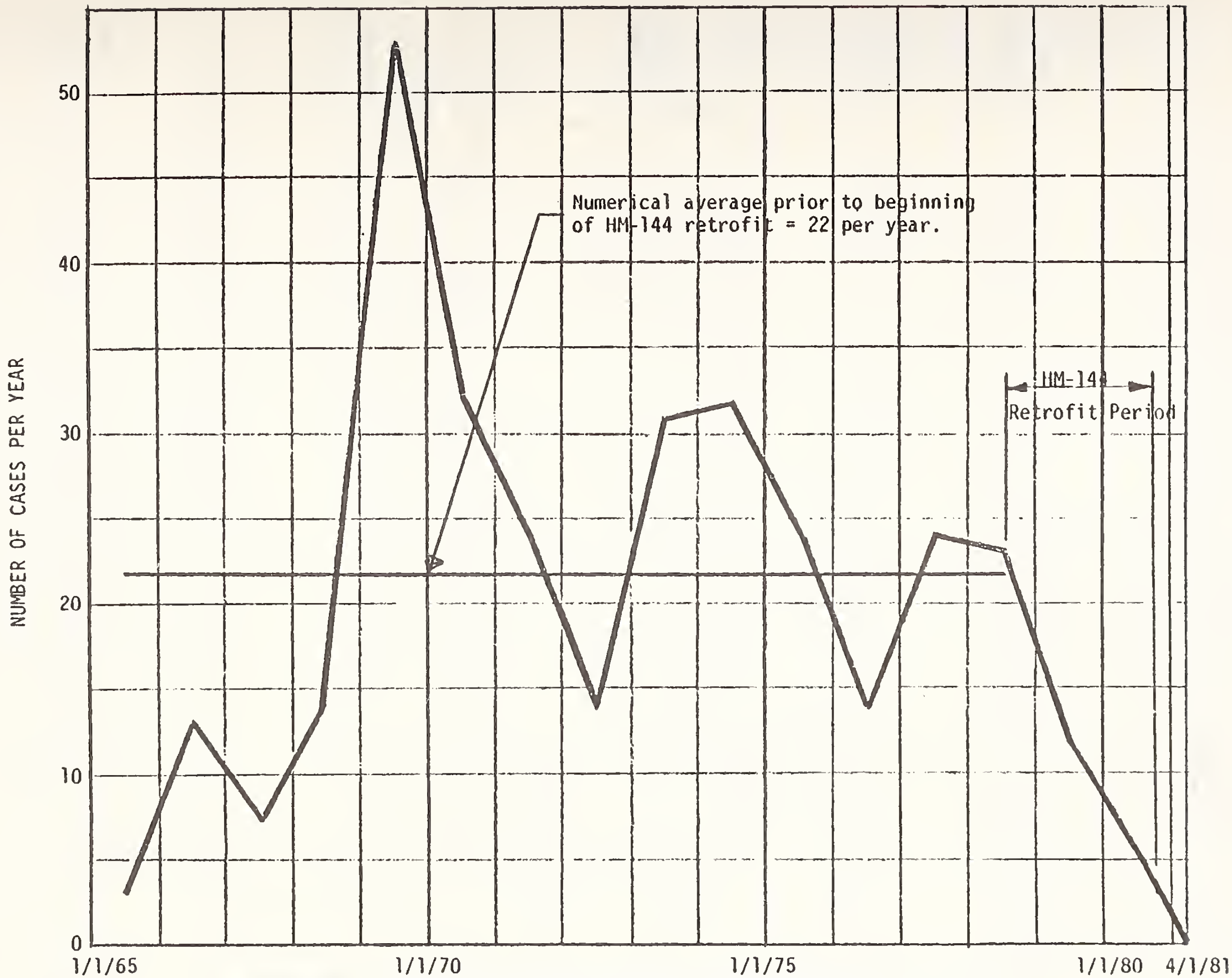


FIGURE 1 -- ALL CASES COMBINED OF HEAD PUNCTURES, SHELL PUNCTURES AND FIRE INDUCED RUPTURES OF 112(114) CARS

shippers, tank car builder-owners, and government agencies all were anxious to organize a program to reverse the trend. There emerged a cooperative industry project under an agreement between the Association of American Railroads (AAR) and five major tank car builders acting through the Railway Progress Institute (RPI). Policy direction was furnished by a steering committee called the Project Review Committee comprised of high level representatives of the tank car companies, the AAR and the railroads.

The project has been administered by Dr. William J. Harris, Jr., Vice President, AAR Research and Test Department. It is directed technically by Earl A. Phillips, Project Director, who is assisted by L. L. Olson of the AAR Technical Center in the capacity of Deputy Project Director.

The project began on March 30, 1970. It has been supported by contributed funds and manpower from the AAR and the tank car companies through the RPI. Funds and manpower are contributed on a 10:3 and 5:2 ratio, respectively, by RPI:AAR.

A total of 30 man-years of effort has been expended through 1980. Project costs, including the value of this manpower, have been over 4 million dollars. The Federal Railroad Administration (FRA) of the U.S. Department of Transportation (DOT) also has conducted substantial research in the area of tank car safety, and test programs in many areas were carried out cooperatively between the project and the FRA.

IV. PROJECT OBJECTIVE

The project objective is to seek means to improve tank car safety in accidents. More specifically, it is to develop, through study, tests, and research, improved knowledge of the factors that cause punctures, ruptures, and rocketing of tank car tanks in accidents; and to develop, through the use of this knowledge, practical and proven means to reduce the probability of these events.

V. PROJECT SCOPE

The project scope covers only tank car behavior during the accident period as influenced by the associated mechanical and thermal environment.

Importantly, the scope does not include the initial cause of an accident unless a tank car tank, or a component peculiar to a tank car, is responsible. Thus, the failure of a tank car wheel, which is like all other freight car wheels, is outside the project scope. This exclusion was natural since research on accident prevention is a much broader activity, being the continued responsibility of the railroads, the AAR, the FRA, and the supply industry. Post accident handling and cleanup activity is also outside the scope.

VI. PROJECT PHASES AND REPORTS

Project technical activity has been divided into 18 phases. Forty-two technical reports have been issued on completed studies in specific areas. These are listed at the end of this paper and will be referenced as appropriate.

Report distribution has been widespread with copies being sent to approximately 160 recipients, including personnel of railroads, the AAR, various shipper oriented trade associations, and United States and Canadian government agencies concerned with rail transportation.

VII. RESULTS OF RESEARCH

Phase 01 - Accident Data Collection

Beginning with the year 1965, every known accident in which a tank car was damaged has been logged into a computerized data base. The procedures used in this activity are described in {16}. This is an ongoing activity, currently being carried out by a staff of two full-time and four part-time people.

The data, all of which are treated proprietary, are collected from many sources, including:

The AAR Bureau of Explosives
The Federal Railroad Administration
(Rail Equipment Incident Report, Form FRA F 6180-54)
(Office of Safety, Accident Investigation Reports)
Materials Transportation Bureau
(Hazardous Materials Incident Reports, DOT F 5800.1)
National Transportation Safety Board
Canadian Transport Commission
Railroads
Tank Car Owner-Builders
Tank Car Owner-Shippers
Chlorine Institute
Chemical Manufacturers Association
American Petroleum Institute
Compressed Gas Association
National LP Gas Association

For the period 1965 through March, 1980, over 18,000 tank cars were damaged in accidents. One out of seven lost lading. A general breakdown of the data is shown in Figure 2.

Phase 02 - Accident Data Analysis

The analysis of the accident data is handled under this Phase. Different types of analyses have been made throughout the project studies as various needs arose.

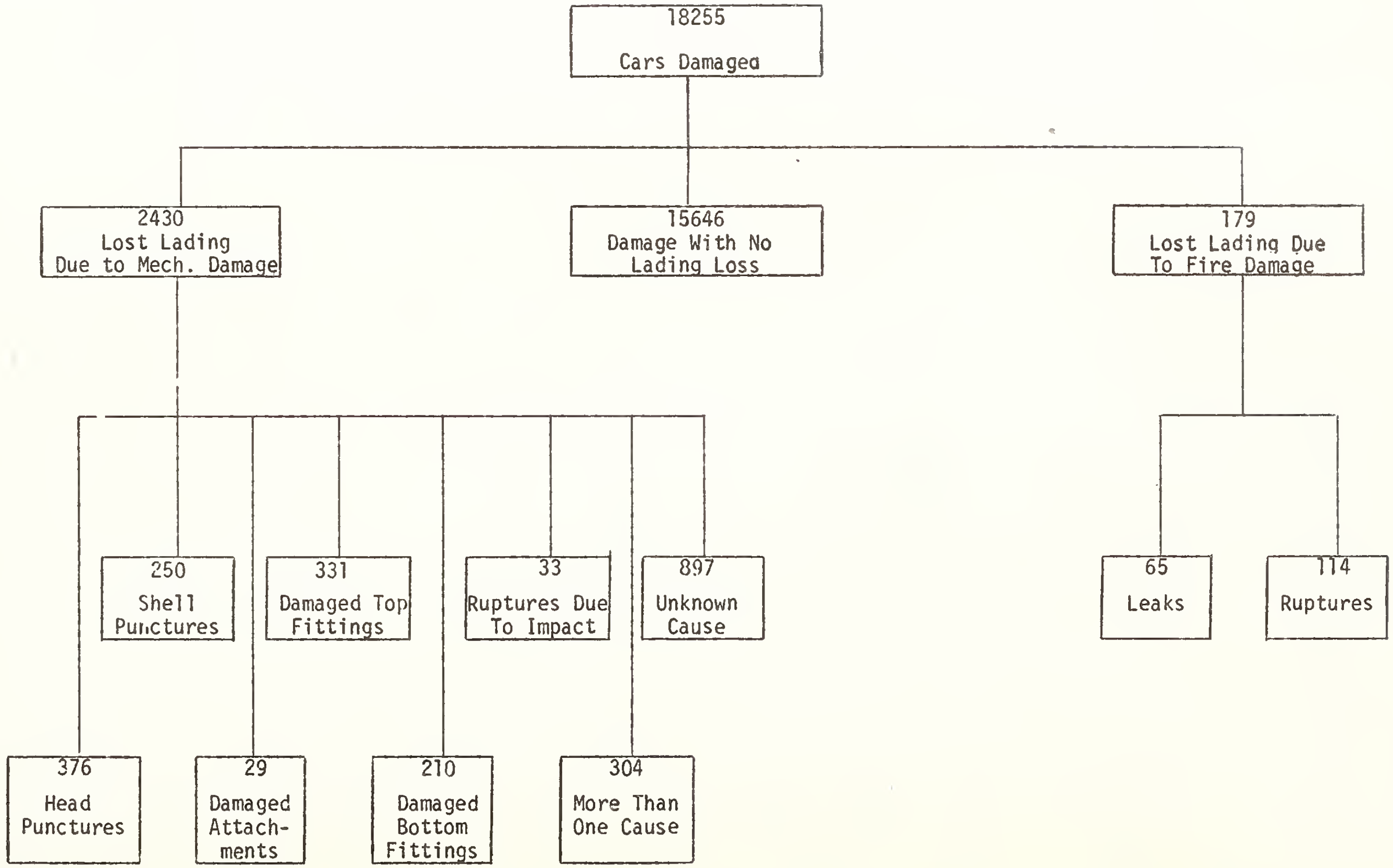


FIGURE 2 -- SUMMARY OF TANK CARS INVOLVED IN
ACCIDENTS DURING 15¼ YEAR PERIOD JANUARY, 1965 THRU MARCH, 1980

Cost-effectiveness was important early in the program since there were a large number of solutions to evaluate and compare. At that time, dollar losses incurred by the railroads due to lading loss from the tank cars were determined for each accident. These losses included the cost of the lading lost and the costs of damage to other equipment, and to society, caused by the lading loss. The losses were catagorized by the specific events which caused them; e.g. head puncture, fire induced rupture, etc. From these, the potential values of design solutions were determined.

During the first two years of the project, this type analysis was carried out for all cars which lost lading in accidents over the 6 year period 1965-1970 {10} {18}. Some dollar losses were found for which the cause, the class of car involved, or both, were unknown. These were included by prorating them over all known causes and classes of cars.

These losses were then converted to "benefit" values of solutions. The philosophy used was that the dollar expenditure which could be justified for a design solution was the "present value" of all future savings which would be realized by its application. The accident spectrum of the future was assumed to be similar to that of the past.

Head punctures of 112A(114A)[1/] cars can be used to illustrate the methodology. These accounted for losses of about 4 million dollars over the 6 year period. There was an average of 12,000 such cars in service over this period, so the loss per car per year was 4,000,000/6(12000) = $55.

For a solution having an expected life of 30 years, and assuming a 10% rate of return (reasonable in 1972), the present value of the $55 per year stream of payments, using accounting principles, is $527. This is the justifiable expenditure for the solution if it is 100% effective.

Present values of other solutions were developed in similar fashion. The results are shown in Table I, from {22}. These numbers are out of date today, but in 1972 they served as a basis for establishing priorities. As seen, head shielding, shell shielding, and thermal shielding of 112A(114A) cars headed the list.

To carry the head shield example further, from a detailed review of accident and test data, it was estimated that a 5' wide x 4' high x ½" thick steel head shield would have prevented one-half of all past head punctures on 112A(114A) cars. The actual estimated benefit of such a shield is then $527/2 = $264. The estimated cost of the shield was $474, so it was not considered cost-effective.

1/ The 112A car is a "pressure" car that typically carries anhydrous ammonia and liquified flammable compressed gases. The 114A car is essentially the same except for variations in fittings.

TABLE I

MAXIMUM JUSTIFIED INITIAL COST OF DESIGN SOLUTIONS WHICH ARE 100% EFFECTIVE ASSUMING 10% COST OF CAPITAL

(Data from RA-02-2-18 and RA-02-1-10, rounded to nearest dollar)

			TYPE OF TANK CAR								
			Welded Steel Non-Pressure						Steel Pressure		Other:
	Service Life of Design Solution (Years)	Riveted Steel	Dome Cars - 103W		Cars Without Domes - 111A-W						Aluminum, Cryogenic etc.
					Full Underframe		Stub Sills				
Cause of Lading Loss			Non-Ins.	Ins.	Non-Ins.	Ins.	Non-Ins.	Ins.	Ins. 105A-W	Non-Ins. 112A-W	
Head Puncture	30	$5 1/	8	1	296	4	336	65	31	527	16
Shell Puncture	30	7 1/	12	7	29	3	63	7	8	1136	11
Attachment Damage	30	1 1/	0	0	0	0	21	0	3	5	0
Bottom Fittings Damage	30	0 1/	1	3	10	2	79	13	-	4 2/	0
Top Fittings Damage	30	0 1/	3	4	35	10	28	0	3	28	9
Head Puncture By Coupler On Adjacent Car	10	< 5 1/	< 8	< 1	194 3/	< 4	221 3/	< 65	< 31	347	< 16
Fire Induced Rupture	10	← Small (not determined) →								1808	Small
Rocketing of Tub Section	30	0	0	0	0	0	0	0	0	416	0

NOTES:

1/ - based on 5 year remaining life of tank car and shield

2/ - applies to DOT 114A-W cars only

3/ - not derived directly from accident data, but calculated by assuming same ratio as for head shield loss between 112A cars and respective car

These were 1972 benefit and cost values. As time went on, circumstances changed, and head shields became justified. They now have been installed on over 20,000 112A(114A) cars.

Recent accident data have been reviewed to quantify the safety improvement resulting from the application of these "shields" on 112A(114A) cars {44}. Dollar losses were not considered; instead, only the frequency of lading loss cases was developed. The results (including shell punctures and fire induced ruptures) are shown in Figure 1. An average of 22 cases per year occurred over the period 1965 to 1978. The drop-off beginning in 1978 is evident as the retrofit program began. The effectiveness of the head shield appears to be significantly higher than predicted, although it is still too early to accurately quantify the improvement.

Phase 03 - Tank Car Steels

The objective of Phase 03 was to seek improvements in steels which would reduce the probability of punctures and ruptures. Of primary concern were steel toughness and strength properties at various temperatures.

A large amount of test work was conducted on currently used steels, on steel samples removed from tank cars involved in accidents, and on steels used on older cars which were being scrapped. The object was to characterize their physical, chemical, metallurgical, and fracture toughness properties. The latter were determined using the Charpy v-notch test, drop weight tear test, dynamic tear test and the NDT drop weight test. The results are published in {33}.

Materials for the most part met specifications. Variations beyond specified ranges in chemistry and tensile properties were few. In the case of accident samples, differences in mechanical properties were largely attributed to alterations caused by the accident. None of the above variations was believed to be a factor in the cause or extent of damage in any of the accidents.

Figure 3 from {32} summarizes one fracture mechanics aspect of the study. This figure applies to TC-128-B steel which has been commonly used for 112A(114A) cars since mid-1960. The vapor pressure of the liquified gases which these cars carry is a function of temperature; hence, the hoop stress in these cars is a function of temperature. This hoop stress-temperature relationship is shown in Figure 3 for propane, ammonia, and vinyl chloride. Based on dynamic tear tests, the 50% shear fracture temperature was determined for a number of specimens in the "normalized-stress-relieved" and the "as-rolled-stress-relieved" condition. This temperature point is called the "Yield Criteria" (YC). At this temperature, the steel is sufficiently resistant to fracture to require yield level stresses for continued propagation of a through crack. Above this temperature, the fracture is stable and requires

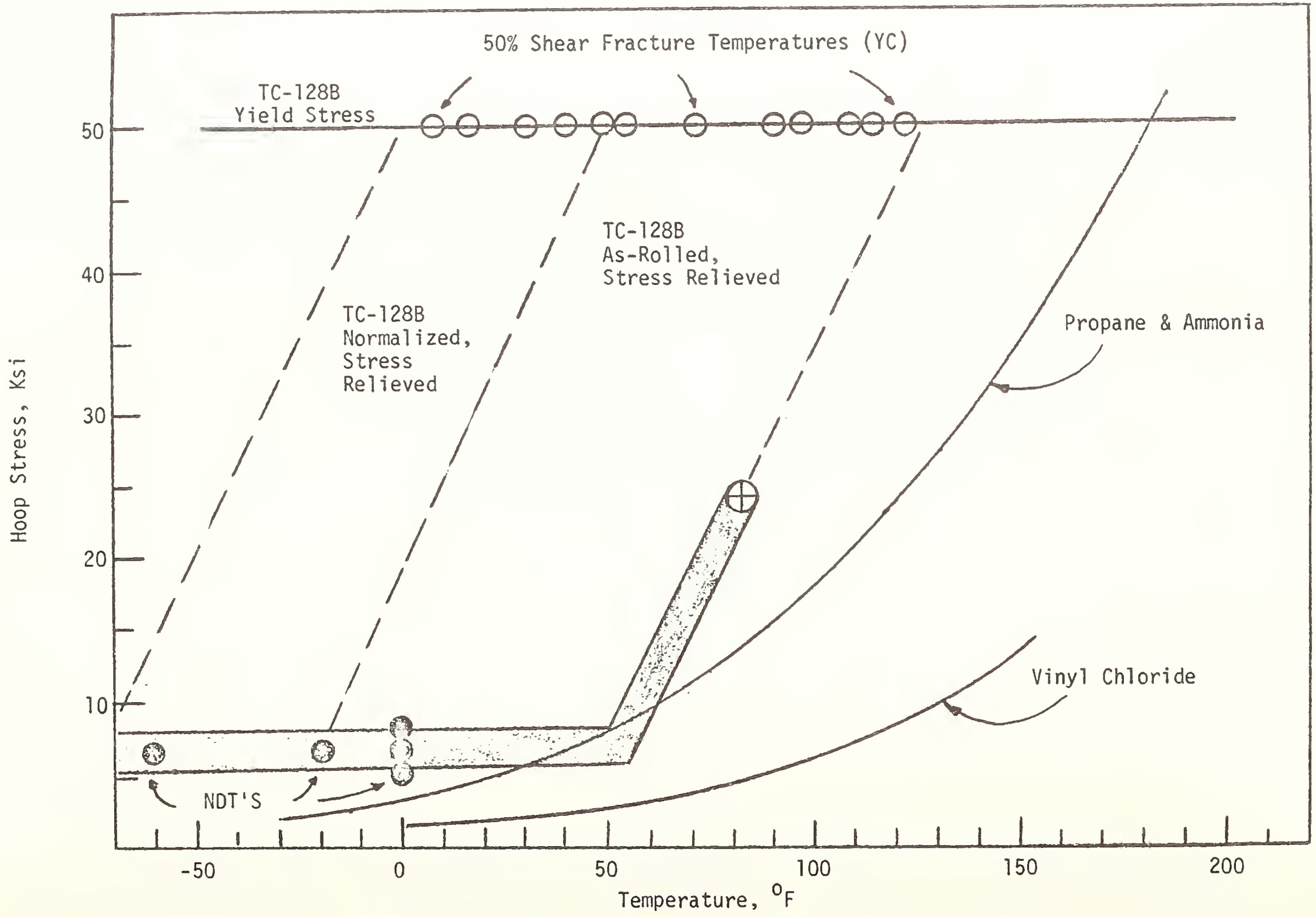


FIGURE 3 -- HOOP STRESS - TEMPERATURE RELATIONSHIP COMPARED TO STRESS REQUIRED TO PROPAGATE FRACTURES IN TANK CARS

higher than yield (plastic) stresses to propagate. Nil Ductility Transition (NDT) temperatures were determined from drop weight tests. At temperatures below NDT, fractures are highly brittle. The actual YC and NDT points for the specimens tested are shown in Figure 3. The YC points are plotted at the yield stress and the NDT points over a stress range of 5 to 8 psi, below which fractures will not propagate. For as-rolled, stress relieved steels falling in the high transition temperature range, the figure shows that fractures will not propagate in a brittle manner. Below 50°F the hoop stress is too low for brittle fracture propagation. A very small probability of brittle fracture exists around 50°F. These conclusions relate only to pressure induced hoop stresses. If large areas of high stresses are temporarily created in an accident by external mechanical forces, the probability of brittle fracture is increased if the tank is sufficiently cold. The accident data confirm, however, that this is a rare occurrence. In summary, the detailed study of TC-128-B steel showed it to be an optimum product for the service, refs. {32} {33}.

Phase 04 - Literature Review

Background experience and literature in the technical areas of interest under the Project has been continually under review. A reference library was established and is being maintained under this Phase.

Phase 05 - Tank Car Heads

Phase 05 was devoted to the development of a method to reduct the frequency of head punctures. The first activity was a conceptual design study carried out under contract to the FRA (Phase 13 of the Project). The study concluded that a 5' wide x 4' high x ½" thick steel plate spaced in front of the tank car head was the most promising design to progress in a test program.

Initial tests were conducted on old riveted tank cars. The heads on these cars were impacted with a standard "E" coupler mounted in elevated position on a specially built "ram" car. Forty-two impacts were conducted on 21 cars. Variables studied were car weight, outage, internal pressure, impact height, number of backup cars, and the protection offered by standard insulation and then by the ½" thick head shield.

Next, two series of 1/5 scale tests were conducted. The objective of the first series was to determine if a difference in puncture vulnerability existed between two different designs of the head-to-sill connection on stub sill tank cars. The two lefthand sketches in Figure 4 show one design, and the two righthand sketches show the other. Three backup cars were coupled behind the test car in each test. Thirty-two tests were conducted on both ends of 9 non-pressure tank cars and 9 pressure tank cars. The results (scaled up to full scale) are shown in Figure 4. They indicate that the design incorporating the reinforcing shoe between the head and sill is slightly more vulnerable (lower

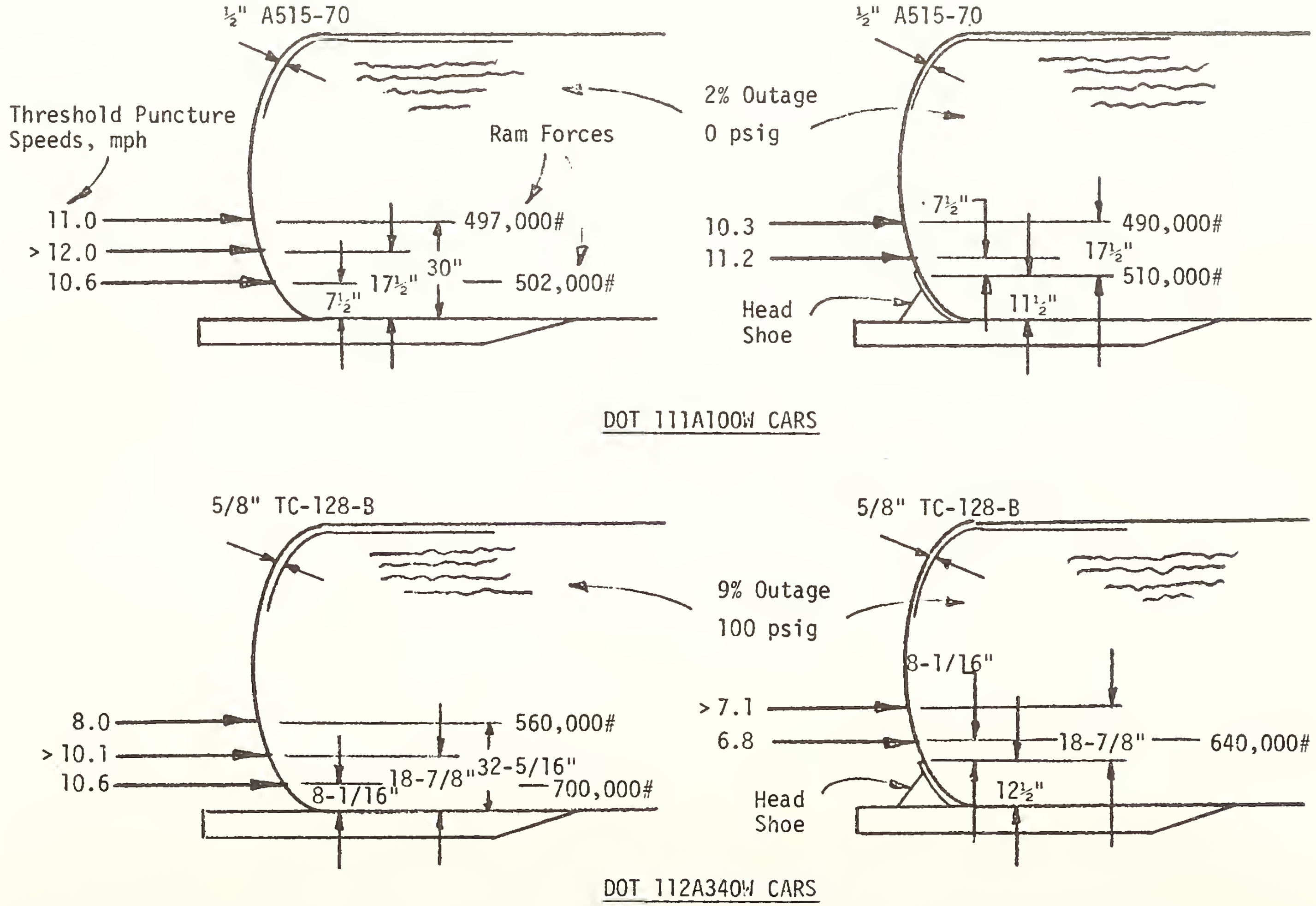


FIGURE 4 -- RESULTS OF 1/5 SCALE HEAD IMPACT TESTS TO COMPARE VULNERABILITY WITH AND WITHOUT HEAD SHOE

puncture speeds), but it was concluded that the difference was not significant.

The objective of the second series was to study head punctures of 112A340W[1/] pressure cars. These 1/5 scale tests were useful in reducing the scope of the final, and expensive, full scale tests. Thirty-two tests were conducted covering several variables. The results are plotted in Figure 5, from {17}. As seen, vulnerability increases with increasing internal pressure, decreasing head thickness, and decreasing head material tensile strength. With the (scaled down) ½" thick head shield, impacts up to 17.5 mph produced no puncture. A final impact was conducted using a 26% heavier ram car. At 17.8 mph, no head puncture occurred. With the standard ram car, and assuming momentum governs, this would be equivalent to a 22.4 mph impact, which is the top point plotted in Figure 5.

The final test program employed two identical full size 33,500 gallon 112A340W cars, one without head shields and the other equipped with ½" steel head shields at each end. Each car was loaded with water and pressurized to 100 psig. Each was backed up by three loaded cars. The results also are plotted in Figure 5. On the car without a head shield, a 9.3 mph impact produced a 9 inch dent in the head, but no puncture. In a second test on the other end of the car, the head was punctured at 12.7 mph. On the car with the head shield, no puncture occurred at 15.5 mph. It is emphasized that in a derailment these impact speeds are not train speeds, but are the closing speeds between cars. On the other hand, they are the actual speeds in a switchyard where one car is stationary. The second (higher speed) test planned for the head shield on the other end of the car could not be carried out because the ram car was damaged beyond repair in the 15.5 mph test. Thus, the improvement offered by the shield could not be quantified, but it was clear from all tests that it was significant.

All 112A(114A) cars which carry hazardous liquified compressed gases now have been retrofitted with the ½" thick steel head shield. It can be the form of a trapezoid spaced in front of the lower half of the tank head; or, if the car has a blanket type thermal shield (see Phase 11), it can be a ½" thick steel jacket head covering the insulation. The ½" steel jacket design also now is required on new 105A cars carrying flammable liquified compressed gases and ethylene oxide (see Phase 17).

Phase 06 - Safety Valve Liquid Discharge Capacity

When a tank car carrying liquified compressed gas is heated in a fire, its contents can expand to where the tank car becomes liquid-full. The safety valve must then maintain safe tank pressure by momentarily discharging liquid. It may also be called upon to discharge liquid in the event the tank is overturned and exposed to fire.

1/ This is a 112A car that has a test pressure of 340 psig.

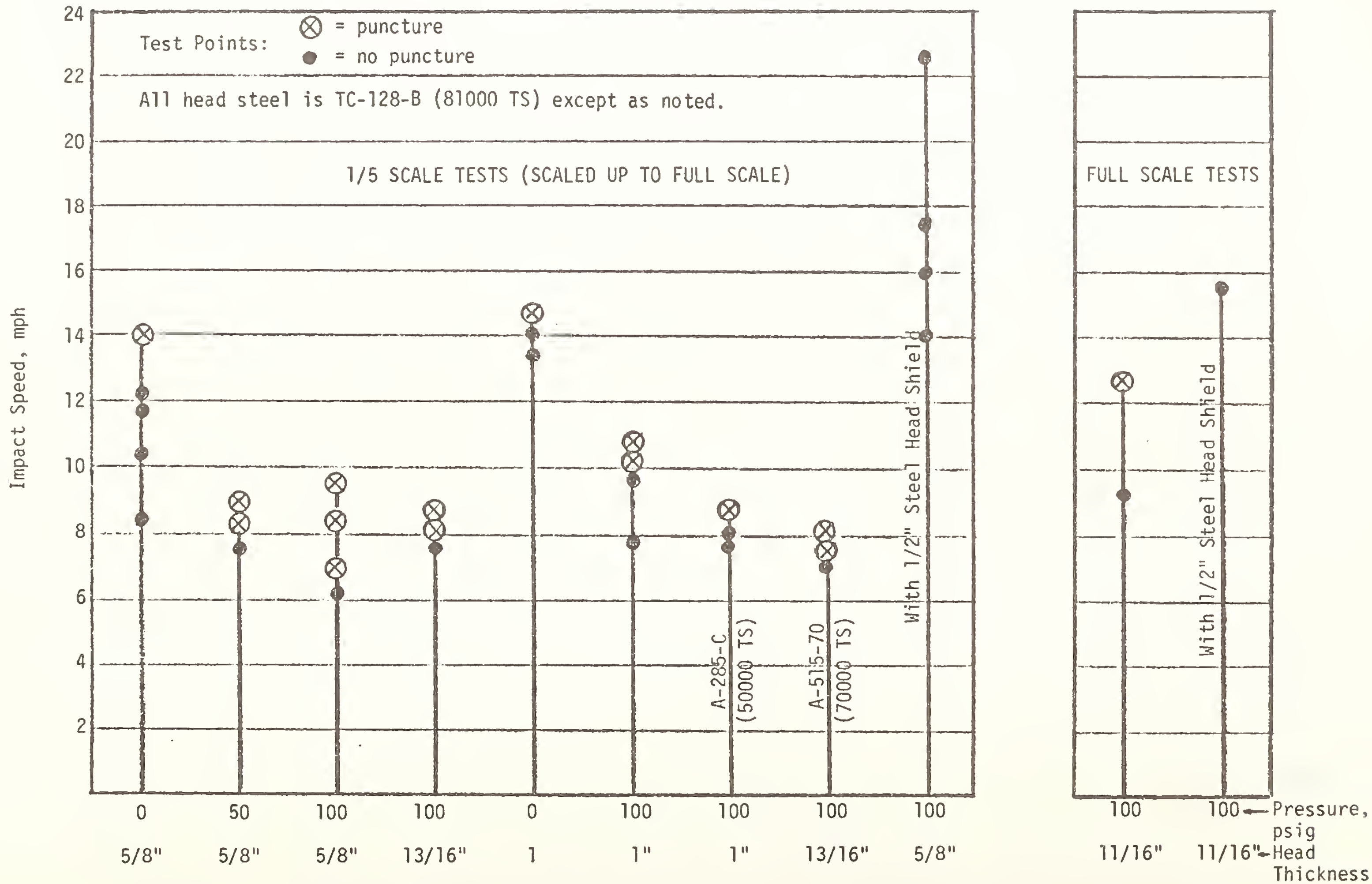


FIGURE 5 -- SUMMARY OF 1/5 AND FULL SCALE HEAD IMPACT TESTS

As in other pressure vessel codes, the tank car specifications require that safety valves be sized on the basis of vapor discharge. There being no firm data on liquid discharge capacities, this Phase was established with the objective of determining such capacities.

A special 20,000 gallon test tank was furnished by the Project and installed at Edwards Air Force Base in California. Tests were conducted by the Air Force Rocket Propulsion Laboratory under contract to the FRA. Products tested were water, nitrogen, and propane. Means were provided for mounting the safety valve at both the top and bottom of the tank to measure vapor and liquid discharges. These were "blow-down" tests. Tank pressures were raised above the safety valve setting, then released to the safety valve which discharged until the pressure dropped to the valve's reseat setting.

There are two manufacturers of safety valves for 112A(114A) tank cars. Both valves exceeded the specification requirements for vapor discharge. Problems with the test set-up led to difficulties in assessing their liquid discharge capabilities.

Phase 07 - Safety Relief Device - General

This Phase covered all types of safety relief devices on all classes of tank cars. Its objective was to develop design changes to assure safer containment, or safer release, of hazardous products in accidents. Activity did not progress beyond initial planning since the evidence did not show that current devices were deficient.

Phase 08 - Reduced Scale Model Studies

Consideration was given to conducting scale model derailments to study the influence of variables such as truck securement, coupler types, train makeup, and train speed. Because there was no easy way to scale gravity, this approach was abandoned, and a computer simulation was developed.

The computer simulation provided car motions, speeds, forces, and accelerations in derailments. Simple F = ma was the underlying theory.

Following successful validation of the computer model by comparison with an actual derailment, a number of results were obtained. Some are shown in Figure 6. This parametric study used as a base case:

- A consist moving at 40 mph with 61 cars trailing the first car derailed.
- All cars identical --
  43' truck centers
  160,000 lbs. gross weight on rail

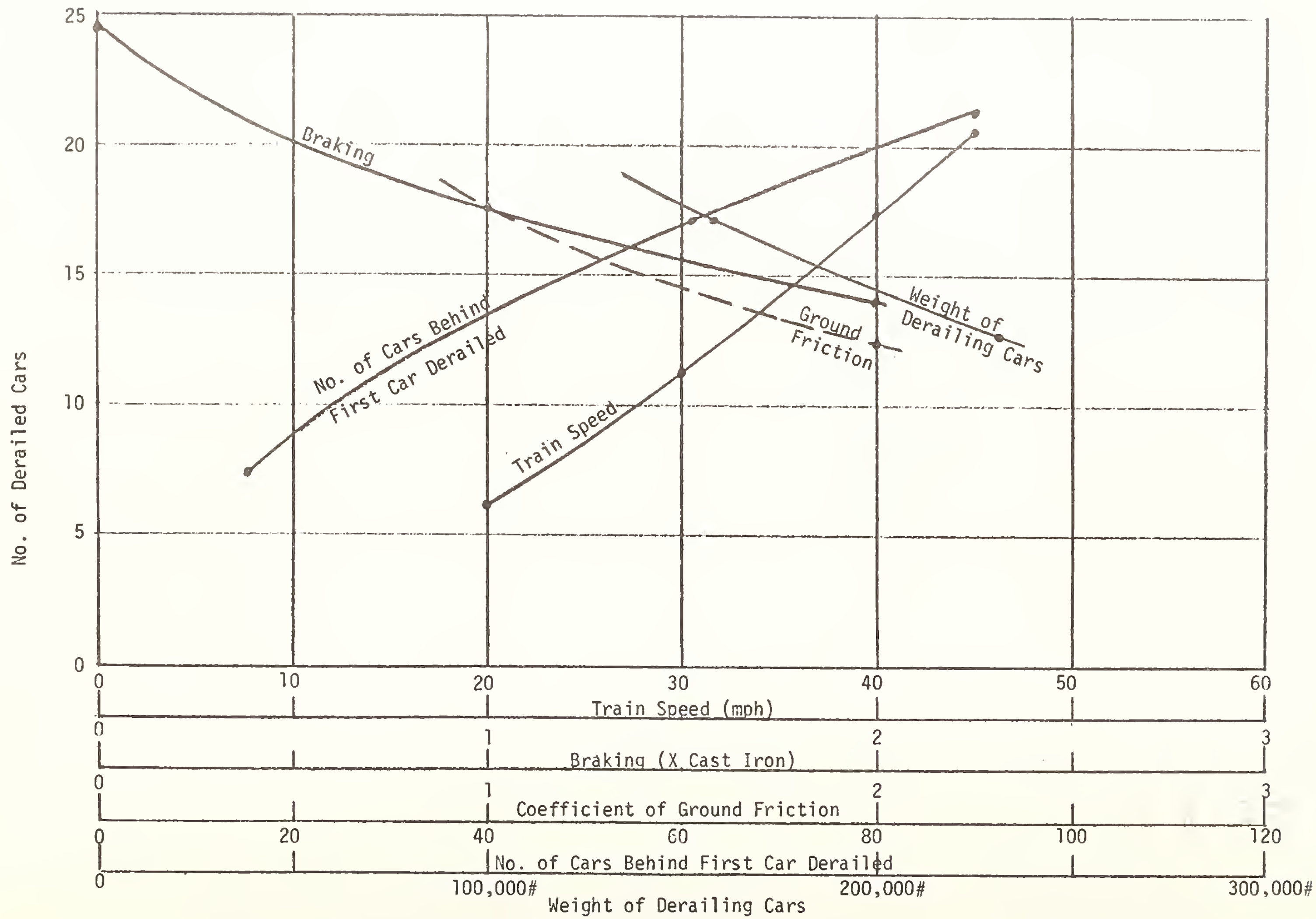


FIGURE 6 -- INFLUENCE OF VARIOUS PARAMETERS ON NUMBER OF CARS DERAILED

E couplers
Cast iron brake shoes

- o Coefficient of friction between the derailing (sliding) cars and the ground = 1.0

This base case was selected from a statistical averaging of many derailments. In this base case, 17 cars derailed, the duration of the derailment was 26 seconds, and coupler forces up to 473,000 pounds were created.

The results, from {12}, showed:

- o The influence of braking on the cars approaching the derailment site was significant, but no great reduction in the number of derailed cars could be achieved with any practical change in the braking system. For example, 17 cars derailed in the base case with emergency braking applied in the normal manner. As shown in Figure 6, 24 cars derailed with no braking. With emergency braking applied with a braking force twice that achievable with cast iron, 14 cars derailed. With emergency braking applied instantaneously to all cars (not shown in Figure 6), 16 cars derailed. Finally, as a limiting case (also not shown in Figure 6), if braking was sufficient to cause all cars to slide at the instant of derailment, 10 cars derailed.

- o The number of cars derailed increased significantly with an increase in the number of cars trailing the first derailed car (Figure 6).

- o An increase in ground friction decreased the number of cars derailed (Figure 6).

- o The number of cars derailed was almost inversely proportional to car length. This is primarily because of the large rotational inertia of the longer cars and the resulting ability to absorb more energy through ground friction.

- o Distribution of car weight also had a significant effect on the derailment. With heavy oncoming cars followed by light ones, and maintaining the same total weight as in the base case, fewer cars were derailed (Figure 6). When reversing the order of cars, considerably more were derailed.

- o When mixed cars were in the consist, the derailments were more unsymmetrical and more spread out.

- o The parameters which cause differences between passenger and freight train derailments have never been fully understood.

In cases run with the same train speed and total mass, the passenger train experienced fewer derailed cars. This effect was primarily due to car length.

o The effect of train speed was significant (Figure 6), with the number of derailing cars increasing in greater ratio than train speed.

Phase 09 - Design Study - Tanks, Attachments, and Fittings

Phase 09 concerns tank shells (portion between the heads), and attachments and fittings.

Dollar losses due to shell punctures of 112A(114A) cars have been high (Table I). Several solutions were examined {22}, such as thickening the shell, adding a protective steel jacket, and increasing the strength of the shell material. None was cost-effective. Shell punctures generally have been very severe and are caused by a variety of complex circumstances (compared to head punctures which predominantly are caused by couplers on adjacent cars). As a result, the effectiveness of solutions was difficult to estimate, and probably would be very low for any.

The most vulnerable fitting was found to be the bottom outlet on non-insulated non pressure stub sill cars (Table I). Stub sill cars do not have the full length underframe which offers some protection to the outlet. Sumps on 112A and bottom outlets on 114A cars also were responsible for large losses. Cost-effectiveness studies were made for numerous solutions in 1973-74 {24}, {27}, and several recommendations resulted. Subsequently, the "benefit" values of solutions increased significantly when new EPA spill laws appeared. The AAR Tank Car Committee has now mandated bottom fitting protection on all new cars and on a large number of existing cars. Protective "skids" or recessed fittings are required. Because of the potential for fatigue damaging stresses to occur in the tank shell adjacent to a welded-on skid, an extensive series of road and non-destructive impact tests were conducted to help develop design criteria {41}. Destructive impact tests on cars with several different skid designs also were run {41A}.

Phase 10 - Design Study, Non-Tank Portion of Car

Couplers were of prime interest under this Phase. In 1970, the type "F" interlocking coupler was required on all new tank cars. Unfortunately, it has a significant disadvantage in that it does not always protect the car on which it is installed against head punctures. It has only a bottom shelf which prevents an adjacent coupler from disengaging downward, but not upward (unless the other coupler is an F coupler). Thus, when cars bounce and disengage in a derailment, the chances are over 50% that the adjacent coupler will disengage upward and puncture the F coupler equipped car. Accident data confirmed this.

For example, during one 11 month period, 5 cars equipped with F couplers incurred head punctures, one at both ends.

Project effort was directed toward the "E" coupler with a top and bottom shelf and the "F" coupler with a top shelf. A sketch of a type E coupler is shown in Figure 7. Both of these couplers can restrict separation in either direction when mated to any type coupler. Accordingly, they will protect both adjacent cars and cars on which they are installed.

Following a successful service trial of 225 carsets of E shelf couplers and 100 carsets of F top shelf couplers, the following a number of tests {11}, analyses {19}, and accident experiences {25}, {28}, the Project recommended that all new tank cars be equipped with the E shelf coupler and all existing 112A(114A) cars be retrofitted with the E or F shelf coupler, as applicable. Changes in regulations implementing these recommendations did not occur until the shelf coupler effectiveness was further proven in a series of destructive impact tests conducted un Phase 15.

Phase 11 - Thermal Effects Study

The whole thermal question, including fire environment and thermally induced stresses, was covered under this phase.

Early in the program a theoretical study was made on the effect of fire on 112A(114A) cars {5}. This demonstrated the sequence of events from fire initiation to tank rupture.

The major activity was a search for a practical heat shield material, such as an ablative or intumescent coating, or a high temperature resistant insulative material, that could be applied to 112A(114A) cars. As Table I shows, fire induced ruptures of these cars led to the highest dollar losses of all over the 6 year base period. Many sample materials were studies in a laboratory fire test apparatus designed for initial screening {15}. They were tested on 6" diameter by 5/8" thick steel discs representing the tank shell. Thermocouples on the backside of the discs measured the temperature rise of the steel.

Two of the most promising materials were selected for application to 1/5 scale tank cars furnished by the Project for fire tests at the White Sands Missile Range in New Mexico. The tests were conducted by the U.S. Army Ballistics Research Laboratory (BRL) under contract to the FRA. The objectives were to ascertain some of the properties of fires which were not yet well defined and to prepare for subsequent full scale tests. These 1/5 scale tests are discussed in {14} and {26}.

Next, two full scale 33,500 gallon 112A340W cars also furnished by the Project, were tested in fully enveloping JP-4 fueled fires {31}. One

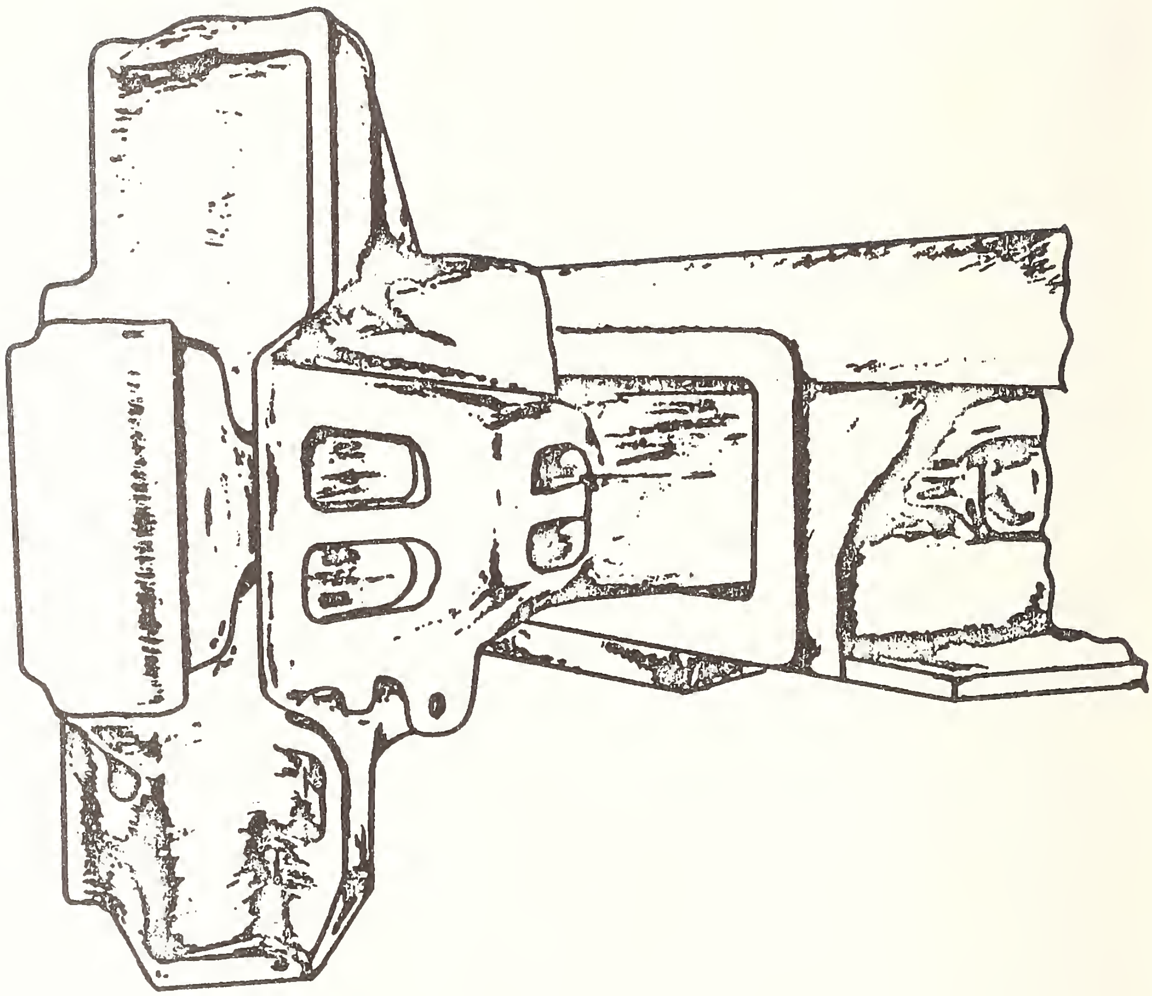


FIGURE 7 -- TYPE "E" TOP AND BOTTOM SHELF COUPLER

car was bare (standard white alkyd paint only) and the other was insulated with approximately 1/8" of a sprayed-on thermal coating. Both cars were filled with propane to standard loading conditions. The safety valve start-to-discharge pressure setting was 280.5 psig, standard for cars of this type. The temperature-time curves for two points on the top of each tank are plotted in Figure 8 from ref. {34}.

In the bare tank test, rupture occurred 24.6 minutes after ignition. The tank initially carried 31,900 gallons of propane, and more than half of this had been discharged through the safety valve prior to rupture. The tank pressure rose to 347 psig 17.5 minutes after ignition, then dropped off, ending at 334 psig at rupture.[1/] Considering that this fire was far more extreme than the typical tank car accident fire, the safety valve performance was considered quite adequate, maintaining a reasonable pressure and yielding a discharge rate closely approximating 100% of the theoretical maximum efflux rate. As expected, the fracture origin was at the top of the tank over the vapor space, and metallurgical analysis of the steel at the precise origin indicated that it had reached a temperature in the range 1050-1100 F. This time-to-rupture and temperature range at rupture is shown as the black rectangle labelled "H" in Figure 8.

Using the simple hoop stress formula, and using the ultimate tensile strength of the steel determined from high temperature tensile tests, the pressure at rupture was calculated to be in the range 301-412 psig. This "indirect" approach of estimating pressure at rupture correlated fairly well with the measured pressure. This metallurgical technique was used to determine the tank pressure at rupture and the steel temperature at the rupture origin in seven accident cases. The pressures at rupture in these cases were not significantly high (above the safety valve setting). These are shown in Figure 8 as rectangles A through G. The times-to-rupture for many other accident cases were known, but steel samples were not available for determining maximum temperatures. These cases are shown as vertical lines in Figure 8.

In the coated tank test, rupture occurred after 93 minutes. At that time the tank was nearly empty, and the rupture was considerably less violent than in the first tests, most of the propane having been discharged. Pressures were less than in the first test. Again, the temperature of the steel at the rupture origin was determined metallurgically, this time as being between 1100°F and 1160°F. This case is shown in Figure 8 as rectangle "I".

In addition to these all enveloping "pool" fire tests, the "torch" type fire was studied. This is a highly conventive fire involving local impingement as compared to the highly radiative pool fire. The torch fire studies were conducted by the BRL at the DOT Test Center in Pueblo, Colorado and are discussed in {36}.

1/ These pressures are considerably below the car's 850 psi design burst pressure.

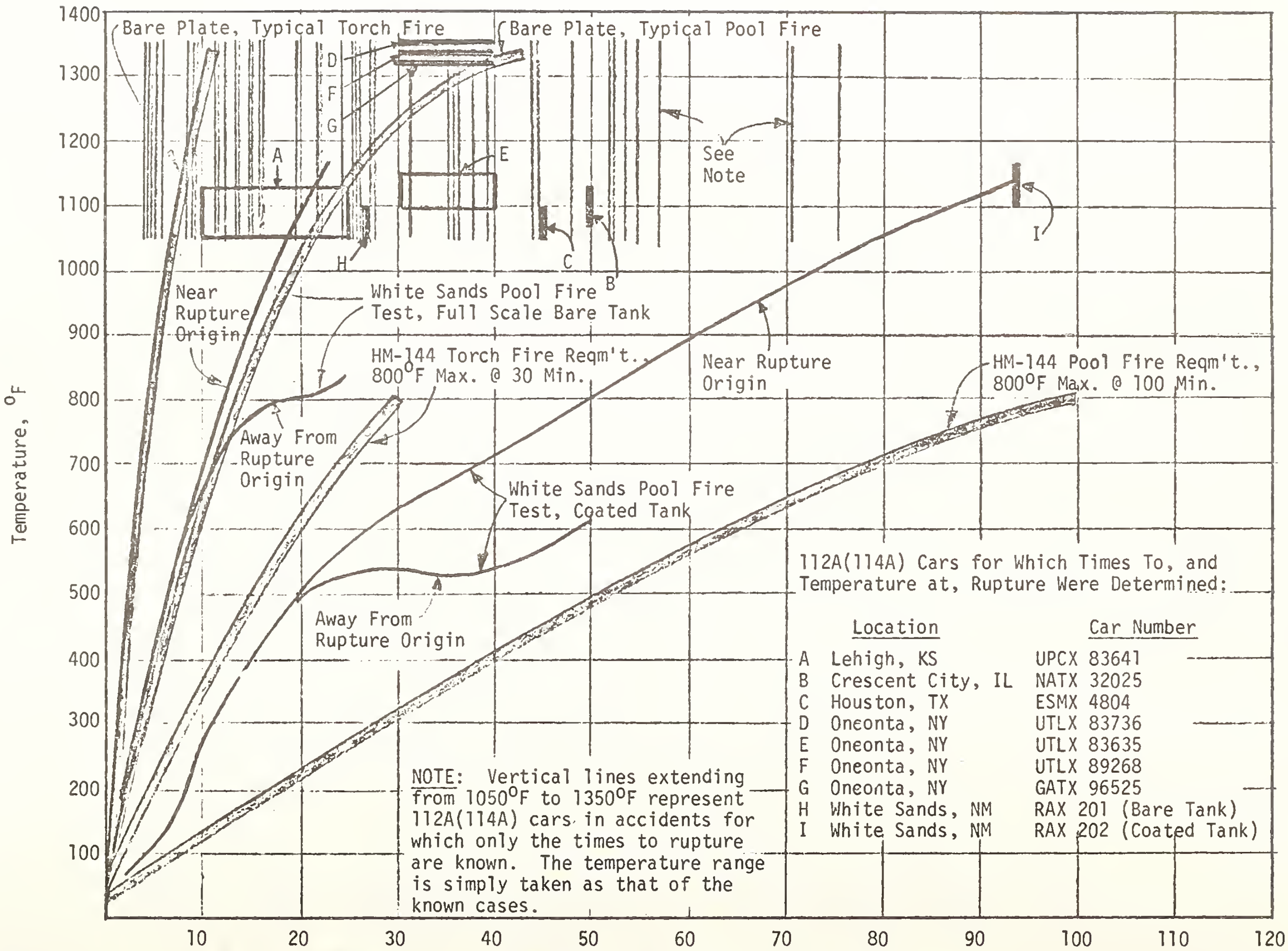


FIGURE 8 -- RUPTURES OF 112A(114A) CARS IN TESTS AND ACCIDENTS

TANK PLATE TEMPERATURE VS. TIME FROM IGNITION

Based on these pool and torch fire tests a regulation was issued (HM-144) which required thermal shielding on all new and existing 112A(114A) cars carrying flammable gas. Most cars have been shielded with a high temperature blanket type insulation covered with a protective steel jacket, but a substantial number have been covered with a sprayed-on coating.

The Pueblo torch fire apparatus ultimately was rendered capable of producing a fire of near pool fire characteristics and has been used to qualify thermal shield systems under both torch and pool fires. The two curves representing these qualification requirements are shown in Figure 8.

The study of recent accident data {44} shows that these thermal shields have led to a significant reduction in fire induced ruptures. This is predictable from Figure 8 since the two HM-144 qualification curves are far below and to the right of the accident cases. As seen, if any ruptures do occur, they most likely would be caused by a sustained torch fire. Of course, any car that loses its thermal shield during the derailment also will be vulnerable.

Phase 12 - Vessel Failure Research

This has been the basic research phase related to tank ruptures. Its objective was to explain fundamentally a number of phenomena associated with the catastrophic type rupture. This includes brittle fracture, the hurling (rocketing) of tank tub sections, and the characteristic crack propagation patterns which lead to flattened center sheets and circumferentially separated end tubs.

Battelle Memorial Institute was engaged to conduct this research. The final report is given in {20}.

Based on the Phase 03 tank car steel tests, detailed examinations of dozens of tank cars which ruptured in accidents {6}, and basic fracture mechanics theory, conclusions were reached as to the potential benefits of various changes in steel properties and fabrication techniques. These included increasing thickness and strength, normalizing, reducing grain size, and cross rolling or changing rolling direction.

None of these changes was found to be effective, and all were extremely cost-ineffective. Increased toughness could not be justified since only a few accidents involved brittle fracture, and in most the destructive forces were so great that product release probably would have occurred in any event.

Considerable theory was developed on critical crack lengths (below which crack propagation will arrest). The critical crack lengths for tank car shells and heads were very large under most accident conditions. This explains why so many punctures remained local. In the case of thermally overheated tanks, the study showed that "practical" changes in material properties and thickness, and in tank diameter, would have little effect on the critical crack lengths; therefore, they would not be effective in alleviating the violent rupture problem.

A significant finding was that in every known case of a fire induced rupture of the flammable compressed gas car, the crack originated in the vapor space where the steel was overheated. The seven accident cases and the two test cases labelled A through I in Figure 8 confirm this. Other accident data also confirmed this {1}, {7}, {9}. High overpressure caused by inadequate safety valve capacity has not turned out to be the basic cause of rupture.

Battelle also examined the "rocketing" phenomenon {23}, but no cost-effective solution to prevent it was found {20}.

Phase 13 - DOT(FRA) Head Shield Study

This Phase covered the "conceptual" head shield design study carried out under contract to the FRA, see discussion under Phase 05.

Phase 14 - Stub Sill Tank Car Buckling

Tank shell buckling has occurred inboard of the stub sill termination on certain designs of non-pressure stub sill cars. The tank of a stub sill car carries the train loads, and these cases of buckling have occurred under high compressive train action and sometimes during switchyard impacts. The problem has been limited to empty cars. When stub sill cars are loaded, the tensile stress in the bottom fibers of the tank produced by the lading weight is sufficient to offset the otherwise critical compressive stress.

The object of this Phase was to determine what design and test loads should be specified for stub sill cars to assure that their resistance to buckling is at least as good as that of other freight cars.

Tests involved static squeeze and dynamic impacts of 10 stub sill cars of different design, 5 of which experienced a history of buckling and 5 of which were of new and improved design. Approximately 80 strain gage rosettes were employed on each car.

The results and conclusions from this extensive test program are given in {38}. While interesting from the standpoint of experimental stress analysis techniques, it will suffice here to simply reference the report.

Phase 15 - Switchyard Impact Tests

Several catastrophic switchyard accidents involving tank car head punctures have occurred. These constitute the third accident scenario given at the beginning of this report. They involve the striking of a light empty freight car by one or more loaded cars. In some cases the lead loaded car has been a tank car, and it is punctured by the coupler of the empty car. In other cases, the empty car is driven into and punctures a loaded tank car standing ahead of it.

To study these accidents, a full scale test program was undertaken in cooperation with the FRA and carried out at the Pueblo test center. Single empty hopper cars were impacted by loaded tank cars at destructive speeds and driven into other loaded tank cars. The objective was to assess the ability of the shelf coupler, the head shield, or both in combination to prevent punctures in this particular accident scenario.

Results are given in {40} and are summarized herein in Table II. The last four tests were particularly severe because the number of impacting cars was increased from 1 to 3. The last test was the severest because the speed was high (17.1 mph), the spacing "S" was 3.75 ft., an optimum distance for override, and the internal pressure in the test cars was 125 psig (vs. 100 psig in all other tests).

These Phase 15 tests were considered final proof of the effectiveness of the head shield, and particularly the shelf coupler, in preventing head punctures.

Phase 16 - Component Wear Study

Wear rates of various running gear components on tank cars were measured under this Phase. This had no relationship to tank car behavior in accidents so is not described in this paper.

Phase 17 - Class 105A Car Study

The 105A tank car is an insulated pressure car. The insulation specifications are based on protecting the commodity from ambient temperature conditions and not from fire. Class 105A cars are authorized to carry many of the same products that the (now retrofitted) 112(114) cars are authorized to carry. New 105A cars carrying these products have recently been required to have the 112(114) car type protection; but an inconsistency exists with existing cars. The objective of Phase 17 is to assess the significance of the inconsistency.

Initially, a population study and accident data review was carried out to compare various 105A cars and to compare them with 112A(114A) cars (before retrofit) {43}. The 105A cars as a group were found to be significantly less vulnerable to punctures and fire induced ruptures

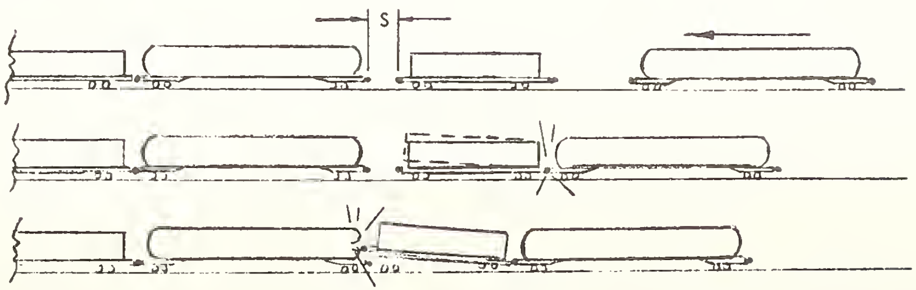
TABLE II -- PHASE 15 SWITCHYARD IMPACT TEST RESULTS

Impact Mechanism (See Sketch)	Date	Impacting Car(s) Number	Impacting Car(s) Velocity	Impacting Car(s) Head Shield	Impacting Car(s) Coupler Type	Standing Car Head Shield	Standing Car Coupler Type	Spacing ("S")	Standing Car Damage	Impacting Car Damage	Remarks
Double	04-15-76	1	12.7	No	Std.'E'	No	Std.'E'	2.5'	Dent	None	
Double	04-29-76	1	14.9	No	Std.'E'	No	Std.'E'	3.5'	Dent	None	
Double	05-06-76	1	16.5	Yes 1/	Std.'E'	No	Std.'E'	3.5'	Dent	Dent	
Double	05-20-76	1	15.4	No	Std.'E'	No	Std.'E'	2.5'	Dent	Dent	
Double	05-27-76	1	16.9	No	Std.'E'	No	Std.'E'	2.5'	Punct.	Dent	Puncture at head reinforcing plate.
Double	07-08-76	1	18.6	Yes 1/	Std.'E'	Yes 1/	Std.'E'	2.5'	Dent	Punct.	Puncture at top of shield.
Double	08-05-76	1	19.4	No	'E'Shelf	No	'E'Shelf	2.5'	None	Dent	
Double	08-17-76	1	19.0	No	'E'Shelf	No	'E'Shelf	3.75'	Punct.	Scrape	Small tear 18" above sill.
Double	09-02-76	1	19.2	Yes 1/	Std.'E'	Yes 1/	Std.'E'	3.5'	Dent	Dent	Impact on both head shields.
Dynamic 3/ Squeeze	09-17-76	3	18.1	No	Std.'E'	No	Std.'E'	0.0'	Dent	Dent	Hopper car body buckled.
Double	10-13-76	3	14.0	No	Std.'E'	No	Std.'E'	2'11"	Dent	Punct.	Puncture 1 ft. above sill; dent @ head reinf. plate.
Dynamic 3/ Squeeze	10-22-76	3	15.0	No	Std.'E'	No	Std.'E'	0.0'	Dent	Dent	
Double	11-23-76	3	17.1	Yes 2/	'E'Shelf	Yes 2/	'E'Shelf	3.75'	Dent	Dent	

1/ Trapezoidal Type

2/ 1/2" Jacket Type

3/ Dynamic Squeeze Occurs When "S" = 0



than 112A(114A) cars. Class 105A cars carrying ethylene oxide experienced a higher rate of fire induced ruptures than other 105A cars. Class 105A chlorine cars were found to perform particularly well, and 105A cars in vinyl chloride service were found to be particularly vulnerable to head and shell punctures.

In another activity, mechanical impact tests and fire tests were conducted on 2 ft. square and 4 ft. square specimens representing several common 105A car tank-insulation-steel jacket constructions. As in other Phases the tests were carried out by the BRL under contract to the FRA and in cooperation with the Project which furnished the test specimens. Fire tests were run with the steel jacket bare and then covered with various thermal coating systems, the concept being directed toward shielding existing cars. At this time the results have not been analyzed. Even if cost-effective, it remains to be demonstrated that the coatings will maintain their integrity when applied to the thin (11 gage) steel jacket.

Phase 18 - Integrity of Damaged Pressure Cars

A risk situation arises when a tank car, particularly a 112A(114A) or a 105A pressure car, is damaged in an accident but does not release its lading. The wounded tank suffers a reduction in its burst strength. If the pressure rises above that which existed at the time of the accident, it may approach or reach this reduced burst pressure. Personnel involved in wreck clearing operations can be exposed to great danger under this condition.

The plan under this Phase is to hydrostatically test to burst a number of wounded tank cars which did not lose lading in accidents. If a correlation can be developed between the reduction in burst pressure and the type of damage, possibly some guidelines will emerge which will help protect workers at the scene. Work under this Phase has just begun.

VIII. CLOSING REMARKS

Major activities have been directed toward reducing risk through design changes. A number of correlary benefits have emerged from the program. The effectiveness (and efficiency) of carrying out research on a cooperative basis between the railroads, shippers, tank car builders and the government has been demonstrated. As new situations or questions arise, means now are available to promptly study and resolve them. Of particular interest will be a report to be issued shortly which will present all of the accident data in the Project files over the 16 year period 1965 through 1980. These data will be invaluable in analyzing many risk situations. It is not possible to predict all future uses of the data, but experience has shown that problem areas do continually arise. Some are unexpected and result from accidents involving a new phenomenon. For this reason, the collection and analysis of accident data is being carried out on a continuous basis.

REFERENCES: Following is a chronological list of reports to date by the RPI-AAR Tank Car Safety Research and Test Project. Copies may be ordered from:

Mr. J. G. Britton, Executive Director
AAR Technical Center
3140 S. Federal Street
Chicago, Illinois 60616

Prices include mailing costs within the United States.

Ref. No.	T.C. Report No.	Title	Date	Price
1	RA-01-1-1	Phase 01 Report on Sequence of Events Following Crescent City Derailment	8/19/70	$ 2.50
2	RA-03-1-2	Phase 03 Report on Metallographic Examination of Specimens Removed from Sections of Tank Cars in Crescent City Derailment	10/15/70	$ 2.50
3	RA-01-2-3	Phase 01 Report on Summary of Ruptured Tank Cars Involved in Past Accidents (Superseded by RA-01-2-7)	12/4/70	--
4	RA-03-2-4	Phase 03 Report on Metallographic Examination of Specimens Removed from a Section of Tank Car in South Byron, New York Derailment	3/10/71	$ 1.20
5	RA-11-1-5	Phase 11 Report on Effects of Fire on LPG Tank Cars	4/19/71	$ 7.50
6	RA-12-1-6	Phase 12 Report on Tentative Tank Car Accident Investigation Guide (Revised 4/26/73)	4/26/73	$ 1.20
7	RA-01-2-7	Phase 01 Report on Summary of Ruptured Tank Cars Involved in Past Accidents (Revised 7/1/72)	7/1/72	$ 3.80

Ref. No.	T.C. Report No.	Title	Date	Price
8	RA-04-1-8	Phase 04 Report Review of Literature and Related Experience	7/30/71	$ 5.00
9	RA-01-3-9	Phase 01 Report on Sequence of Events Following Houston, Texas Derailment	10/19/71	$ 1.20
10	RA-02-1-10	Phase 02 Report on Dollar Loss Due to Exposure of Loaded Tank Cars to Fire - 1965 Thru 1970	2/23/72	$ 2.50
11	RA-10-1-11	Phase 10 Report on Determination of Moment Characteristics in a Horizontal Plane of Mated Combinations of Types "E" and "F" Couplers	2/15/72	$ 1.20
12	RA-08-1-12	Phase 08 Report on Computer Derailment Study	2/17/72	$ 3.80
13	RA-03-3-13	Phase 03 Data Display Report on Results of Tests on Mill Plate Samples and Samples Removed from Tank Cars Involved in Accidents (Superseded by RA-03-5-33)	4/19/72	$ 6.20
14	RA-11-2-14	Phase 11 Report on Analysis of 1/5 Scale Fire Test Data	4/12/72	$ 2.50
15	RA-11-3-15	Phase 11 Preliminary Report on Laboratory Fire Test Apparatus for Evaluating Thermal Shield Materials	5/24/72	$ 3.80
16	RA-01-4-16	Final Phase 01 Report on Accident Review	6/30/72	$ 3.80
17	RA-05-1-17	Final Phase 05 Report on Tank Car Head Study	7/14/72	$ 6.20
18	RA-02-2-18	Final Phase 02 Report on Accident Data Analysis	8/14/72	$12.50

Ref. No.	Report No.	Title	Date	Price
19	RA-10-2-19	Final Phase 10 Report on Couplers and Truck Securement	8/14/72	$ 2.50
20	RA-12-2-20	Final Phase 12 Report on Tank Car Fracture Analysis	9/29/72	$10.00
21	Not Issued			
22	RA-00-1-22	Overall Project Summary Report	10/6/72	$ 3.80
23	RA-12-2-23	Phase 12 Report on Analysis of Tank Car Tub Rocketing in Accidents	12/13/72	$ 2.50
24	RA-09-1-24	Final Phase 09 Report on Tanks, Fittings and Attachments in the Mech. Environment of Accidents	2/15/73	$ 5.00
25	RA-10-3-25	Phase 10 Report on July 1, 1973 Accident Involving Type "E" Top and Bottom Shelf Couplers	11/16/73	$ 2.50
26	RA-11-5-26	Phase 11 Report on Analysis of 1/5 Scale Fire Tests	12/12/73	$ 5.00
27	RA-09-2-27	Final Report on Phase 09 Extension Study of Tank Car Bottom Fittings	5/31/74	$11.20
28	RA-10-4-28	Phase 10 Report on February 9, 1974 Accident, Involving Type "E" Top and Bottom Shelf Couplers	4/22/74	$ 5.00
29	RA-05-2-29	Phase 05 Report on June 9, 1974 Accident Involving Head Shields	8/28/74	$ 1.20
30	RA-10-5-30	Phase 10 Report on Development of Shelf Couplers	9/6/74	$ 5.70
31	RA-11-6-31	Phase 11 Report on Full Scale Fire Tests	12/12/75	$11.00
32	RA-03-4-32	Phase 03 Report on Fracture Properties of Tank Car Steels - Characterization and Analysis	8/20/75	$ 7.50

Ref. No.	T.C. Report No.	Title	Date	Price
33	RA-03-5-33	Final Phase 03 Report - Material Study on Steels Used in Current and Former Tank Car Construction and From Cars Involved in Accidents	8/21/75	$20.00
34	RA-11-7-34	Phase 11 Report on Specifications for Thermal Shield Systems on DOT 112A(114A) Tank Cars	1/23/76	$11.20
35	RA-05-3-35	Phase 05 Report on Head Shield Fatigue Tests	11/10/75	$ 4.00
		Technical Progress Reports Nos. 1 through 31	5/28/70 Thru 6/6/75	$ 1.00 each
36	RA-11-8-36	Evaluation of RPI-AAR and BRL Torch Fire Tests of Tank Car Insulations	9/9/76	$ 2.00
37	Not Issued			
38	RA-14-1-38	Final Phase 14 Report on Stub Sill Buckling Study	7/29/77	$20.00
39	RA-11-9-39	Phase 11 Report on Inspections of Insulation-Jacket Type Thermal Shields on Tank Cars in Accelerated Life Tests	1/13/78	$ 2.00
40	RA-15-1-40	Phase 15 Report on Switchyard Impact Tests	8/2/78	$ 5.00
41	RA-09-4-41	Phase 09 Final Report on Bottom Fittings Protection Test Program	12/22/78	$ 5.00
41A	RA-09-4-41A	Addendum to Phase 09 Final Report on Bottom Fittings Protection Test Program	4/23/79	$ 4.00
42	RA-09-5-42	Study of Bottom Discontinuity Damage on Non-Pressure Stub Sill Tank Cars in Derailments During 1977 and 1978	2/14/80	$ 5.00

Ref. No.	Report No.	Title	Date	Price
43	RA-17-1-43	Phase 17 Report on Study of Class 105A Tank Cars	8/22/80	$ 4.00
44	RA-02-3-44	Phase 02 Report on Effectiveness on Shelf Couplers, Head Shields and Thermal Shields	5/15/81	

PERFORMANCE TESTING TO
REDUCE LOSS AND DAMAGE

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Abstract: Performance testing of shipping containers can help reduce loss and damage by increasing the efficiency of package development. The cumulative effect of successive handlings can be predicted in the laboratory. This insures a higher probability of success during test shipments in the field.

In 1980 damage claims to railroads amounted to $324.5 million. The staff of the Damage Prevention Section of the Association of American Railroads is working to reduce these claims through educational programs, field studies, field testing, and laboratory evaluations. Educational programs for various commodities are prepared after extensive field studies. Field tests and laboratory evaluations try to determine the best methods of packaging, unitizing, loading, and bracing for rail shipment.

When conducting field studies, we take special note to handling of products prior to rail shipment. A slightly damaged product placed in a rail car may be further damaged by normal forces occurring in transportation. For example a fatigued container in the bottom layer could result in breakdown of part of a load in a rail car. The result could be further damage to adjoining containers.

Impact tests and field tests are conducted on fully loaded rail cars. Impact tests approximate the shocks a rail car may experience while moving through several rail yards. Up to forty of these tests on closed cars are conducted each year by the AAR. Fifty percent of the impact tests are successful and move to the field test stage. Field testing eliminates another fifty percent leaving approximately ten successful loading and bracing methods for publication.

Tests for new unitizing and packaging methods follow similar procedures for approval. Extensive field testing is conducted to determine if a method is suitable for rail shipment. The Uniform Classification Committee is responsible for these evaluations. They have done an outstanding job of monitoring the test permit packages. For the benefit of the rail industry and their customers we constantly look at methods to improve evaluations of packages. Two methods are emerging as a means of improvement.

The first is the establishment of a standard packaging code so that rail inspectors can include package conditions in their claim report. These conditions would be included with claim data submitted to data systems of the railroads. Damage to products could then be sorted by package type and damage. Presently a simplified packaging code is optional for claim reporting. The result is very limited reporting of package condition. We hope that in the future more extensive use will be made of this optional packaging code.

The second means of improved evaluation of damage to shipping containers is performance testing. Performance tests can be used to simulate forces occurring in the distribution environment. Railroads have long recognized the importance of performance testing. In 1919, the U.S. Railroad Administration met at the Forest Products Laboratory in Madison, Wisconsin to establish standards for containers based on revolving drum tests. The experiences of World War I had shown the need for performance test procedures to supplement material specifications. Again, following World War II, the shortcomings of material specifications were noted and renewed efforts were made to develop useful preshipment performance tests. In 1948, the National Safe Transit Committee (NST) was formed and the pre-shipment test procedures were developed. These procedures are probably the most widely used of any at this time.

During the past eight years the Committee D-10, on Packaging, of the American Society for Testing and Materials (ASTM) has been developing a performance test for shipping containers. At the AAR Container Laboratory in Chicago we have extensively evaluated this test procedure. By adjusting test sequences and levels we were able to approximate shipping damage in many areas. We have found the test most useful for comparative testing purposes. For these reasons we believe that performance testing will be an important supplement to material specification tests.

For many years, all of our standards for shipping containers have been based on specifications of materials. The carrier shipping rules are based almost entirely on material specifications. These specifications have merit of course, because they establish a minimum level of quality that the manufacturers of packaging materials must meet. However, methods of material handling have changed, the transportation environment is continually changing, and new and cost conscience packaging methods and materials are continually being developed. Use of these new methods and materials may be restricted until their adequacy has been determined.

Current methods of testing involve lengthy periods of trail shipments and evaluation. While field testing is eventually necessary for all containers, it is not always the best way to begin evaluating containers. Performance testing can enhance field test programs by assuring a higher probability of successful tests.

The purpose of performance testing is to provide a rational basis for the evaluation of packaging materials and methods, and to provide a means to determine levels of packaging that are both adequate and economical. Preshipment performance testing should enable the packaging engineer to save time and money and to reduce loss and damage. It will help reduce excess costs of overpackaging, reduce costs of new package development, and reduce total costs of trial shipments, while improving chances of success. Inasmuch as the transportation regulatory agencies recommend and usually require preshipment testing to support the application for test shipment permits, the results obtained from these performance test procedures should improve the shippers' chances of obtaining the test permit and the desired classification.

This proposed ASTM practice is based upon the concept that each mode of transportation or portion of the distribution cycle has its unique set of hazard elements that a shipping container must endure and survive, regardless of the nature of its contents. The practice provides a uniform basis for evaluating, in a laboratory, the ability of shipping containers (or shipping units) to withstand particular distribution environments. This is accomplished by subjecting the containers to a test plan consisting of a sequence of anticipated hazard elements encountered in various distribution cycles. Each identified hazard element is simulated by an established ASTM test method. The tests are conducted in sequence on the same container or lot of containers making up the shipping unit, which is the smallest complete unit that will be subjected to the distribution environment. The recommended test levels are based on the best available information on the shipping and handling environment and current industry practice and experience.

Test levels are adjusted on the basis of probability of occurrence of the hazard and the degree of assurance desired that the package and product will not be damaged in shipment. Three recommended test levels are provided so that the user can determine the appropriate "assurance level" for his product. Level I is a high level of test intensity. If the shipping container passes this test level, it will have a low probability of being damaged in shipment. Level III is the lowest level of test intensity, but the container that passes this test would then have a higher probability of being damaged in shipment (thus a lower assurance level of safe shipment). Level II is midway between the other two test intensity levels and would be most commonly used.

Acceptance criteria for the condition of the package and the product at the end of the test must be established prior to testing. The organizations conducting the test must agree on the acceptance criteria, but may choose any criteria suitable to their purpose. Usually, the criteria that the product is damage-free and the package is intact would be used, but other criteria for both package and product may sometimes be desirable.

The shipping environment is categorized in 14 different distribution cycles, each of which has its own sequence of hazard elements and associated standard test methods. These cycles are based on air, truck, rail and combinations of these means of transportation. It is considered that the categorization of distribution cycles used is adequate but not excessive. The format of the proposed standard is such that additional distribution cycles and hazard elements can easily be added as new information is obtained or new modes of distribution developed. The proposed standard is presently limited to domestic common carrier shipments, but export and other modes may be added in future revisions.

Seven hazard elements have been defined, including handling shocks, three forms of vibration, vehicle impacts and compressive loading forces experienced in stacking. Standard ASTM test methods are used to determine the ability of the shipping unit to withstand these hazards.

For an example of how to use this performance test, consider a new industrial package unitized on pallets. The product is shipped by rail directly to the end user. Handling occurs at the production point and the warehouse of the user. We would like to determine if the package can survive the "complete distribution cycle".

Utilizing the ASTM performance test, choose a distribution cycle related to rail shipment of unitized products and an assurance level corresponding to the product value and amount of damage tolerated. Several hazards are identified which are represented by ASTM test procedures conducted in a series.

The first test is a mechanized handling test, consisting of a rotational impact on each base edge of the pallet from a height of six inches. The second test in sequence is a compression test, simulating vehicle stacking. Containers with product are loaded in top-to-bottom compression in a compression testing machine. The test load value is determined by the superimposed load encountered in vehicular stacking, multiplied by a factor to account for the response of the container material to possible creep, moisture, various temperature environments, and other derating factors. When the load value is reached, it is held momentarily and then released .

The third test in sequence is a stacked vibration test. The unitized pallet load is placed on the vibrator and a concentrated dead load is placed on top to equal the weight of the upper tier of the load as stacked in the rail car. The shipping unit is then vibrated at 0.3 "G" over the frequency range from 3 to 100 hertz. Any stack resonances are noted and a dwell time of ten minutes is imposed at each resonant frequency. For test purposes in the AAR Container Laboratory, we sweep between 2 and 25 hertz, dwelling at any stack resonance between 2 and 10 hertz.

Each package or unit is then subjected to horizontal shocks. These shocks are designed to approximate the shock pulse of a standard draft gear rail car at six (two impacts) and four (one impact) miles per hour (these shocks are at the medium assurance level). This is one area where a direct correlation between field testing and lab testing has not been completely established.

The fourth test sequence is another handling test similar to the first one, and the final test is a warehouse stacking test similar to the vehicle stacking test, but with the imposed loads determined by expected warehouse stacking heights.

At the conclusion of the testing, the products and packages are inspected to determine if the acceptance criteria have been met and the report is prepared.

It should be noted that these tests are nondestructive when the level of packaging is adequate for the distributioin cycle used.

The AAR Container Laboratory began use of a modified version of this ASTM performance test earlier this year. A recent test involved a sizable claim against a railroad involving plastic bottles of vegetable oil. The railroad asked the Container Laboratory to determine if the plastic bottles were subject to vibration (abrasion) and impact damage. A performance test was designed to simulate warehousing as well as transportation. The test was much more severe than would normally be found in rail transit. The end result of the test was that container failure did not occur under laboratory conditions. The shipper and carrier are now investigating other causes of the damage problem.

Another example of use of this test involved an evaluation of food products. A rail carrier requested a performance test for a food manufacturer who distributed solely by truck. One product in particular was subject to damage during long distance transportation. A test series was designed to approximate both rail and truck shipment. All test packages performed well in both distribution cycles with one important exception. The damage prone product had a higher incidence of failure during vibration at frequencies normally associated with truck transportation. At the lower rail frequencies damage occurred, but the time frame was three times longer than at higher frequencies. As a result, rail test shipments are now underway.

These examples should show how performance test procedures can be used to determine adequacy of products, packaging and optimum packaging levels.

Performance test procedures are not intended to replace existing material specifications or established preshipment test procedures. It is not to be used with hazardous materials, which must be packaged to meet other rigorous rules that include specific performance requirements, based on the properties and characteristics of the hazardous materials.

Use of performance testing can help reduce loss and damage to products by increasing the efficiency of package development. The AAR Container Laboratory is continuing studies of the best way to apply performance testing and further improvement of existing test methods.

FIELD MEASUREMENT OF THE FATIGUE LOADING ENVIRONMENT FOR RAILROAD FREIGHT EQUIPMENT

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ABSTRACT

Railroad freight equipment in North America operates in a broad range of conditions for up to four decades. Consequently, the equipment must be capable of surviving long term, fatigue inducing loading in addition to short term structural loads.

In order to adequately design for fatigue life, a thorough knowledge of the loading environment that revenue freight traffic experiences during its lifetime is necessary. To obtain this information, a test program was developed to measure loading input into the freight car structure during over-the-road revenue service for extended periods of time.

The program used a stratified random sampling technique to develop statistically valid representation of the loading and utilized a self contained portable data aquisition system to record and store the measured load levels. It is the purpose of this paper to describe both the data sampling techniques developed to define representative load environments, and the actual measurement system and hardware used to obtain the specific load data.

SESSION V

DESIGNING FOR TRANSPORTATION OF HAZARDOUS MATERIALS (NUCLEAR MATERIALS AND WASTES)

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TRUCK AND RAIL SHOCK AND VIBRATION ENVIRONMENTS DURING NORMAL TRANSPORT

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Abstract: The Sandia National Laboratories Transportation Technology Center (SNL/TTC) provides technical management and support for programs which span a broad range of technical problems related to the transport of nuclear materials.

As part of this program, SNL has obtained data on the shock and vibration experienced by large shipping casks during normal transport. These data were initially generated under contract to the U. S. Nuclear Regulatory Commission (NRC) by:

1) Extracting, reviewing, and reducing the shock and vibration environment definitions on file in the DOE/DOD and DOE Transportation Data Banks,

2) Determining the best estimates of the environments for large shipping containers on trucks and railroad cars,

3) Defining and conducting tests to obtain any additional data required,

4) Conducting dynamic analyses of the shock environment experienced by cargo during rail switching and coupling to identify the dependence of the shock environment on heavy cargo weights and on shock attenuation couplers, and,

5) Using the results to refine the estimates of the shock load description.

As part of the follow-on activity at the TTC, sponsored by the U. S. Department of Energy, two approaches for using the data to estimate the shock and vibration environments were investigated. The first is a shock spectra approach which assumes that there is a single-degree-

*This work sponsored by the U. S. Department of Energy under Contract DE-AC04-76DP000789 and U. S. Nuclear Regulatory Commission under Contract A-1049-6.

of-freedom dynamic interaction between the cask, tie-downs, and vehicle bed. The second, or direct measurement approach, assumes that the cask is rigidly attached to the vehicle bed and consequently, for low frequency excitations, no such interaction takes place. Based upon the results of these activities, the need for additional data was identified and tests were planned to obtain the required data. Details and results of the overall program are presented.

Key Words: Cargo tie-down design guide; over-the-road/rail dynamic measurements; rail car coupling tests; shock and vibration data; transportation environments for radioactive material shipping containers.

Introduction

The Sandia National Laboratories Transportation Technology Center (SNL/TTC) provides technical management and support for programs which span a broad range of technical problems related to the transport of nuclear materials. One of these programs involves the gathering of shock and vibration data for the normal transportation environments of large radioactive material shipping containers. These data will be used to formulate criteria for the design of restraint, or tie-down systems for such containers. Data acquisition has been underway at the SNL for several years. The application of the existing data to heavy cargo was originally funded by the U.S. Nuclear Regulatory Commission (NRC). Under the original NRC agreement, SNL was to work on the following tasks:

1) Extract, review, and reduce the shock and vibration environment definitions on file in the Department of Energy/Department of Defense (DOE/DOD) and DOE Transportation Data Banks,
2) Determine the best estimates of the environments for large shipping containers on trucks and railroad cars,
3) Define and conduct tests to obtain any additional data required,
4) Conduct dynamic analyses of the shock environment experienced by cargo during rail switching and coupling to identify the dependence of the shock environment on heavy cargo weights and on shock attenuation couplers, and,
5) Use the results to refine the estimates of the shock load description.

The TTC intends to continue the shock and vibration investigation by building on the work completed under the NRC contract.

This paper will define the problem as currently perceived by the TTC, review the NRC contract work, and finally discuss proposed future work to be completed.

Problem Definition

There are many problems, both technical and administrative, associated with trying to define tie-down loadings for nuclear material transportation systems.

The administrative problems stem from trying to conduct realistic overthe-road type tests. Negotiations must be conducted with both the shipper and the carrier so that necessary waivers are obtained and laboratory personnel are allowed to participate in the shipment. Preparations for the tests described later in this report took from a few months to approximately 2-1/2 years, depending on the type of test and number of agencies involved. However, once the tests were set up, the cooperation of the carriers and their employees was excellent and the measurements were made in a relatively short amount of time.

The technical difficulties arise from the wide variety of hardware configurations which must be considered. Each configuration has many parameters which affect the loads seen by the cask restraints. Some of the important parameters are listed below:

1. cask mass
2. tie-down stiffness
3. tie-down damping
4. vehicle mass
5. vehicle structure stiffness
6. vehicle suspension stiffness
7. vehicle suspension damping
8. speed
9. tire pressure (for trucks)

If only maximum and minimum values are considered for each of the nine parameters, 512 separate configurations must be evaluated through testing or analyses or both. Furthermore if an average value is considered in addition to these extreme values, 19,683 evaluations are required. Clearly the problem must be reduced to manageable proportions. This is done by making worst case assumptions about some of the parameters. For instance, one might assume that the worst shock input will occur when the tires are at maximum pressure and the vehicle is traveling at some maximum speed. It is believed that these assumptions can be made for tie-down and suspension damping, tire pressure, and speed. If this is indeed the case, the number of evaluations can be reduced to 32 if only two values per parameter are considered or 243 for three values.

However, this is still an appreciable problem since the evaluations must be made in three directions (vertically, longitudinally, and laterally) and for at least two, and possibly more container shipping orientations. Although testing provides accurate measurements of loadings for specific configurations, this alone is not feasible from a time and cost standpoint because of the number of tests required. Testing is also limited to existing, available systems and it is therefore difficult, if not impossible, to envelop all possible parameter values.

On the other hand, a strictly analytical approach may be risky, unless the results are correlated in some way to the real world. Consequently, a preferred approach is one which uses analyses to envelop important parameter values and then confirms the analytical results with test data. Such is the nature of the problem. The next section describes work already completed in attempting to solve the problem. The TTC's proposed approach to completing the solution is outlined in the final section of this report.

Completed Work

The DOE/DOD data bank referred to earlier (task 1), was established to make existing shock and vibration data available to both DOD and DOE designers. The transportation data which had been gathered over many years (beginning in the early 1960s) were reviewed in 1975[1] to fulfill the requests in tasks numbered 1 and 2. In using these data, it was recognized that they had been accumulated for purposes of defining the shock and vibration environments during transport of relatively lightweight cargo. Truck data covered cargo weights from essentially no-load to 14 tonne (15 tons). Over-the-road rail data were available for 14-tonne (15-ton) cargo. Rail coupling data were available for cargo weighing about 4.5 tonne (5 tons). Since spent fuel casks can be much heavier than these weights, it was obvious that additional data needed to be gathered to aid in properly defining the dynamic transportation environments. The needed data were those of shock and vibration during normal rail and truck transport of heavy cargo and for rail coupling data when heavy cargo is involved.

Over-the-Road Truck Tests

An opportunity to gather data during truck transport was presented when SNL procured two spent fuel shipping casks weighing 20 tonne (22 tons) and 25 tonne (28 tons) for use in a series of vehicle-container impact tests. Both casks were to be moved over conventional highways from the DOE Nevada Test Site, which is northwest of Las Vegas, Nevada, to Albuquerque, New Mexico for the impact tests.

Each cask was mounted on a dedicated trailer. The cask support and tie-down structures were welded to structural members of the trailer. The 20-tonne (22-ton) cask was carried on an 11 m (35-foot) tandem axle trailer which was equipped with an air bag suspension system (Figure 1). The 25-tonne (28-ton) cask was carried on a 12-m (40-foot), 3-axle trailer which was equipped with a spring suspension system (Figure 2).

Both shipments were moved by similar tandem axle tractors which were equipped with "Velvet Ride" suspension systems.

Instrumentation on these trailers consisted of accelerometers mounted at the base of the cask support structures. The accelerometers were mounted to measure excitations in the longitudinal, transverse, and vertical directions. The leads from the accelerometers were connected to the data acquisition system which was mounted on the trailer.

Data gathering was accomplished by using a SNL developed data acquisition system. This system is self contained in that it can be operated without any external power supply. The system has the capability of recording 14 channels of data. It can receive, condition, and record signals from both piezoelectric (high frequency) and piezoresistive (low frequency) accelerometers. The conditioned signal is recorded on magnetic tape which is one-inch wide and is compatible with SNL's data reduction equipment. At 19 cm/sec (7-1/2 in/sec) recording speed, which provides data up to 2500 HZ, the system can record about 30 minutes of real time data. This limitation makes data sampling necessary.

The data acquisition system can be started and stopped by either switching at the instrument box, by hard wire connection, or by a radio control transmitter which permits taking data samples from a remote location up to about 0.8 km (1/2 mile) away.

The data acquisition system contains a standard interange instrumentation group (IRIG) B time code generator from which a signal can be placed on the data tape for use in identifying events during data reduction.

The route over which the shipment was made was determined by the commercial carrier through their normal routing procedures. Locations where measurements were to be taken for these tests were determined by making preliminary road surveys and by identifying the locations along the highway to be sampled. The locations identified were bridges, cattle guards, rough highways, as well as different types of highways.

During the shipment, data samples were gathered by remotely controlling the data acquisition system from a vehicle carrying Sandia personnel. This vehicle followed the truck carrying the casks and as the truck approached the predetermined location, the recorder was turned on remotely. After the event had been passed, the system was turned off.

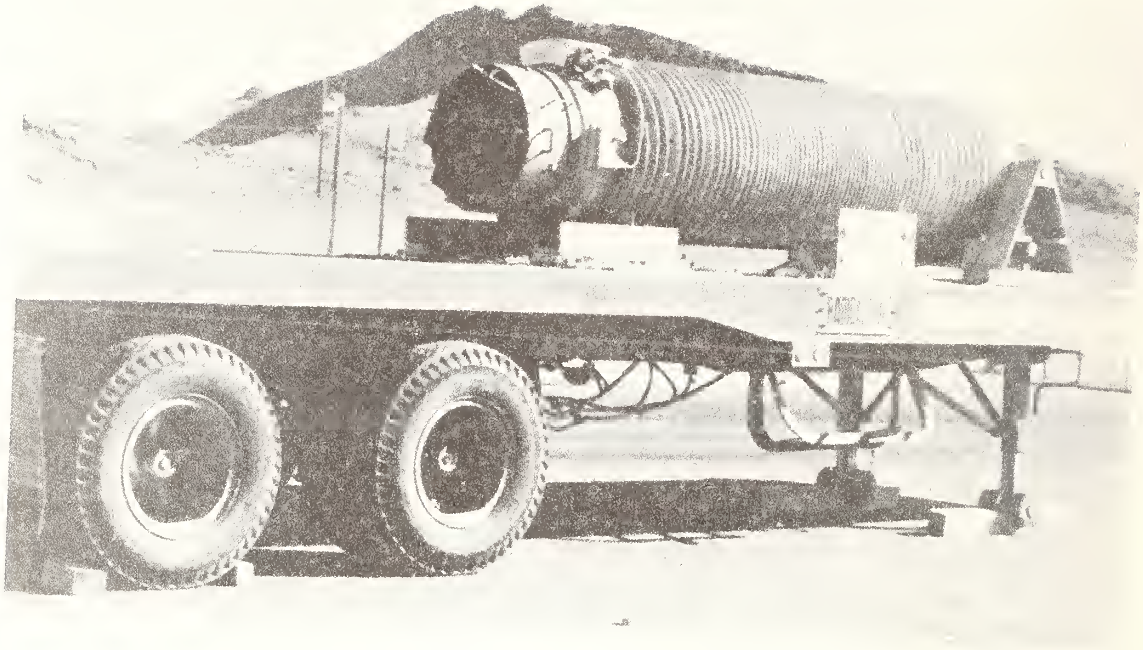

Figure 1. 20 Tonne Truck Cask

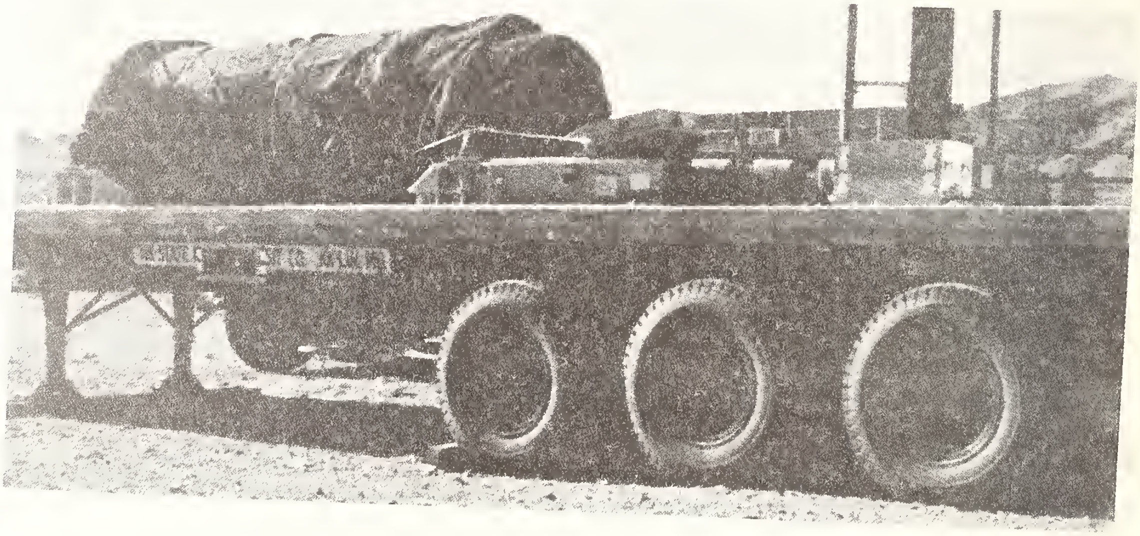

Figure 2. 25 Tonne Truck Cask

For data reduction and definition purposes, the measurements were defined as either shock or vibration. Vibration is considered a steady-state or continuous excitation while the vehicle is in motion. Vibration data samples recorded were generally about one-minute long. Shock is defined as a high-amplitude, short duration excitation. Shock events which occur during over-the-road type measurements are short segments of a data sample and therefore include any background vibration. A typical acceleration-time plot is shown in Figure 3. This data sample is for the vertical axis at the rear of the 25-tonne (28-ton) cask shipment when it crossed a bridge at 89 km/hr (55 mph). These data were filtered at 2,000 Hz low pass and digitized at 5,000 samples per second. Data from these tests are summarized in references 2 and 3.

Rail Car Coupling Tests

Coupling railroad cars together to make a train is sometimes accomplished through the use of a hill or hump. This hump consists of a raised track over which the cars are passed and allowed to roll free. The cars are then directed through a switching system, onto the appropriate track. The coupling is completed when the car impacts the portion of train already made up. This procedure is commonly called humping. Based on 15,648 observations of humping, 99.8 percent of the cars impacted other cars at or below 18 Km/hr (11 mph). Data were available from tests at this velocity but for much lighter weight cargo than was under consideration for spent fuel casks; therefore, additional data were needed.

Savannah River Laboratories planned to conduct rail coupling tests to aid in defining tie-down requirements as part of developing a standard for an American National Standards Institute (ANSI) subgroup. SNL was asked to gather data during these tests. Two spent fuel casks were used in these tests, one weighed 36 tonne (40 tons) and the other weighed 64 tonne (70 tons).

The tests were conducted by running the test car up to the desired speed with a locomotive, releasing the test car, and allowing the test car to impact into four ballast cars weighing about 307 tonne (338 tons) total. The ballast cars had all slack removed from the couplers prior to impact and the brakes were set. Three test cars were used in the tests. These cars were equipped with standard draft gear, a 38-cm (15-inch) hydraulic end-of-car device, and a sliding center sill.

The 36-tonne (40-ton) cask was attached to the rail car using two methods. Longitudinal restraint was provided by steel stops which had been welded to the rail car structure for both tie-down methods. Vertical and transverse restraint was provided by bolts through the skid and car structure in one case (Figure 4) and by cables in the

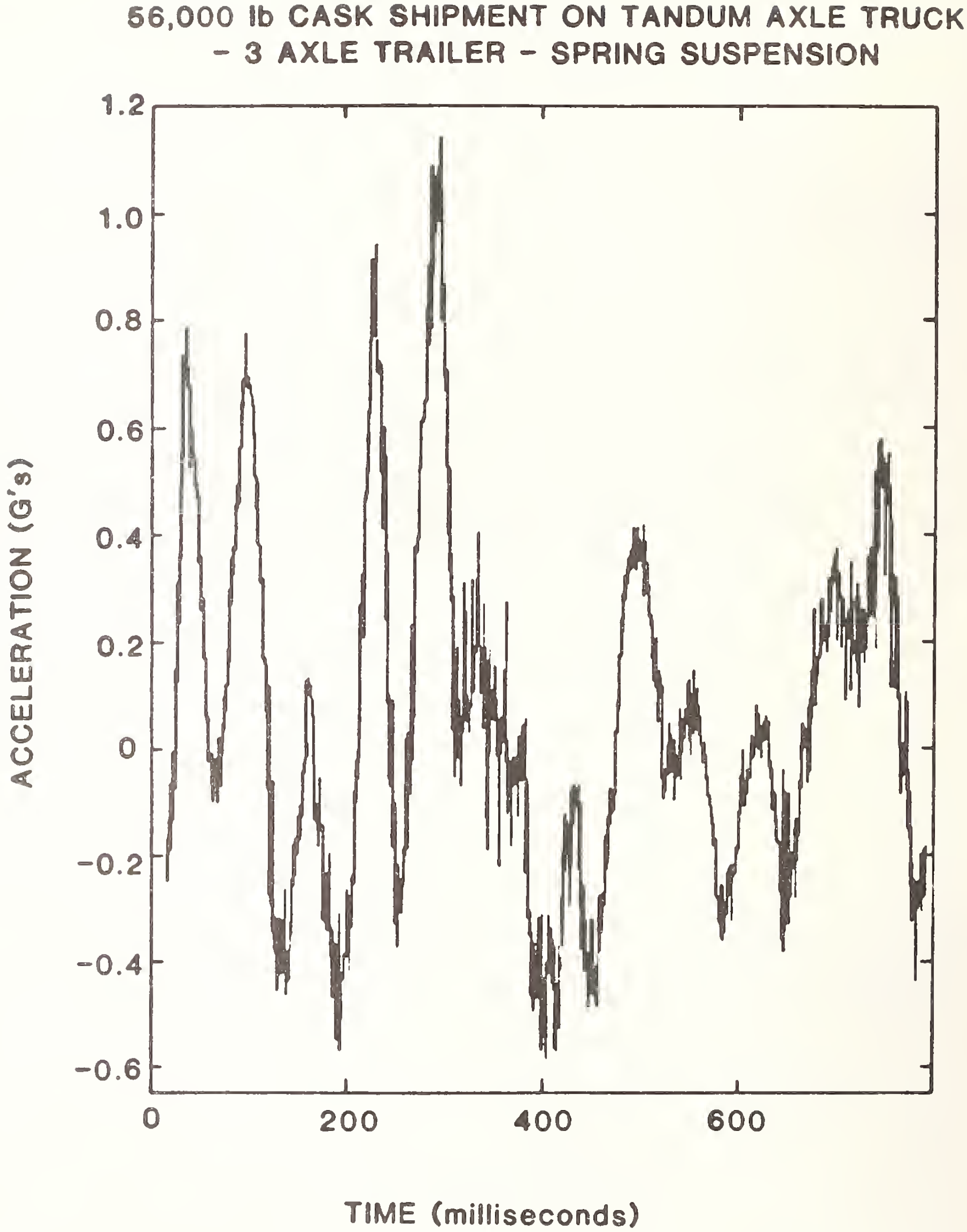

Figure 3. Typical Acceleration - Time Record

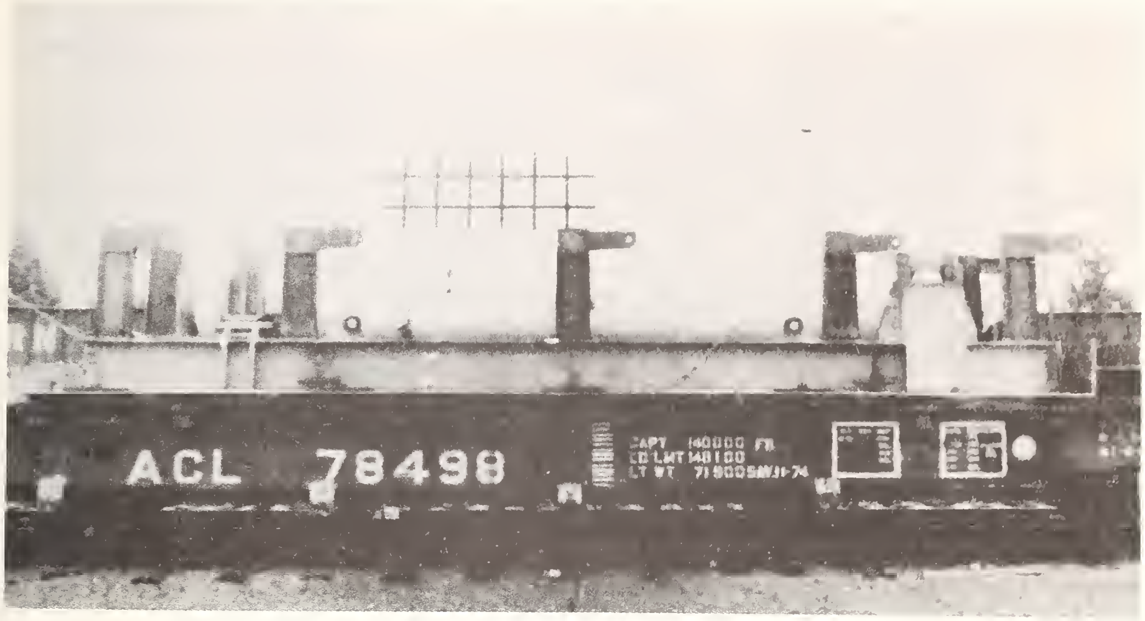

Figure 4. 36 Tonne Rail Car - Bolt Tiedowns

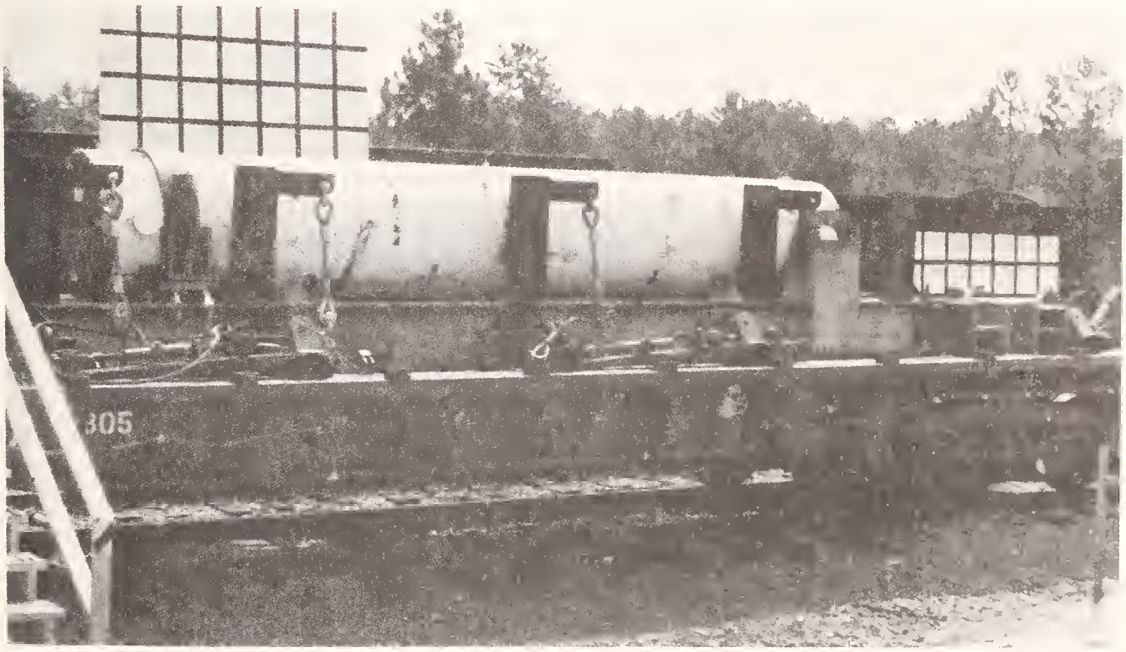

Figure 5. 36 Tonne Rail Cask - Cable Tiedowns

other case (Figure 5). The 64-tonne (70 ton) cask was attached to the rail car using the same horizontal restraint as with the 40-ton cask. Vertical and transverse restraint was provided by bolts (Figure 6).

Instrumentation in the coupling tests consisted of 21 accelerometers, one coupler force transducer, one coupler displacement measuring device, three tie-down force transducers, and two rigid stop transducers. The signals from these 28 instruments were fed to 2 signal conditioning packages, then to a telemetry transmitter, and finally to a SNL-supplied receiving van located near the test site. At the receiving van, the transmitted signals were received and put on magnetic data tapes.

Data reduction was accomplished at different laboratories which participated in the tests. Typical force-time and displacement-time plots for the coupler are shown in Figure 7. A force-time measurement obtained between one of the two horizontal stops and the skid is shown in Figure 8. An acceleration-time plot measurement at the car structure near the coupler for the same test is shown in Figure 9. Table 1 shows the conditions and configurations of the 18 tests which were conducted.[4]

Over-the-Road Railroad Test

An opportunity to gather data during a normal over-the-road rail shipment was presented when SNL was involved in modifying a spent fuel shipping cask. The modifications were made in Denver, Colorado, and the cask was to be shipped to Socorro, New Mexico. The weight of this cargo was about 45 tonne (50 tons).

During this shipment the cask was mounted on a 21-m (68-foot) long flat rail car which was equipped with a 38-cm (15-inch) end-of-car coupling device. Longitudinal and transverse blocking was accomplished by using hardwood blocks. Vertical tie-down was accomplished by two cables (Figure 10).

Twelve accelerometers were mounted on the rail car structure to gather data in the longitudinal, transverse, and vertical axes. Two data channels were used for recording IRIG B time and for electrical noise detection. Signals from the accelerometers were fed to the data acquisition system which was mounted on the instrumented rail car. The data acquisition system was started and stopped remotely by Sandia personnel who rode in a caboose immediately behind the test rail car. Preliminary road surveys were not conducted for this data gathering exercise. The rail carrier was very cooperative throughout the test. Locations for data samples were identified by providing detailed road profile maps and having experienced railroad personnel aboard the caboose with the Sandia personnel.

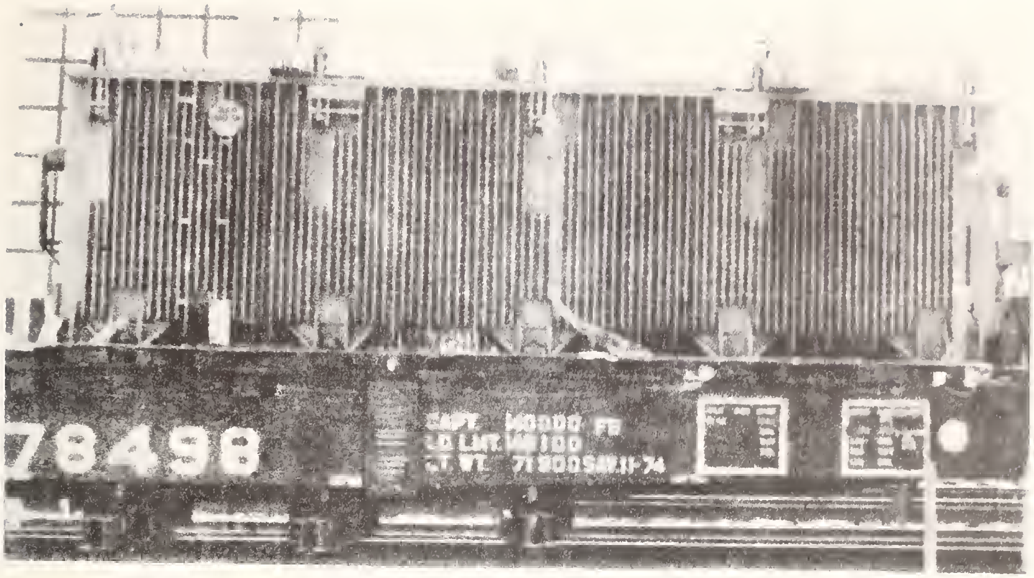

Figure 6. 64 Tonne Cask - Bolt Tiedowns

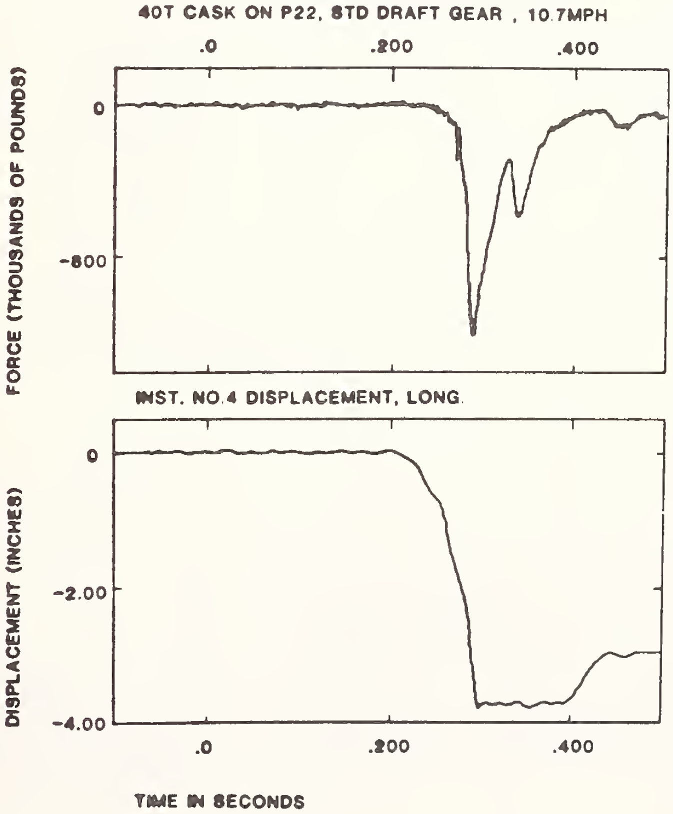

Figure 7. Rail Coupling Tests - Typical Force-Time and Displacement-Time Records

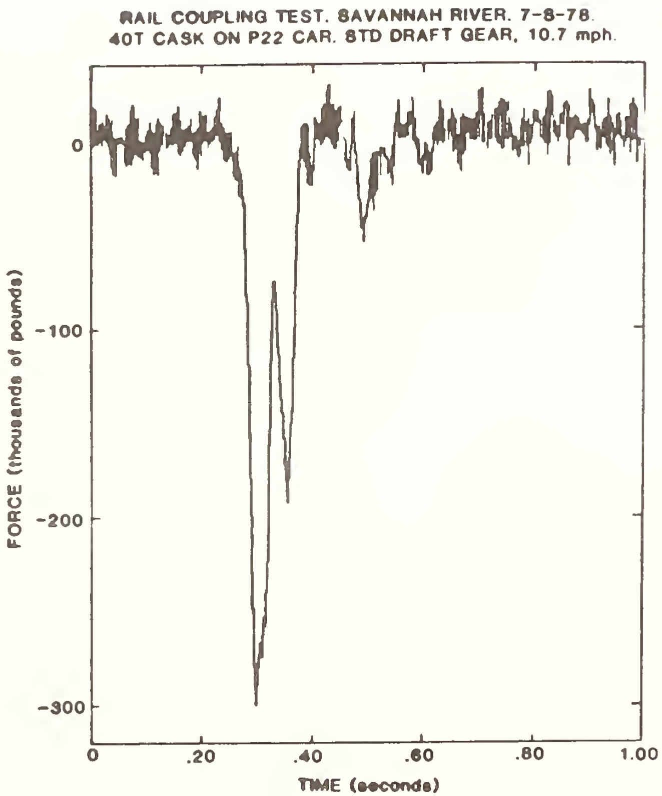

Figure 8. Rail Coupling Tests - Typical Force Time Record

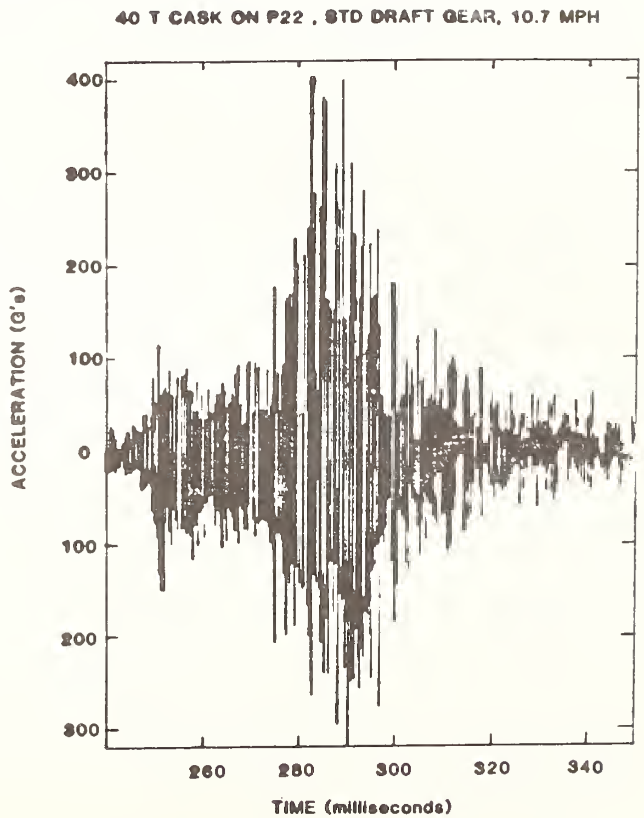

Figure 9. Rail Coupling Tests - Typical Acceleration Time Record

Table 1

Summary of Test Configurations and Impact Velocities

Test No.	Test Car*	Coupler	Cask Weight tonne	(Ton)	Impact Velocities km/h	(mph)
1	A	Standard	36	40	13.36	8.3
2	A	Standard	36	40	14.65	9.1
3	A	Standard	36	40	16.90	10.5
4	A	Standard	36	40	17.22	10.7
5	A	Standard	36	40	16.90	10.5
6	B	End-of-Car	36	40	4.44	2.76
7	B	End-of-Car	36	40	9.06	5.63
8	B	End-of-Car	36	40	14.73	9.15
9	B	End-of-Car	36	40	14.73	9.15
10	A	Standard	64	70	12.91	8.02
11	A	Standard	64	70	17.98	11.17
12	B	End-of-Car	36	40	18.02	11.2
13	B	End-of-Car	36	40	17.86	11.1
14	B	Standard	36	40	8.69	5.4
15	B	Standard	36	40	10.46	6.5
16	B	Standard	36	40	17.38	10.8
17	C	Cushion	36	40	9.50	5.9
18	C	Cushion	36	40	17.22	10.7

*A = SCL car; standard couplers
B = UCC car; mixed couplers
C = SCL car; cushion underframe

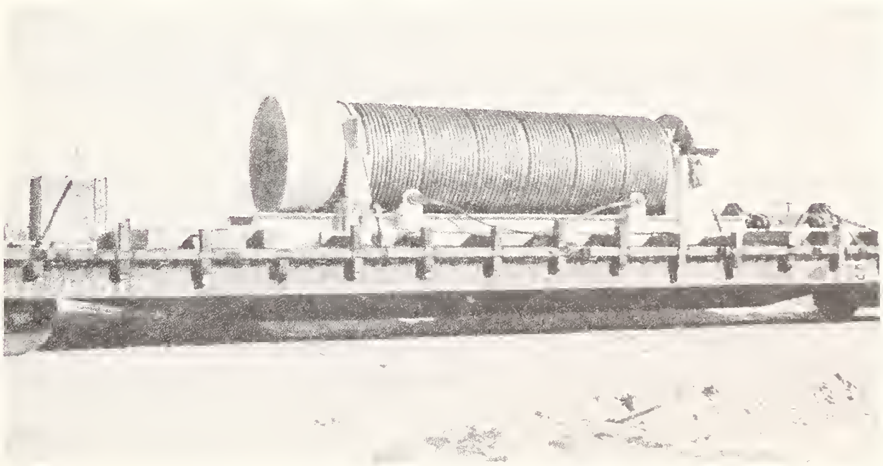

Figure 10. 45 Tonne Rail Cask

The instrumented test rail car was placed at the rear of the trains during the shipment. The only cars behind the test car were cabooses -- one for test personnel and one for the train crew. The trains on which the shipment was made consisted of 43, 46, 53, and 55 cars. The data from this test are still being evaluated.

Proposed Future Work

Several methods of applying the gathered data to the tie-down design problem are being investigated. These investigations indicate that additional data are needed to formulate a general design criteria for nuclear material shipping container tie-downs. The TTC therefore intends to initiate a program which will provide enough data to formulate the necessary design criteria. The steps in this program, in general terms, are listed below.

1. Establish important transportation system parameters.
2. Determine which of these parameters can be held constant by making worst case assumptions.
3. Perform analyses which will use enveloping values for those parameters which cannot be held constant.
4. Validate the analysis results through correlations with experimental measurements, using existing data where possible and gathering additional data if needed.
5. Establish correlations between analytical results and system parameters which can be used to determine design loads for the tie-downs.

These steps are not necessarily in chronological order. In fact, it is expected that item five, for instance, will naturally evolve during efforts to complete the other four.

The ultimate goal of the outlined program is to develop design criteria that can be applied easily and safely. The real question at hand, then, is not whether the program will lead to design criteria, but how easily applied and mistakeproof the criteria will be. In the opinion of the authors, the TTC program will lead to the development of criteria that can be readily applied, with a high level of confidence, to the design of tie-downs for radioactive material shipping containers.

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THE NATURE OF TRANSPORTATION ACCIDENTS
INVOLVING RADIOACTIVE MATERIAL PACKAGINGS*

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Abstract: This paper analyzes the transportation accidents and incidents which have occurred in the United States in the period 1971-1980 based upon the information in the Radioactive Material Transportation Accident/Incident Data Base developed by the Transportation Technology Center (TTC) at Sandia National Laboratories. Principal sources of this information are the Hazardous Material Incident Reporting system of the U. S. Department of Transportation and information from the files of the U. S. Nuclear Regulatory Commission (NRC). Detailed statistics are provided for transportation accidents, handling accidents, package types involved and material transported.

Key words: Accident; data base; failure; handling accidents; hazardous materials; HMIR's; radioactive material; radioactive material transportation; radioactive material transportation accidents; radioactive material transportation analysis; radioactive material transportation incidents; packaging; transportation; vehicle.

This paper analyzes the transportation accidents and incidents which have occurred in the United States in the period 1971-1980 based upon the information in the Radioactive Material Transportation Accident/Incident Data Base developed by the Transportation Technology Center (TTC) at Sandia National Laboratories. The accident/incident data base incorporates the files of the Hazardous Material Incident Report (HMIR) system operated by the Material Transportation Bureau of the US Department of Transportation (DOT) with additional information obtained from the files of the US Nuclear Regulatory Commission (NRC). A principal objective of this paper is to summarize US accident/incident experience for the past ten years,[1,2] thus providing a concise statement of the radioactive material (RAM) package environment for the transport modes of truck, rail and air.

Operated for the US Department of Energy (DOE) by Sandia National Laboratories, the TTC acquires hazardous material incident reports from the DOT on a continuing basis. This information, plus similar information acquired from the NRC, provides a radioactive material accident/incident

*Work sponsored by the US Department of Energy under Contract DE-AC04-76DP00789.

data base which consists of 750 data entries for the years 1971-1980. For the same period, the HMIR system has 105,182 entries for all classes of hazardous materials, including 585 radioactive material entries, or 0.56 percent of the total incidents reported to DOT. This information represents US transportation experience for radioactive materials dating from 1971 through 1980. The accident/incident data base is stored on TTC's on-line data base management system for rapid storage, retrieval, editing and analysis.

The DOT regulatory requirements for reporting a hazardous material incident are specified by the Code of Federal Regulations (49 CFR, 171.15). These regulations specify reporting after each incident that occurs during the course of transportation (including loading, unloading and temporary storage) in which as a direct result of moving hazardous material: (a) a fatality occurs; (b) a person receives injuries requiring hospitalization; (c) estimated carrier or other property damage exceeds $50,000; (d) in the case of radioactive material there is fire, breakage, spillage or suspected contamination; (e) fire, breakage or spillage occurs involving etiologic agents; (f) a situation exists that, in the carrier's judgement, should be reported, e.g., a continuing danger of life exists at the scene of the incident.

The NRC regulatory requirements for reporting a radioactive materials incident are specified by the Code of Federal Regulations. 10 CFR 20.402 concerns the theft or loss of radioactive materials and 10 CFR 20.403 concerns exposure to radiation and release of radioactive materials.

From this summary of the requirements for reporting a radioactive materials incident, one can conclude that the requirements for reporting vary widely. The data entries in the accident/incident data base are those which generally meet the regulatory requirements for notification of DOT and/or NRC of an occurrence. The main theme of this paper is that a detailed examination of the reports in the data base reveals that some of the events can be classified as transport accidents, some events involve handling accident conditions and others are simply incidents which meet the regulatory requirement for reporting an event. Such a classification of events is shown in Table I and is necessary to produce, for example, accurate accident rate estimates for risk assessment studies. Table I represents an up-to-date representation of US radioactive material transportation experience, which show that only 13.5 percent of the incidents involve transportation accidents.

In the interpretive analysis of the basic DOT and NRC information, a distinction is made between an accident and a reported incident. Two types of accidents are categorized, as shown in Table I. For the purposes of this analysis, a reported incident is classified as a transportation accident if there is a vehicular accident involving the vehicle transporting the radioactive material. Other incidents are classified as handling accidents if during the course of handling, loading, etc., an accident occurs to the package, for example, a package is dropped or is punctured by a forklift.

TABLE I

US Radioactive Material Accident/Incident Experience

(1971-1980)

Transportation Accidents	101
Handling Accidents	125
Reported Incidents, Other	554
Total	750

The remaining classification, "Reported Incidents, Other," excludes the two accident categories and comprises the balance of the incident reports in the data base. These may arise from: (1) actual or suspected release of radiation of materials; (2) surface contamination on the package or transport vehicle in excess of regulatory requirements (49 CFR 173.397).

The suspicion of release results from many causes. The most common are minor damage to the outer container, wet packages, liquid leaking from a closed trailer and erroneous meter readings. The wet package occurs often in the shipping of radiopharmaceuticals by air. The first suspicion is that the inner container failed. Subsequent investigation usually determines that the package was loaded aboard the aircraft during rain or snowfall. The liquid leaking from a closed trailer occurs frequently after a truck has been driven through rain and later, during dry highway conditions, the liquid that accumulated inside the rear trailer doors seeped out.

Removable (non-fixed) surface contamination incidents are often noted during radiation surveys in loading and unloading operations. They are most likely to occur when the trailer is being returned to general highway use after a sole use shipment. Such events generally involve no package failure, no release of radioactive contents and the absence of any accident conditions during transport.

The 101 transportation accidents listed in Table I were classified by transportation mode as shown in Table II.

Table III summarizes some of the information that can be retrieved from the accident/incident data base. For example, the 101 transportation

TABLE II

Radioactive Material Transportation Accidents by Mode

Transport Mode	Number of Accidents	Number of Packaging Failures	
		Strong & Tight*	Type A
Air	7	--	3
Rail	7	1	--
Highway	86	60	4
Courier Service	1	--	--
	101	61	7

*49 CFR 173.392

TABLE III

Transportation Accident Analysis Summary

RAM Packages Involved in Accidents	Package Description	Description of Events		Description of Material Released
Packaging Failures* (with release)	53 Strong & Tight 5 Type A	9 Release Events	6 Urban	·Uranium concentrate ·Monazite sand ·RAM NOS ·Radiopharmaceuticals
			3 Non-Urban	·Uranium[3,4,5] concentrate
Packaging Failures* (no release)	8 Strong & Tight 2 Type A			
Packagings in Accident with No Failures	761 Strong & Tight 237 Type A 48 Type B			
TOTAL	1114 packages involved in 101 accidents			

*The 68 packaging failures involved 58 packages which released radio active material.

accidents involved a total of 1114 radioactive material packages. Some accidents had only a single package on board the transport vehicle. Other accidents involved a single transport vehicle with a number of packages on board. Table III traces an analysis of the radioactive material packaging failures in the 101 accidents.

Multiple numbers of packages can be carried on a single vehicle and nine separate and distinct release events involving these packages have been defined. Six of the release events occurred in an urban area. Three of the release events occurred in a non-urban area. For example, the Springfield, Colorado,[3,4] accident involved 29 packaging failures which released approximately 5400 kg of uranium oxide; this was defined as a single release event. For the nine release events, the material released was described as uranium ore, radioactive sand (low specific

activity), uranium concentrate radioactive material (RAM--not otherwise specified) and radiopharmaceuticals. It is interesting to note that an examination of the accident data reveals that 48 Type B accident resistant packagings were included in the set of packagings which were subjected to vehicular accident conditions, but produced no packaging failure and, consequently, no release of radioactive contents.

Radioactive materials being transported in the accidents noted in the data base include the following:

No.	Material
4	Spent fuel casks (2 w/spent fuel, 2 empty)
24	Low level waste (RAM LSA)
14	Radiopharmaceuticals
22	Radiography or oil well logging sources
8	Uranium concentrate
1	Monazite sand
8	Uranium Hexafluoride
1	Teletherapy source
19	RAM LSA or RAM NOS

Tables IV, V, VI and VII present a chronological summary of the data base with respect to the number of reported events, transportation accidents, handling accidents, release of radioactive contents and accident occurrences by transport mode. The following observations apply to Table IV. The accident/incident data base receives most of its data entries from the HMIR system of the DOT. The other principal data source is the NRC. The HMIR data originated in 1971 and the NRC data originated in 1976. Any calculation of rate of occurrence of accidents and incidents should be performed with caution since the overall accident/incident data base consists of elements with differing chronological origination dates and completeness of information.

TABLE IV

Accident/Incident Data Base Chronological Summary

Year	Annual No. Incidents	HMIR* Incidents	Other** Incidents	Transportation Accidents	Handling Accidents	Annual No.*** Release
1971	12	12	N/A	5	0	7
1972	21	21	N/A	1	2	5
1973	24	24	N/A	0	11	7
1974	60	60	N/A	5	16	10
1975	44	44	N/A	5	23	8
1976	62	58	4	4	6	14
1977	113	97	16	10	15	17
1978	149	92	57	20	15	22
1979	171	120	51	30	24	23
1980	95	57	37	21	13	20
	750	585	165	101	125	133

*Reported events received from HMIR system, DOT.
**Reported events received from NRC and other sources.
***Reported events where material or excessive radiation was released (all categories)

TABLE V

RAM Transportation Accident Summary

Year	Transportation Accident Total	Transport Mode: Rail	Highway	Air	Other	Releases
1971	5	1	4	0	0	1 Release-1971-Hwy
1972	1	0	1	0	0	
1973	0	0	0	0	0	
1974	5	1	4	0	0	
1975	5	0	5	0	0	
1976	4	0	4	0	0	
1977	10	1	9	0	0	1 Release-1977-Hwy
1978	20	0	19	1	0	1 Release-1979-Air
1979	30	2	24	4	0	1 Release-1979-Rail
1980	21	2	16	2	1 (Courier)	2 Release-1979-Hwy
						1 Release-1980-Air
	101	7	86	7	1	9 Release Events

TABLE VI

RAM Handling Accident Summary

Year	Handling Accident Total	Transport Mode				Releases
		Rail	Highway	Air	Other	
1971	0	0	0	0	0	0
1972	2	0	0	2	0	1, Air
1973	11	0	2	9	0	2, Air
1974	16	0	1	15	0	2, Air
1975	23	0	6	14	2, Freight Forwarder 1, Marine	2, Hwy 1, Air 1, Frt Frd 1, Marine
1976	6	0	2	4	0	1, Hwy
1977	15	1	4	9	1, Warehouse	3, Hwy 1, Rail 1, Warehouse 2, Air
1978	15	0	8	7	0	2, Hwy
1979	24	0	9	1	1, Courier Service	2, Air 3, Hwy
1980	13	0	8	5	0	1, Air 2, Hwy
	125	1	40	79	5	

The most representative view of accident and incident rates is to express their occurrence as fractions (or percent) of total shipping volume. The most recent estimate of total shipping activity in the US was performed in 1975.[6] Estimates of present shipping activity are presently being investigated. Until new estimates of shipping activity are available, it is not possible to gain a current assessment of the percentage of accidents and incidents based upon numbers of radioactive material shipments.

To conclude, it is important to note that the accident/incident data base described in this paper represents the accumulation and organization of existing data sources available in the United States. This data acquisition is followed by interpretive analysis of the information, and this analysis and data is shared with several government agencies such as the DOE, NRC, DOT, Environmental Protection Agency (EPA), Federal Emergency Management Agency (FEMA) and the International Atomic Energy Agency (IAEA), as well as other interested state and local governments and public interest groups. One important fact that has come from the preparation of this report is the importance of performing a detailed study of the incident reports and any attached documentation. It is recognized that after such an examination of radioactive material transportation incident data that all of the reported events are not transport accidents, hence the stratification of the accident/incident data as shown in Table I. Such a classification as shown in Table I is necessary because the varying requirements for reporting hazardous material incidents allow a broad spectrum of data entries which must necessarily be subclassified (i.e., transport accidents, handling accidents and reported incidents).

TABLE VII

RAM Handling Accident Analysis

Package Description	No. Fail	No. Release	Transportation Mode (No. of Events)		Materials Involved
Strong and Tight	9	8	Highway	4	RAM Small Quant.
			Water	1	RAM NOS
			Freight	1	Uranium Concentrate
			Forwarder		RAM LSA
			Warehouse	1	
			Rail	1	
Type A	65	33	Air	10	Radiopharmaceuticals
			Highway	9	RAM LSA
					RAM NOS
					Radiography Sources
Cause for Handling Accident Report					
	Dropped in Handling				46
	Run Over by Vehicle				27
	Fell in Transit				24
	External Puncture				21
	Misc.				7
					125

The information presented represents the most complete analysis to date of the transportation accident/incident data base for radioactive materials. The results of such an analysis can be used to provide information about the environmental impacts associated with the transportation of radioactive materials. In addition, such an analysis can provide information which is useful in the formulation of the regulations governing the safe transportation of radioactive materials.

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PERCEIVED SAFETY OF TRANSPORTING HAZARDOUS MATERIALS*

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INTRODUCTION

This paper addresses an apparent anomaly in the transportation of hazardous materials which involves the issue that safety as embodied in rules and regulations to limit the public's risk to "acceptable" levels is not the same as the perceived safety of the transport activity. The basic assertion of this paper is that there are reasons for this difference between the measured or defined degree of safety in transportation and the safety of this activity as perceived by the public. Safety, the condition of being free form undergoing or causing hurt, injury, or loss, appears to incorporate the probability of exposure to a hazard or danger. While safety may be difficult to measure in absolute terms, it is possible to make decisions on how free from harm an event might be. This decision process is judgmental in nature, usually different for each individual or group and is based on the likelihood of exposure to a hazard or danger. Further, any consideration of safety usually involves some consideration of risk. Risk assessment techniques can be used to determine the probability that an unwanted event will occur and what detrimental consequences will result if it does, but perceived safety appears not to rely on scientific analyses. Perceived safety, however, can have a greater influence on decisions than do objective analyses of probabilities and postulated consequences.

Despite a wealth of data and information on the risks of a variety of personal and industrial hazards faced daily, public attitude remains uniformly indifferent or, at most, is exercised selectively for short periods of time. This public indifference seemingly disappears when transportation of hazardous--particularly radioactive--materials are considered.

The scope of this paper is narrow and tries to describe the relationship that exists between many variables that appear to influence the public view of safety in hazardous materials transportation with particular emphasis on radioactive materials. The major problem in

*This work done for the U. S. Department of Energy under Contract DE-AC04-76DP00789

analyzing this issue of perceived safety is the lack of an accepted framework or methodology in which a meaningful discussion and assessment of the issue can occur by the institutions in society and government. This paper presents a framework represented by an equation of several variables and thoroughly describes each term.

PROPOSED FRAMEWORK

This paper, written from a technologist's viewpoint, suggests that such a framework can be constructed which will relate the variables applicable to perceived safety of transporting hazardous materials. This framework may have other application to activities considered hazardous or presenting some risk but those are left to the reader. Using this framework, the discrepancies that exist between public understanding and analytical measures of consequences and risks occurring in the transport process can be better described.

The suggested framework relies primarily on information that is currently available on transportation systems and processes. This information is summarized in Appendix A. The framework organizes this information into a relationship which, for convenience sake, appears as an equation. However, since the variables describing public perception are unlikely to be mathematically independent, a conditional (Boolean) relationship is given in Equation (1) which indicate varying degrees of unquantified dependence. Perception (P) is considered to be conditional ($\Longrightarrow$) on the variables shown in Eq. (1). The variables themselves are considered to be conditional on each other or bi-conditional ($\Longleftrightarrow$).

$$P \Longrightarrow TF \Longleftrightarrow PP \Longleftrightarrow SR \Longleftrightarrow UA \Longleftrightarrow MI \Longleftrightarrow F(t) \qquad (1)$$

where P = Perception

TF = Technical feasibility
PP = Political palatability
SR = Social responsibility
UA = Utility assessment
MI = Media interpretation
F(t) = Familiarity as a function of time

Equation (1) brings together the significant variables and organizes public perception into a relationship in which discussions and assessments can occur. The purpose of this conditional equation is to present a body of information in a concise form. While it may be impossible to compel an individual or group to accept a statement on the degree of safety of hazardous material transportation, it is important to try to have an individual or group notice a body of information and take notice precisely.

Equation (1) does not have a rigorous mathematical foundation. Considerable work is needed to determine if and how a mathematical model can be developed. In addition, the manner in which the variables are weighted and how the mathematical manipulations are accomplished need to be determined. The intent is to present a framework to more completely address this divisive safety issue. It is hoped that by using this conditional equation, an improved understanding of perception and eventually a method for the technologist to be able to comprehend the public's perception of activities they consider hazardous. The relationship between public perception of an activity considered hazardous or how the public accepts the associated hazards (risks) is not addressed in this paper. However, it is suggested that, at a minimum, some relationship exists between public perception and acceptance. The conditional relationship proposed in Eq. (1) could be a basis for evaluating public perception and, in turn, public acceptance.

The remainder of the paper is a discussion of each term of Eq. (1) and how available information is incorporated into the variables.

TECHNICAL FEASIBILITY

Technical feasibility as defined by Parsons (Ref. 19) is that body of data, information, and options that indicate a capability to do something in both an efficient and safe manner. Coupled with this set of quantified data elements is the research supporting the anticipated results from technology. However, implementation of the results of technical efforts often run into opposition. Recipients of technological advances require some form of supervision and control of the scientific and engineering activities.

The need to transport nuclear materials requires that transportation be made technically feasible. The major question is how to move the materials safely. Therefore, packagings and containers are developed, standards for their design and testing are formulated, and shipping systems developed or adapted from transportation industry are used to move these materials. The feasibility assessment is a thorough consideration of the adequacy of the analyses, designs, tests, operations and if the unanticipated has been provided for. This term of Eq. (1) is based on four major public and environmental concerns: (1) the degree of protection offered by the packagings, containers, and shipping systems, (2) the types of hazards posed by releases of the materials or wastes in the presence of dispersal mechanisms (or ignition sources), (3) the immediate and lingering aspects of detrimental health effects and damage to the ecosphere from exposure to hazardous materials, and (4) indications that human errors have been reduced and carelessness avoided. The technical feasibility term of Eq. (1) is a recognition of the requirement to perform the transport process in a safe, efficient, and cost-effective manner.

POLITICAL PALATABILITY

The political palatability term of Eq. (1) encompasses all those actions by elected or appointed governmental officials in considering an activity whose impact may be hazardous. This assessment is based primarily on external considerations usually reflected in preferences which often result from a public consensus regarding some action or proposal. Decisions in the political arena are often made with some sense of real or implied urgency in which incomplete scientific evidence is available. Because hazardous materials transportation affects virtually every community within the country and is increasing in quantity and frequency, many citizens are concerned with its impact on their locality. Such citizen concern directly impacts the politcal palatibility. For example, strident public demands that radioactive materials not be shipped in or through their communities directly affects political palatibility.

The political process mandates that the elected official tries to adopt a position reflecting his concern for public health and safety while securing a favorable appraisal of his position by the electorate. The politician will seldom take a position that would be considered as forcing some hazardous activity on his constituents without some compensating benefit.

Therefore, transportation of hazardous materials exhibiting selected characteristics (corrosive, explosive, flammable, oxygen denying, radioactive, or toxic) and which do not portray a politically palatable benefit to a community that is commensurate with the hazard presented will affect the perception of its safety.

SOCIAL RESPONSIBILITY

Social responsibility encompasses the concern that current actions and decisions will meet the tests of time. This third term of Eq. (1) reflects the obligation to protect current and future generations from unwanted consequences from hazards produced today. The public view of how responsibly these concerns are being met will influence the view of safety. These concerns are based on two assumptions: (1) that consistent historical patterns of accident data have emerged in terms of consequences to the public and (2) that established societal institutions are sufficiently enduring that they can be used for predictive purposes. Therefore, social responsibility addresses the accuracy of these two assumptions (1) that history is a predictor of future problems and (2) institutions will continue to protect society.

Many devices and checks and balances currently exist to insure that the concern for social responsibility is satisfied. Among them are: (1) federal, state and local laws, (2) regulations at the same levels of goverment, (3) enforcement agencies, and (4) self-imposed standards and checks developed by manufacturers, users, transporters, and disposers of hazardous materials and wastes.

The fundamental question before society is the degree of protection that can and should be offered current and future generations. What appears to be desired is a continued situation in which the fruits of technological advancements can be used and enjoyed balanced with assurances that mankind, the ecosphere, and the genetic inheritance can be maintained in a status quo or improved for each succeeding generation.

UTILITY ASSESSMENT

This fourth term of Eq. (1) as used here is the recognition that a planned action satisfies or aids some group. For example, the public knows it has become necessary to organize, store, and dispose of hazardous wastes. The manner in which the processes associated with these activities is performed, who is affected, and who will be rewarded are the elements to be considered.

The basic need to transport hazardous materials from manufacturers to users and the waste products to disposal sites is a premise of this paper. The public has a limited, but sufficiently valid, understanding that these activities causing this need has or could provide benefits to them.

However, the utility assessment of transportation is questioned when one or more of the following occur: (1) an incident or accident (2) a new or different material is transported through a community, (3) studies, analyses, or environmental impact statements are released which give data defining worst case conditions.

The major problem in evaluating the utility of a hazardous activity is separating the issues or possible effects involved. No one suggests that society should accept a potentially hazardous situation just because their economic life depends on it. However, society should at least attach a utility value to the operation that is meaningful and in their interest.

MEDIA INTERPRETATION

The purpose of news gathering and dissemination services is to analyze and report the news while obtaining and keeping readers and listeners. The media play an important part in defining that which is newsworthy. The interpretation of incidents, accidents, or activities associated with transportation of hazardous materials is based on what the media has determined will be of interest to its users. The more that a particular event can appeal to basic human concerns (i.e., fear), the more likely the event will be publicized.

Another aspect of media interpreation is the interaction with technology developments. The more complex a technology is and the more terms and jargon required to explain that technology to the public, the more likely it will be scrutinized by the media. This type of technology presents the opportunity for the media to present spurious arguments that are not easily refuted and understood by the public.

Transportation of hazardous materials satisfies these two requirements for being newsworthy because (1) the term hazardous means something to be afraid of and (2) hazardous materials are often associated with complicated technologies.

The media interprets the causes and effects of an event. There is a distinct difference in the media treatment of a single "big" event and a series of similar individual events scattered over time. Clearly, the single big event is more newsworthy. Because accidents or incidents involving hazardous materials occur infrequently, these events are usually treated as major, causing fear, and represent something more complicated than routine accidents.

FAMILIARITY AS A FUNCTION OF TIME

DuPont (Ref. 9) indicated that one important factor in public evaluation of hazardous activities is whether something is familiar or unfamilar. Those dangers which occur commonly such as travel by automobile representing a risk of one in four of a serious personal injury accident over a lifetime are not considered equivalent to risks resulting from a much less frequent but unfamiliar event. Dangers from commonly accepted activities are treated on a "what is" basis and usually disregarded in decision-making by the public. However, hazards from unfamiliar activities are discussed on a "what if" basis and are incorporated into the same set of decisions.

If two activities exhibit equivalent detrimental consequences to public health and safety and they were introduced concurrently, then these two activities will probably be judged as an equal hazard by the public. However, if one of these activities was introduced at some prior date (i. e., 30 years ago), then the newer activity would be undergo significantly greater public scrutiny. As the public becomes accustomed to hazardous activities, they have characteristically become less concerned about them. The more often a group is exposed to a hazardous activity the more familiar they will become with it and exhibit fewer concerns. Perception can be changed only be repeated exposure to a feared situation.

CONCLUSIONS

A framework for relating the variables involved in the public perception of hazardous materials transportation was presented. The framework consisted of a conditional mathematical equation in which perceived safety was described by six basic terms (technical feasibility, political palatibility, social responsibility, utility assessment, media interpretation, and familiarity as a function of time). The resulting framework provides the technologist with an initial formulation to better understand public perception.

APPENDIX A

AVAILABLE INFORMATION

Shipment Information

There is a continuing need to know the quantities and types of hazardous materials, packagings and containers and shipping systems which move routinely in commerce. This information provides the backbone for risk assessment. Currently the only nationwide data available for radioactive materials is contained in the 1976 Battelle Pacific Northwest Laboratory study. The Battelle study is currently being updated to establish a data base that can be used to generate quantitative descriptions of the paramters of the transport of radioactive materials. There have also been a number of studies performed within particular states.

Less complete shipment information is available for other categories of hazardous materials. Two sources are referenced (23, 24). Among the problems in obtaining more accurate information are the following: (1) the large number of shipments (estimated at more than 200 million) per year, (2) the proprietary nature of many shipments, and (3) the costs of obtaining such information.

Accident/Incident Data

The definition of accident and incident must be explained because of the legal requirements for reporting occurrences. The U. S. Deparment of Transportation (DOT) requirements for reporting a hazardous material incident are specified in the Code of Federal Regulations (49 CFR, Para. 171.15). These regulations specify that an incident must be reported if (1) a fatality occurs, (2) an injured person requires hospitalization, (3) estimated damage to the carrier or property exceeds $50,000, (4) continued danger of life exists at the scene, (5) fire, breakage or spillage occurs involving etiological agents, and (6) for radioactive shipments suspected contamination. The data is gathered and organized by DOT into the Hazardous Materials Incident Report (HMIR) system. As a subset of this data base, the Transportation Technology Center (TTC) of Sandia has developed the Radioactive Materials Incident Report (RMIR) system.

It must be emphasized that all incidents are not necessarily accidents involving vehilce damage. The data from 1971-1980 for radioactive material shows that about 13 percent of all reported incidents involved vehicular accidents. The 85 reported accidents involved 711 packages or containers. In the same time period, there were 11 accidents involving explosions on interstate highways throughout the country. The data give a brief summary of the accidents in which hazardous materials are involved.

Role of Regulations

Through the Nuclear Regulatory Commission (NRC) and the Department of Transportation (DOT), the federal government of the United States regulates such aspects of hazardous materials transportation as packaging, vehicle safety, and placarding, and for radioactive materials, physical security. Within the past decade, state and local governments have also become involved in the regulation of hazardous materials transportation. This state and local involvement gives rise to important policy questions: (1) To what extent should the federal government regulate the transportation of hazardous materials throughout the country? and (2) To what extent is state and local regulation permissible under federal law?

Central to the above two questions is an understanding of the philosophy of regulations. Historically, control of safety related activities received less emphasis than the interests of the shippers, carriers, and suppliers. This emphasis shifted in 1967 with the creation of the Department of Transportation. This shift resulted in a government agency as the primary mover of regulations governing hazardous materials transportation.

The technical questions in the regulatory area are primarily concerned with (1) determining if existing regulations provide a known level of safety or should they be changed and (2) if the regulations are changed will the level of safety be affected (improved). A side issue not necessarily of lesser importance is that of evaluation of cost-benefit aspects of regulatory policy.

There is a large body of information available on regulations. Environmental impact statements contain summaries and rationales for regulations. These regulations provide for normal operations as well as standards for design and development of packages and virtually all facets of the transportation system.

Liability and Insurance Coverages

As the number of shipments of hazardous materials has increased, more people have become concerned about public liability and compensation issues. Radioactive materials shipments have received most of the attention. One of the most misunderstood aspects of the regulatory structure that applies to transport of hazardous materials is the coverge for public liability.

In the case of radioactive materials, the regulatory structure provides for very comprehensive system under the omnibus provisions of insurance and indemnity coverage for public liability. The insurance industry and the federal government have recognized that, even with stringent standards for packaging and operation of transportation systems, the possiblility of a transportation accident resulting in some measure of damage to persons or the environment cannot be eliminated. Therefore, the federal government and the insurance industry provide coverage through the Price-Anderson Act and the federal tort system.

In addition to these coverages for radioactive materials, other systems are used for public liability for hazardous materials. The Superfund (the "Comprehensive Oil and Hazardous Substances Pollution Liability and Compensation Act") provide for large coverages. Recent regulations have also required carriers to have their own liability coverages to protect the public and the environment.

Packages and Shipping Systems

The design standards and requirements for packagings and containers for shipments of hazardous materials are the responsibility of more than one federal agency. For shipments of all hazardous materials except radioactive, DOT has defined the requirements and certifies the packagings and containers. DOT determines that criteria which define adherence to accepted and proven engineering practices are followed. The U. S. Nuclear Regulatory Commission (NRC) has established the requirements for packagings for radioactive materials. These requirements are based on performance characteristics for normal transport (leaktightness) and post-accident conditions including impact, puncture, fire, and submersion.

Emergency Preparedness

Federal, state, and local agencies have responsibilities for planning, obtaining equipment, training, and providing assistance and direction in the event of emergencies involving hazardous materials. In addition, industry has organized its own system to aid those affected by spills or releases of hazardous materials.

The non-existence of emergency preparedness programs has been cited as a reason to restrict the flow of hazardous materials in commerce. However, public concern has applied pressure to governmental agencies and has resulted in more resources applied to the the concerns for protection from the effects of these materials. In addition, the prompt actions by industry to respond to accidents and the development of companies who specialized in cleaning up spills of materials have also addressed these concerns. The result is an improved perception that the existing emergency response systems would adequately respond to accidents involving hazardous materials and control the associated detrimental consequences.

Information Requirements

There is a continuing awareness of public officials and the public of the key role transportation plays in the utilization of hazardous materials. This awareness is reflected in issues involving radioactive materials transportation including safety, the role of the federal government, and current and evolving policy and policy trends for areas not now covered. These issues and questions generate the need for factual information and analyses of transport policies. In addition, policymakers, program planners, and corporate decisionmakers need accurate and timely data on which to base their program planning. The requirement to provide accurate and objective information to such a diverse group is a large task for transporters of hazardous materials. This requirement is being met by a variety of informational tools, techniques and media forms.

Information on many aspects of the transportation of hazardous materials exist in files and libraries and more data are being added on a continuing basis. A central repository and standard query system do not exist. The problem with this aspect of what is known about hazardous materials transportation is an evaluation of what is "good" or "bad" or that which is relevant, useful, and objective and that which is not.

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ANALYSIS, SCALE MODELING, AND FULL-SCALE TESTING OF SHIPPING CONTAINERS FOR RADIOACTIVE MATERIALS

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INTRODUCTION

Abstract: This paper will discuss techniques for predicting the structural response under accident conditions of shipping containers designed for the transportation of radioactive materials. The techniques include numerical analysis methods as well as physical scale modeling. These methods will be discussed with some examples, and comparisons will be made between analysis results and those of full-scale tests conducted subsequent to the analyses.

Key words: Spent fuel, radioactive materials transportation, spent fuel cask, impact tests, physical scale modeling, computer analysis, lumped parameter analysis, finite element anlaysis and full scale testing.

The class of containers that these techniques have been principally applied to are heavy (23,000 - 68,000 kg) lead shielded casks used for the shipment of spent fuel rods. These techniques, however, are now being applied to a different type of container designed for low level waste materials. This paper will focus on work related to spent fuel casks.

Regulatory requirements for the containers include a specific series of tests that a container must pass prior to being issued a license. These tests include among others severe drop and puncture tests as well as fire testing. The regulations require that it be shown either by analysis or test that a particular container design can survive the prescribed sequence of tests without loss of contents. The techniques described in this paper can be utilized for this purpose. In addition to being utilized for regulatory requirements, these methods over the past few years have been utilized to analyze what will be termed "extra regulatory" tests. These are tests which have been designed to be more easily related to what could conceivably occur in the transportation environment, but having an extremely remote probability of occurrence [1]. An example of such an event is the broadside impact of a locomotive, traveling at high velocity, against a shipping container. This paper will briefly review work which has been done in the area of applying numerical analysis and scale modeling techniques to analyze shipping containers subjected to this type of environment. Results of these analyses will be described and related to actual full-scale test results.

Reference 2 lists three accident scenarios for which the complete cycle of mathematical analysis, physical scale modeling, and full-scale testing have been completed. These accident scenarios include: (1) a head-on impact of a tractor-trailer system, carrying a spent fuel shipping cask, into an extremely rigid reinforced concrete target at a velocity of 129 km/hr (80 mph), (2) a railroad crossing accident scenario where a spent-fuel cask is impacted by a locomotive traveling at 129 km/hr (80 mph), and (3) a head-on impact of a special railroad car carrying a spent fuel cask into the same target as test 1 above and at the same velocity.

The results of these tests are discussed in detail in references 3, 4, and 5. These results indicated that the techniques described in the paper are valid for analyzing these types of impact environments. In this paper results from the first two test scenarios (grade crossing and head-on impact), which were the only ones where the basic structure of the cask underwent any amount of permanent deformation, will be used as illustrations.

NUMERICAL METHODS

Numerical methods used to analyze the dynamic impact response of shipping casks include lumped parameter modeling and dynamic finite element modeling. Both methods are dynamic techniques which discretize the structure into elements. The lumped parameter technique uses a much coarser discretization and is generally a one-dimensional model. It is based on being able to estimate the force-displacement characteristics for the structure and solving the dynamic equations of motion for the resultant spring-mass system, which can have linear and nonlinear springs (couplings). This method has been used to model the vehicular system and the cask in order to predict the approximate velocity at which the cask would impact a hard surface after crushing the vehicle structure. It can also be used to approximate the forces acting on the cask during the impact. Figure 1 is an example of such a model. Here the discretized structure is given an initial velocity in the direction of the target and the response of the system in terms of displacements, velocities, and accelerations is obtained. The model shown in Figure 1 was used in conjunction with the SHOCK [6] computer program which solves the equations of motion. This program has capabilities for linear and nonlinear couplings with hysteresis effects, which are essential in this type of modeling.

Lumped parameter models have also been used to analyze only the container. In this case the container is discretized into a number of elements for the impact analysis. In reference 7 it was found that a model containing about twelve elements gave good results for stress values at different points in the cask. This work was limited to end-on impacts but analysis for a lateral impact can also be accomplished using lumped parameter techniques.

Finite element analysis techniques can be used to analyze container response in much more detail. This technique can be used as a second step in the analysis process. For example, the model of Figure 1 indicated the velocity at which the cask might be expected to impact the rigid target. This impact velocity can then be used as input to a finite element model in order to obtain a more refined analysis. Figure 2 illustrates such a model formulated as the second step in the analysis. This model was used in conjunction with the HONDO [8] finite element program and represents an axisymmetric impact condition. The HONDO program has the capability of providing an impact boundary condition which simulates a very rigid target by applying restoring forces to prevent node penetration across the target surface. This model included nonlinear material properties and sliding interfaces between the lead and stainless steel shells. As with the lumped parameter model, the structure was given an initial velocity into the target. The plot of Figure 2 illustrates the deformed condition that was calculated for the impact-end of the cask assuming an unyielding target.

The finite element model described above was axisymmetrical and as such it was a 2-dimensional model. Two-dimensional finite element models of this type are very feasible and the computer run times are quite reasonable. Some problems which are basically 3-dimensional in nature can sometimes be simplified into a 2-dimensional configuration and still provide some very good indications of the response. An example of this type of simplification is the analysis of a side impact between a locomotive and a shipping cask. Figure 3 illustrates the simplified model (undeformed finite element mesh) and a calculated deformed shape for an instant of time that is 15 milliseconds after the locomotive made contact with the cask. This model also utilized the HONDO computer program and its sliding interface features. A number of conservative simplifying assumptions were made in the development of this model. The results provided much insight as to the response of the system and indicated that the cask would not be breached by such an impact.

Three-dimensional modeling of these types of impact problems where large deformations are encountered requires very large amounts of computer time and is only feasible for simple problems. However, as the state of the art progresses, it is expected that 3-dimensional analyses will become practical. At the present time, they are very difficult to accomplish. When a complete 3-dimensional analysis is desired, scale modeling becomes a more practical analysis technique.

PHYSICAL SCALE MODELING

Scale modeling has proven to be a very practical method for modeling complicated and severe impact problems [3,4,5,9,10]. The theoretical basis for physical scale modeling originates from dimensional analysis studies which indicate that if material properties are the same in the model and the prototype, then the deformations will be similar given that the impact velocities are the same. Two restrictions which are placed on this technique are that the material strain-rate effects and gravity effects should be negligible. The first restriction is seldom a problem. The second one is not a concern when one is analyzing high velocity impacts where the structures are subjected to decelerations which are much higher than gravity.

Theoretically the gravity field in a subscale test should be greater than the earth's by the scale factor (The scale factor is the inverse of the scale. For example, in a quarter-scale test the scale factor is 4). Since the gravity field cannot be conveniently scaled, gravity effects must be assumed to be negligible. Experience has shown that this is a valid assumption in most impact problems where decelerations having high g numbers are generally encountered. One item which should be noted, however, is that the rebound phase of the impact will not scale correctly; subscale models exhibit much higher rebounds. This phase of the impact however is not of concern as secondary impacts are generally much less severe and the significant container damage, if any occurs, will happen during the primary impact.

The basic relationships which exist between the model and the prototype parameters are listed below. These are valid for impact velocities which are equal between model and prototype. Here the subscript m denotes the model, p the prototype, and the letter n denotes the scale factor. The basic relationships are

Deflection	$Y_m = Y_p/n$
Stress or Pressure	$S_m = S_p$
Force	$F_m = F_p/n^2$
Energy	$E_m = E_p/n^3$
Mass	$M_m = M_p/n^3$
Time	$T_m = T_p/n$
Acceleration	$A_m = n(A_p)$

In particular it can be seen that deformations (deflection) in the model and prototype are similar. From the time relation, it can be seen that events in the model occur faster by the scale factor. Thus accelerations in the model are higher by the scale factor.

Previous work [9,10] using scale models involved experimental work using exact replica models for verification of the analysis. The model and the prototype were constructed of similar materials and were geometrically scaled. In both cases, agreement between the model and the prototype was found to be excellent.

The structures for the tests described were considerably more complicated and the additional assumption that a somewhat simplified model will give valid results was made. Substitution of an approximate (simplified) model, as opposed to an exact geometrical model was often done for practical considerations and was necessary in order to reduce costs. Such a model is usually termed an "adequate" model (11). The models described in this paper were designed according to this concept, although the shipping cask models were very detailed.

In the scale modeling work performed at Sandia National Laboratories, one-eighth and quarter-scale models have been successfully utilized to model shipping casks and complete shipping systems, including vehicles.

Figure 4 illustrates photographs of eighth scale models used to analyze the tractor-trailer impact and the grade crossing test. These tests were conducted at a sled track facility where the models were accelerated up to the impact velocities by small rocket motors.

The scale model tractor-trailer test resulted in the cask impacting the target in very close to an end-on condition. This impact resulted in slight "mushrooming" of the head end of the cask but did not threaten containment. The scale model grade crossing impact test resulted in two small deformations on the side of the cask where the longitudinal locomotive underframe members impacted the cask. The containment of this cask was also not threatened. These results will be discussed further below.

FULL-SCALE TESTS

Full-scale tests were conducted to confirm the results of the analyses techniques and to obtain data on the response of full-scale systems. Large rocket motors attached to a pusher sled were used to propel the vehicles up to speed on a test track which guided the equipment to the impact point. Used surplus equipment was used for all the tests. Data were obtained by use of instrumentation and high speed photography.

Figure 5 is a photograph of the full-scale tractor-trailer system that was impacted into a rigid barrier at 135 km/hr (84 mph). At this velocity the vehicular system was completely destroyed. The shipping cask plowed through the cab and completely crushed the impact limiter. The cask then impacted the target in close to an end-on condition. The complete impact event took approximately 0.2 second with the cask (system) rebounding slightly from the target as is seen in Figure 6.

The response was extremely close to what the model had predicted and the cask was only slightly deformed on the impact-end. At the worst point there was approximately a 3% diametral expansion.

Figure 7 illustrates the full-scale locomotive and tractor-trailer rig used for the grade crossing test. In this test the cask was hit broadside by the locomotive at a velocity of 131 km/hr (81 mph). Normal placement of the cask on its shipping trailer resulted in an impact of the top of the locomotive frame 20.3 cm (8 in) below the centerline of the cask. Figure 8 is a photograph obtained from the high speed films. The underframe impact against the cask caused two local indentations where the ends of the underframe I beams made contact. These were approximately 2.5 cm (1 in) deep on the outer shell but did not propogate to the interior shell. The underframe impact imparted a spin to the cask causing it to roll into the locomotive superstructure. The cask attained a horizontal velocity of about 20 m/s (65 ft/s) in about 0.1 second and the trailer structure was completely wrapped around the locomotive in about 0.2 second.

The results of the full-scale tests confirmed the analyses results. In both tests the behavior predicted by both the analytical analysis and the scale models was observed. The correlations between analyses and full-scale tests are discussed further below.

DISCUSSION

The high speed films from both tests were digitized in order to obtain kinematic data for making comparisons with the analyses. Post-test comparisons of the hardware were also made. Figure 9 illustrates the results of the lumped parameter model for the velocity-time history of the cask as well as the results from the full-scale test. As can be seen, the results agreed quite well. The model not only predicted the time duration of the impact but also the velocity the cask would have upon contacting the hard target.

The finite element model results for this test proved to be somewhat conservative in that they overpredicted the "mushrooming" on the impact end of the cask by about a factor of two. This is attributed to a number of conservatisms in the model. The eighth-scale test for this head-on impact produced results which very closely predicted the results of the full-scale test. The deformation pattern on the impact end of the model cask was virtually identical to what was observed in the full-scale test.

The finite element model developed to analyze the grade crossing test also proved to be conservative as it overpredicted damage to the cask. This again is attributed to assumptions made in formulating a 2-dimensional model for a 3-dimensional problem. The finite element model, nevertheless, gave very good indications of what the cask response would be. It indicated that the corner of the locomotive underframe would be crushed and that the cask would roll up into the superstructure and sustain some deformation in the area of contact with the underframe. The results also indicated that the deformations would not be severe enough to cause the cask to be breached or to seriously threaten its containment.

The one-eighth scale locomotive and cask models for the grade crossing test gave excellent results in terms of predicting damage to the full-scale hardware. Figure 10 is a composite photograph showing the full-scale and model casks after testing. Both casks exhibited indentations where the longitudinal underframe members impacted the cask. Because of the odd shape of the indentions, it is difficult to make quantitative comparisons. However, it is estimated that the deformation depth in the outer shell of the casks agreed within about 15%. Neither the full-scale nor the model cask had any detectable distortions of their inner cavity.

CONCLUSION

The analysis techniques that have been briefly discussed in this paper have proved to be very effective methods for predicting the response of shipping casks subjected to very severe and mechanically complicated impact environments. Lumped parameter modeling was found to give very good results for the overall system response. The finite element calculations were found to give conservative results in that they overpredicted deformations to the casks. This is attributed to conservative assumptions in the model formulations. the results, however, are still considered quite good and useful in that they provide the analyst with much insight ito the mechanics of the problem and the response of the hardware.

Physical scale modeling has proved to be a very practical and reliable method for the analysis of complicated problems such as those discussed above. Complicated and severe hypothetical accident situations can be investigated with a high degree of confidence that the results will very closely parallel what would happen in the real situation.

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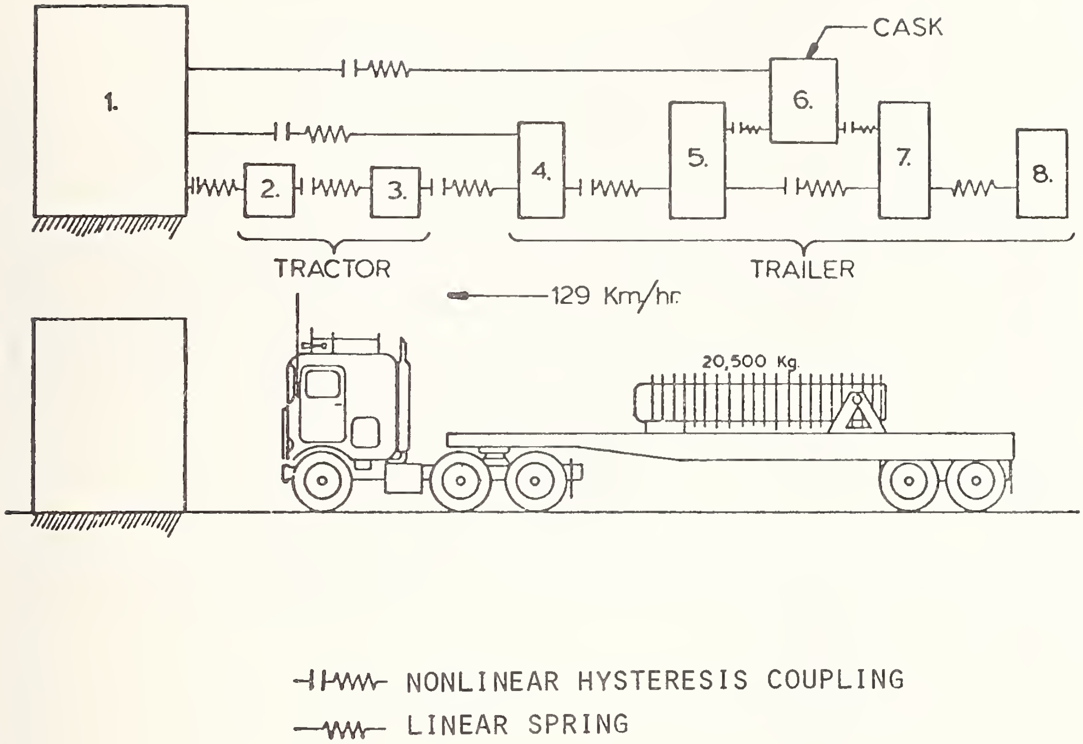

Figure 1. Illustration of the schematic of a lumped parameter model for analyzing the impact of a tractor-trailer system into a rigid barrier. This model provides an estimate of the velocity with which the cask impacts the hard target.

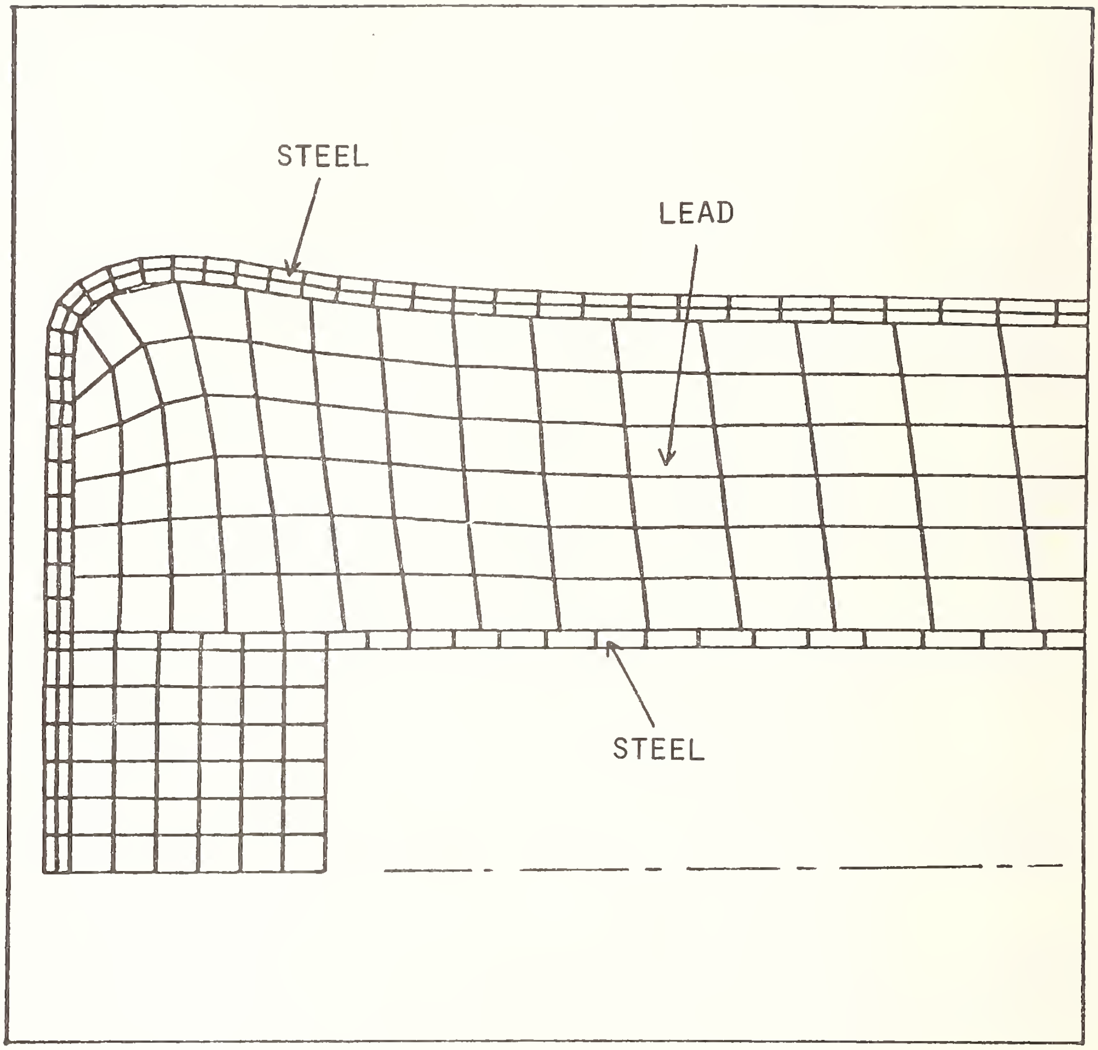

Figure 2. Illustration of the mesh for the finite element analysis. This figure shows the calculated deformed shape for the impact-end of the cask.

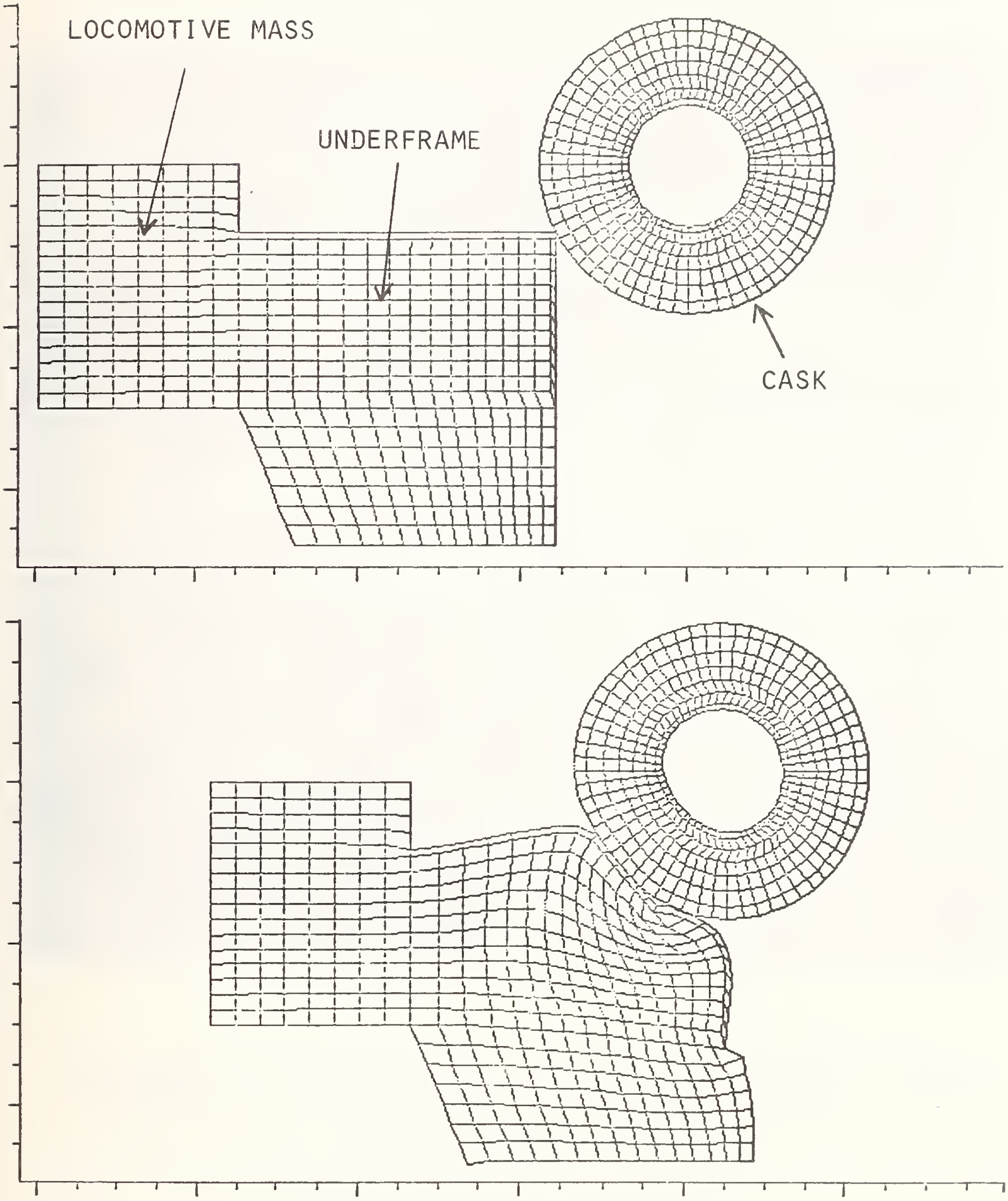

Figure 3. Undeformed and deformed shapes for the mesh used to perform the finite element analysis for the grade crossing test. The deformed shape is at .015 second.

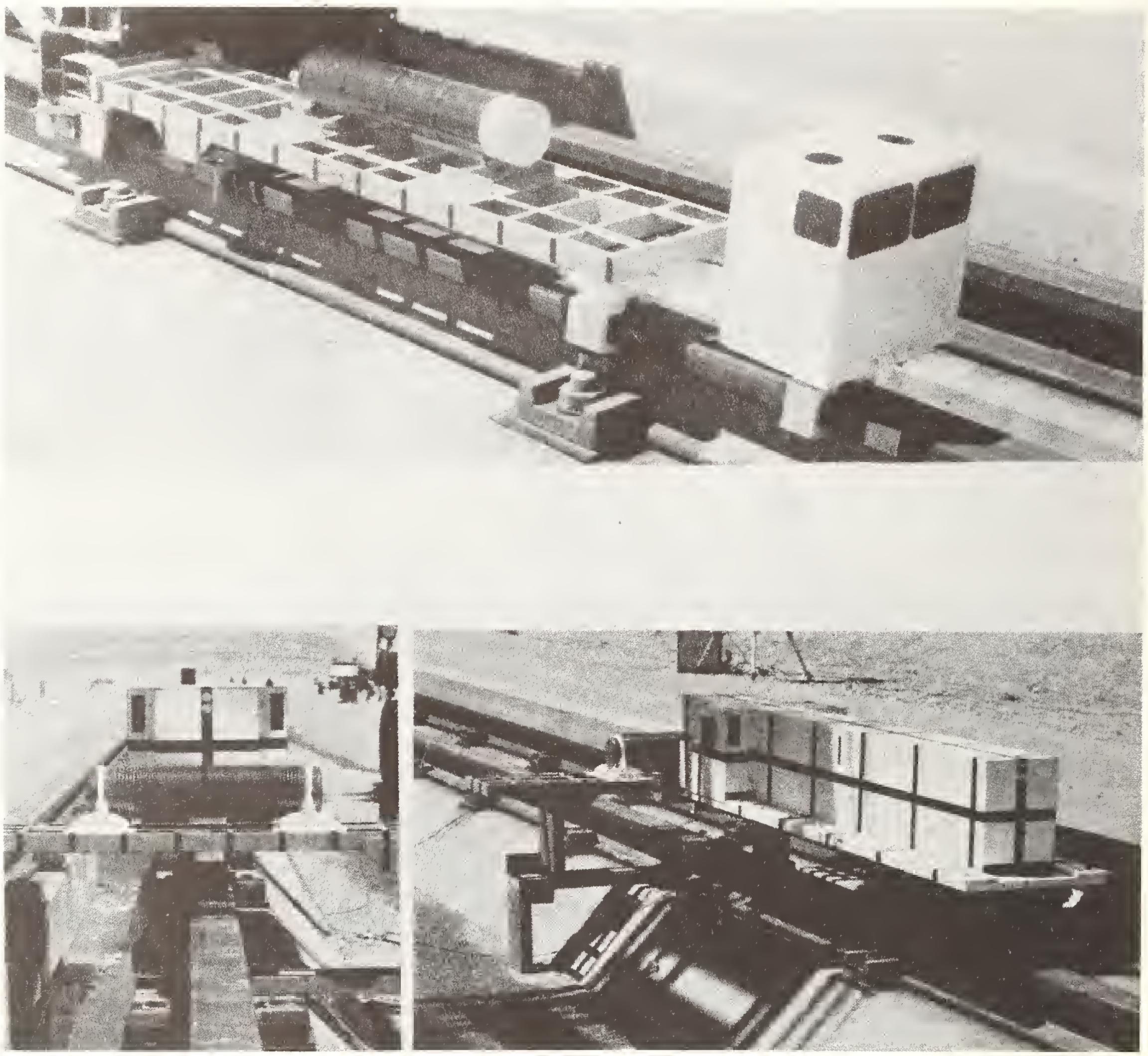

Figure 4. Photographs of the scale models utilized to analyze the head-on tractor-trailer and the grade crossing tests.

Figure 5. Photograph of the full-scale tractor-trailer system.

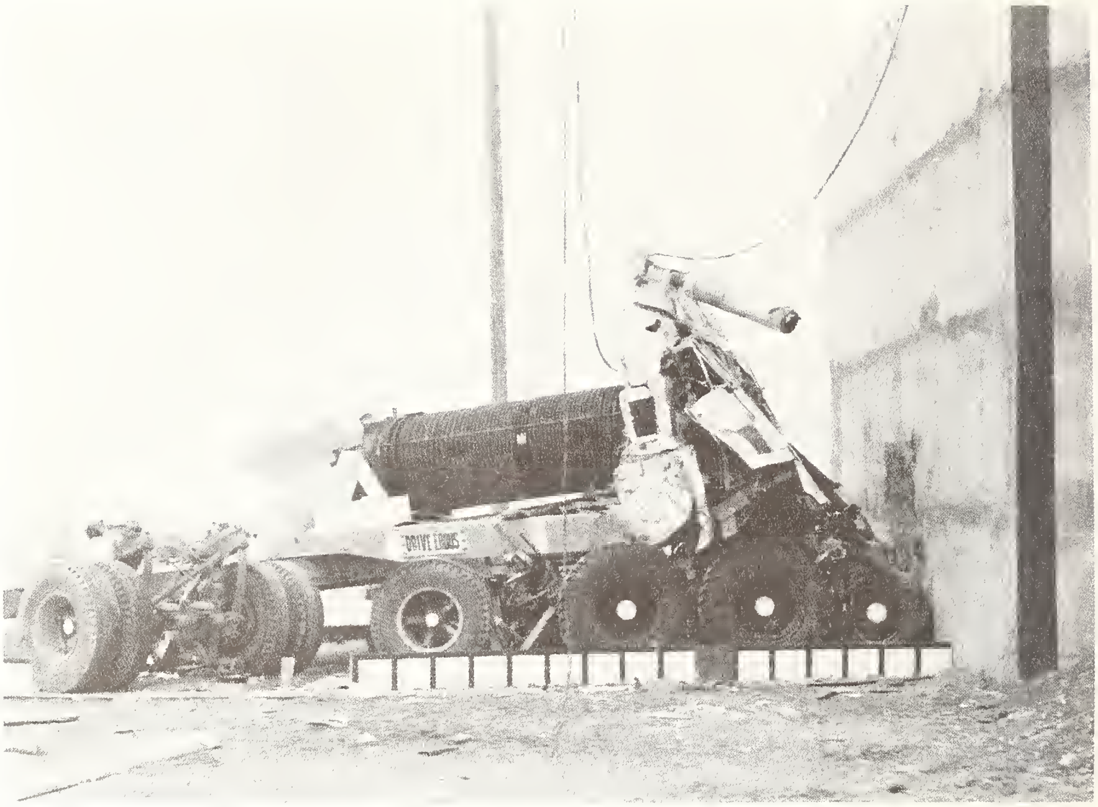

Figure 6. Photograph of the full-scale tractor-trailer system after the crash.

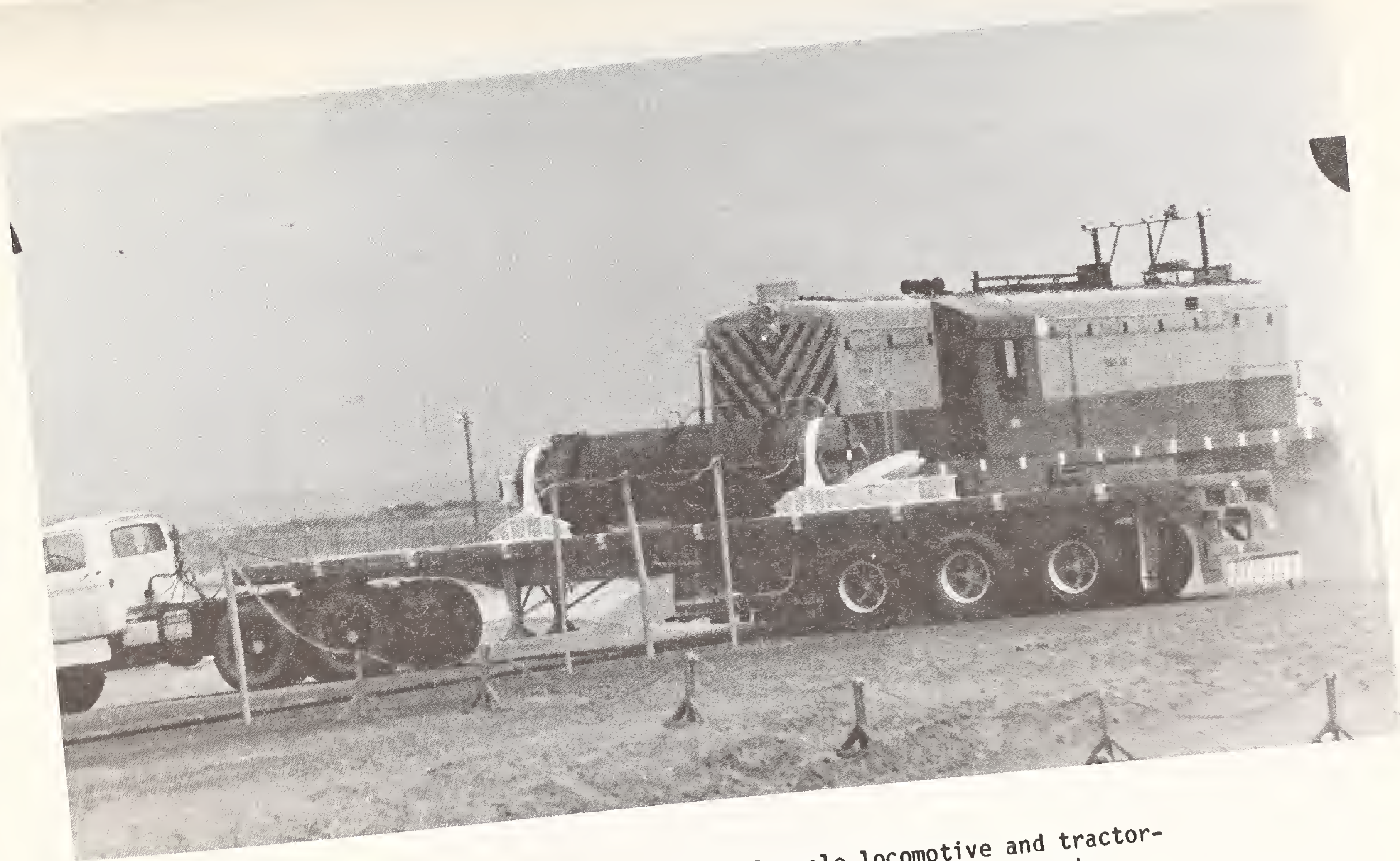

Figure 7. Photograph of the full-scale locomotive and tractor-trailer system used in the grade crossing test.

Figure 8. Photograph taken during the grade crossing impact.

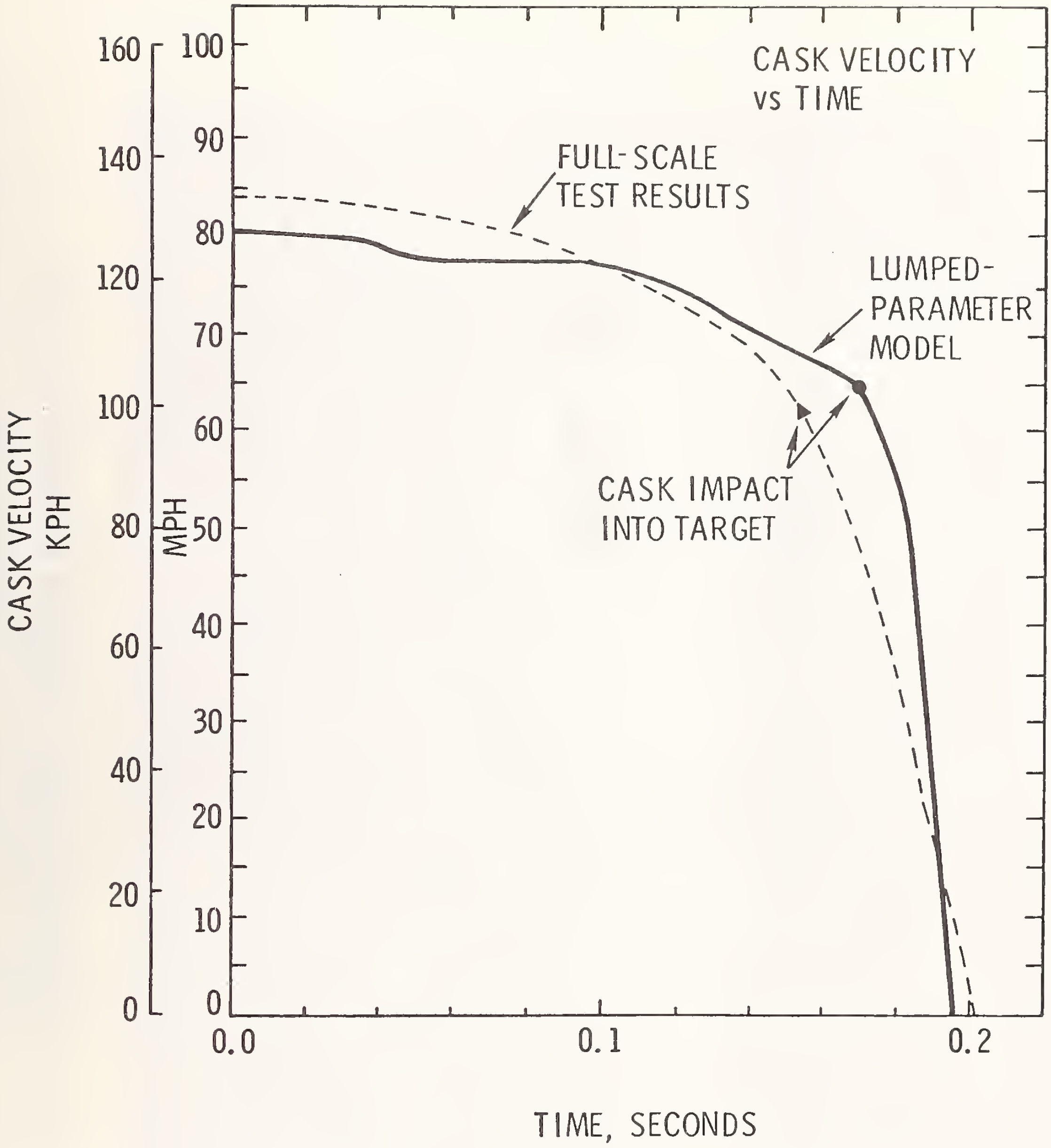

Figure 9. Results from the lumped parameter model and the full-scale test results for the velocity-time history of the cask.

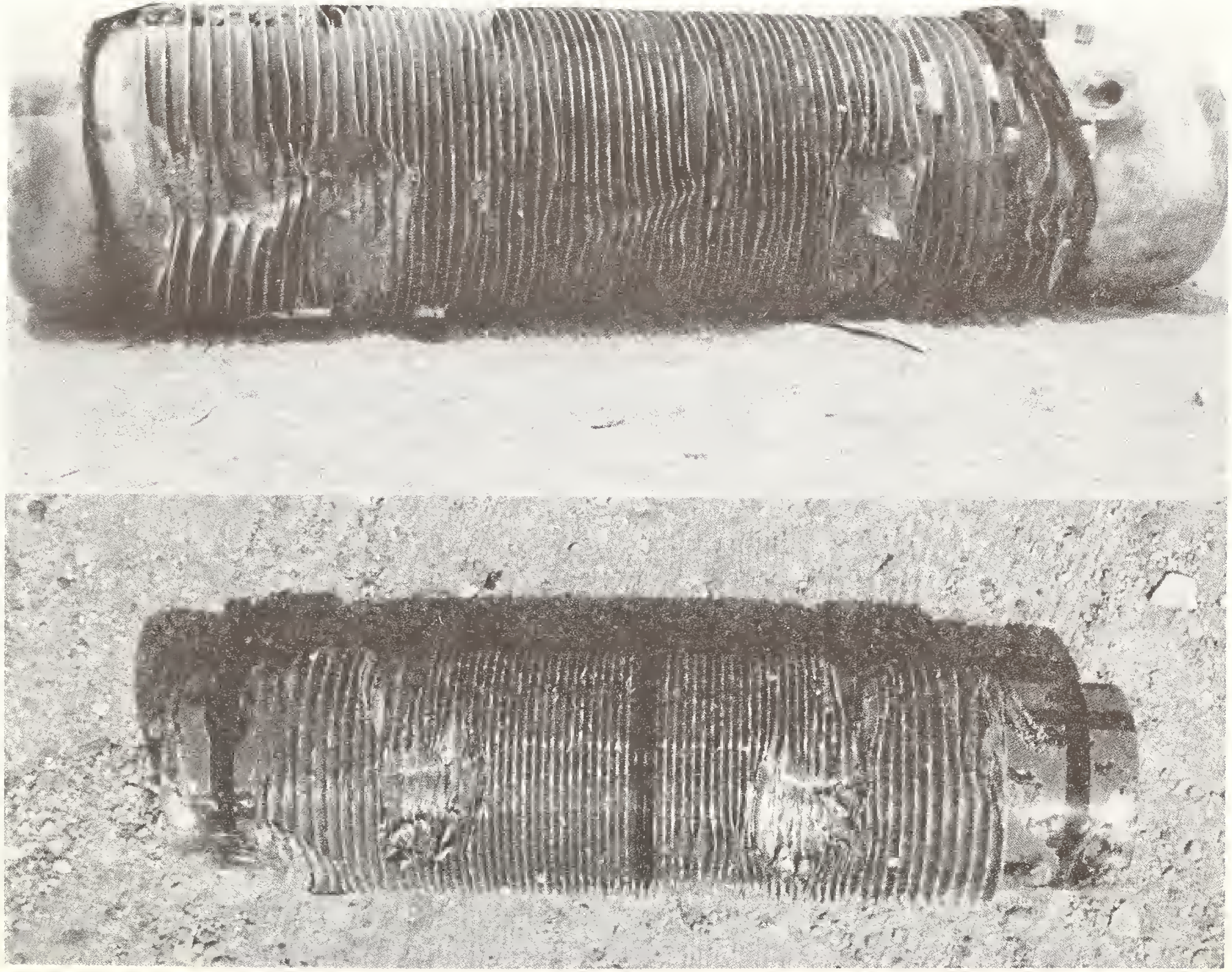

Figure 10. Photographs of the full-scale and model casks after testing.

RESPONSE OF RADIOACTIVE MATERIAL WASTE DRUMS TO ACCIDENT ENVIRONMENTS*

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ABSTRACT

Using computer analyses, subscale model tests, and selected full scale tests, it is possible to predict the structural behavior of 208 liter (55 gal) drums resulting from accident environments. This was verified by analyzing and testing DOT-17C drum arrays containing simulated contact-handled transuranic waste (CH-TRU) in simulated accident environments.

This paper briefly describes the results of the static and dynamic tests using full scale 17C drums and compares them to the results obtained from subscale tests and computer analyses. Methods of eliminating or minimizing the damage to the drums are mentioned. More detailed information can be found in Ref. [1].

Key words: accident environments; accident response; computer analysis; contact-handled transuranic waste; scale model testing; Type A containers; 55 gallon drums.

INTRODUCTION

The Transportation Technology Center (TTC) at Sandia National Laboratories (SNL) is currently involved in programs addressing the transport of nuclear materials. The structural response of 208 liter (55 gal), 17C drums containing contact-handled transuranic waste (CH-TRU) is being evaluated as part of a program to predict the

* This work was supported by the U.S. Department of Energy Contract DE-AC094-76-DP00789.

** A U.S. Department of Energy Facility.

response of low-level nuclear waste transportation systems to accident environments. Early analyses of the entire transportation system indicated that the drum behavior had a strong effect on the structural response of the system and must be included in the evaluation. It is also desirable to predict the failure of the drums as part of the assessment of system response in accident environments to serve as input to environmental impact statements.

In order to determine the adequacy of predicting CH-TRU drum response utilizing existing computer codes and subscale models, static and dynamic tests using full scale drums were conducted. The results of the computer analyses and subscale tests were then compared to the full scale data.

DESCRIPTION OF PROTOTYPE AND SUBSCALE DRUMS

The full scale drums used in the tests were obtained from Rockwell International's Rocky Flats Plant. This assured that the drums tested were the same as those used to package low-level waste. The drum assembly, Figure 1, consisted of a thin polyethylene bag filled with waste contained inside a rigid polyethylene liner. This liner is in turn contained by the 17C drum. Full scale drums were tested in four configurations: filled with 91 kg (200 lbs) of simulated combustible waste, filled with either 227 or 318 kg (500 or 700 lbs) of simulated sludge waste, or empty. The weights of the filled drums were representative of those shipped from Rocky Flats.

Scale model drums were simulated by using appropriately sized, tinplated steel cans normally used for food packaging. In the tests being reported, 1/4 scale drums were modeled using #12 cans. The diameters and heights scaled to within 3 percent, but the material in the scale model was 15 percent thinner. This was offset due to the fact that the yield strength of the material was 70 percent higher. The scale model drums were filled to the appropriate weight with material taken from the prototype drums or with locally obtained material of similar composition. The lids were attached to the cans using a commercially available can sealer.

No attempt was made to model the polyethylene bag, rigid liner, or the lid clamping ring. Prior to the tests, it was felt that the difference in the results obtained would not justify the added expense of modeling these secondary features.

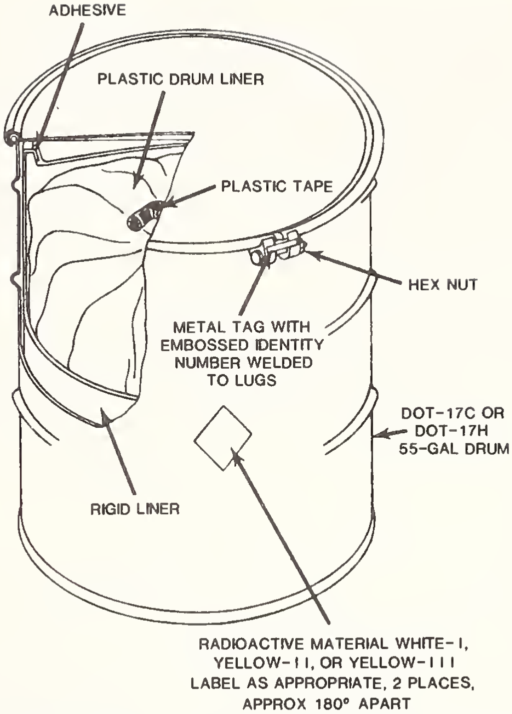

Figure 1. Typical 17C drum, Type A Waste Container with a Rigid 2.5-mm-thick Polyethylene Liner

STATIC TESTS

The first tests consisted of quasi-static crush tests of the full scale and subscale drums. The results of the tests would identify the failure mode and quantify the load-deflection behavior of the prototype drums. An early estimate of the adequacy of the subscale models was obtained by comparing their load-deflection data with the scaled prototype data. If the correlation had not been reasonable, the containers could not have been expected to correlate in the dynamic tests, and a different subscale model would have been obtained.

The full scale and subscale model drums were tested in universal testing machines equipped with rigid platens large enough to assure adequate contact area during crush. The crush force versus platen displacement was obtained graphically from recording equipment on the testing machines. Time lapse motion pictures were also taken to document the deformations and failure modes of the drums. The drums were crushed in a radial direction at a rate of 2.5 cm/min (1 in/min). The maximum crush of the drums was arbitrarily chosen to be one-half of the drum diameter.

Eight full scale drums were tested along with six 1/4 scale models. Results from the static crush tests of the full scale containers demonstrated very consistent load-deflection behavior (Figure 2). The initial crush phase and the occurrence of lid separation, characterized by the sudden drop in the crush force, can be seen to be independent of the contents. The contents and the amount of free volume in the drum did affect crush stiffness following lid failure and the onset and severity of lockup. The behavior of the drums containing either 227 or 318 kg (500 or 700 lbs) of simulated sludge was virtually identical, while the drums containing 91 kg (200 lbs) of combustible waste tended to be noticeably stiffer.

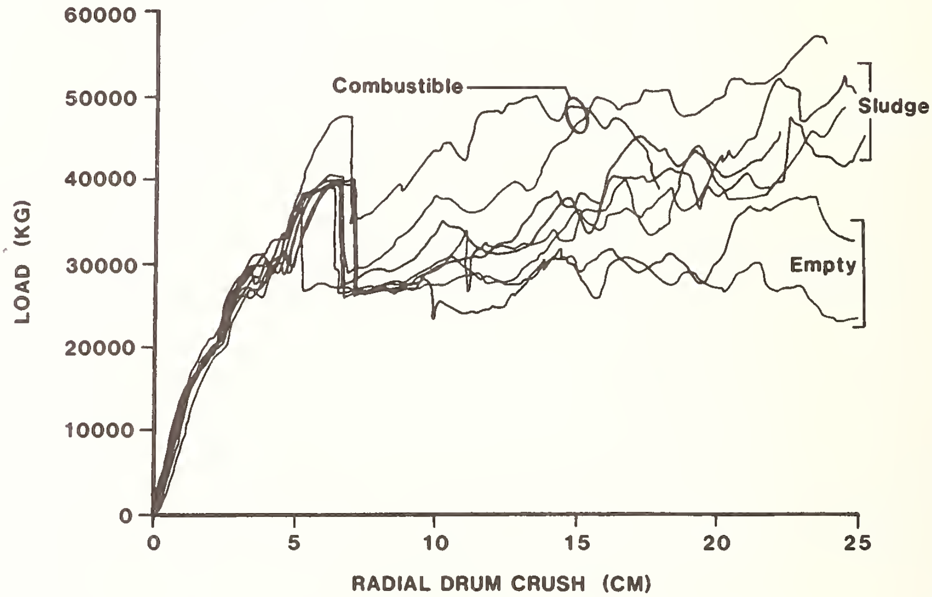

Figure 2. Comparison of the Static Load Deflection Behavior of Prototype Drums - Various Contents

The results of the static crush tests for full scale drums containing 227 kg (500 lbs) of combustible waste are shown in Figure 3 with the results of the comparible 1/4 scale tests, appropriately scaled to the prototype dimensions, superimposed. Good correlation between the prototype and the scale model drums can be seen. This correlation was observed for all the tests.

It was found that the 1/4 scale models could not replicate the separation of the drum lid from the body that was observed at approximately 10 percent diametral crush on the prototype drums. This was due to the fact that the prototype drums used clamping rings to secure the lid while the scale model lids were rolled and crimped to the container body. The behavior of the scale model lids was very similar to that of the bottom of the prototype which was attached in a similar manner.

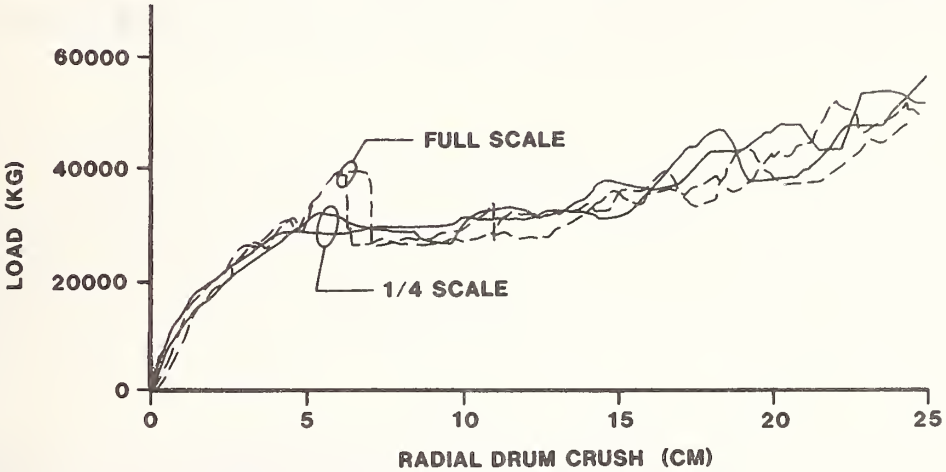

Figure 3. Comparison of the Scale Load-Deflection Behavior for 1/4 and Full Scale 227 kg Containers

DYNAMIC TESTS

A series of dynamic tests was conducted following the completion of the static tests. These tests established the characteristic deformed shapes of the prototype and subscale drums and gave quantitative results of the damage as a function of the impact conditions. Both single and multiple drum configurations were tested at impact velocities that would be expected to occur in a severe transportation accident. The multiple drum configurations included 8 high x 1 column, 2x3, 3x2 and 8x3 container arrays. In one series of tests, a closed cell, rigid polyurethane foam, located between the container and the target, was used to simulate the impact mitigation properties of a typical transport container.

All dynamic tests were conducted at SNL's Coyote Canyon Drop Test Facility. The full scale drums were allowed to free-fall onto an unyielding target. Since the smaller, subscale models tended to rotate during free-fall and impact in undesired orientations, these specimens were guided to the target using small diameter cables passed through guides attached to the cans. The target consisted of a 227 tonne (250 ton) reinforced concrete foundation topped with 10 cm (4 in) of hardened steel plate. In order to maintain vertical alignment of the waste containers in the multiple drum drops, frames constructed of heavy angle iron were utilized. No bulging or excessive deformations of the frames were observed during the post-test inspections. Data collected included measurements of the radial drum deformations, flattened impact area and, where applicable, polyurethane foam deformation.

As expected from the results of the static tests, the correlation between the prototype and the scale model drums was good. Both the radial deformations and the general deformed shapes of both the models and prototype were in agreement. A comparison of the deformed shapes of a prototype and subscale drum impacted at 47 kph (29 mph) is shown in Figure 4. The inclusion of the polyurethane foam had little, if any, effect on the crush behavior of the drums in the multiple drum array impacts.

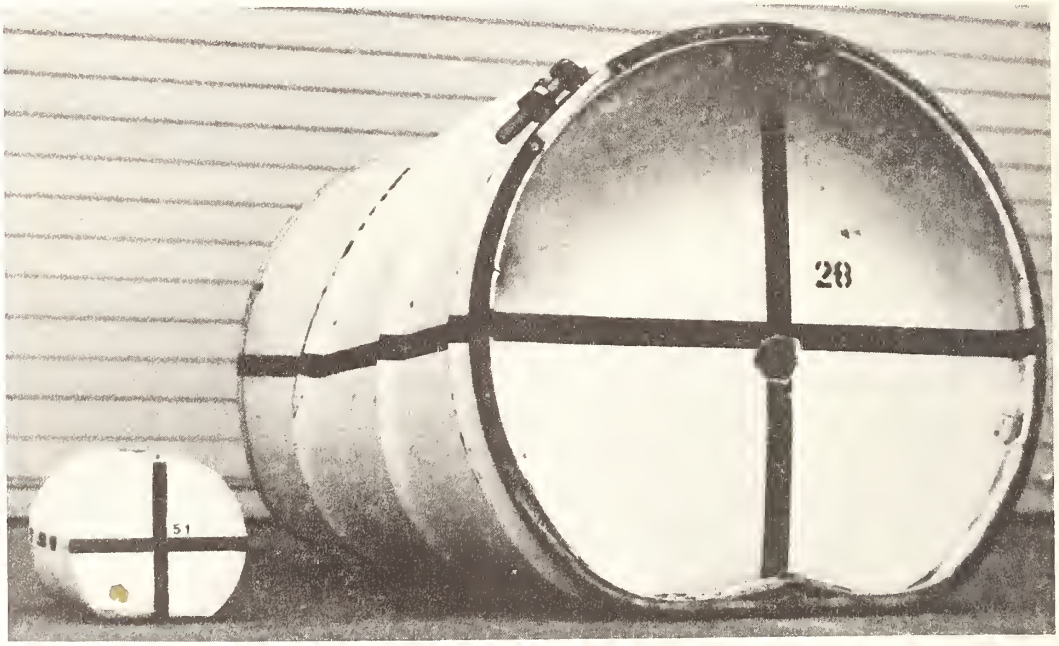

Figure 4. Post-test Appearance of the 1/4 and Full Scale 318 kg Containers Following a 47 kph Impact

As was found in the static tests, the separation of the drum lid could not be visually determined from the subscale models due to the different method of attachment. However, after examining the full scale and subscale static and dynamic tests, it was felt that lid separation in prototype tests could be predicted from the scale model data. In both the static and dynamic tests, the lid separation consistently occurred at approximately 10 percent crush. Therefore, any subscale drum which exhibited more than 10 percent crush after testing would infer that the full scale drum would open in the same test. This criterion can also be applied to predict drum openings from the results of the computer simulations.

ANALYTIC RESULTS

The 8 high x 1 column array configuration was analyzed using existing computer codes to predict drum damage when dropped from either 6.4 or 9 m (21 or 30 ft). The first method used a lumped parameter code [2] to calculate the crush of each drum and the final array height. The drums were modeled in the code as a series of masses and springs which represented the weights and stiffnesses of the drums and contents. The stiffnesses were obtained from the earlier full scale static crush tests.

A finite element model using a SNL developed computer code [3] was also used for damage predictions. In this case the drums were represented as solid cylinders of material. Average material properties were used to obtain the correct drum behavior. These material properties were determined from the previously conducted full scale static crush tests [1].

The crush of individual drums in the eight high array was predicted with reasonable accuracy by both analytical models. In order to compare the analytical results, which predict maximum deformations, to the experimental results, which were final deformations, the former had to be corrected for the elastic spring-back of the drums. This was accomplished by subtracting a measured spring-back of approximately 3.8 cm (1.5 in) from the predicted maximum deformations to determine the final deformation. The results of the lumped parameter model and the finite element model, corrected for spring-back, along with the full scale and 1/4 scale test results for the 9 m (30 ft) drop, are shown in Figure 5. The finite element model slightly overpredicts the crush of the top and the bottom two drums. This result can be attributed to using a solid cylinder to represent the drum. The lumped parameter model gives a somewhat better prediction of individual drum crush, but no indication of the deformed drum shape. However, both models give results which are within the expected repeatability of the tests. Again, it is felt that any drum that has a predicted radial deformation greater than 10 percent of its diameter would show lid separation in a full scale test.

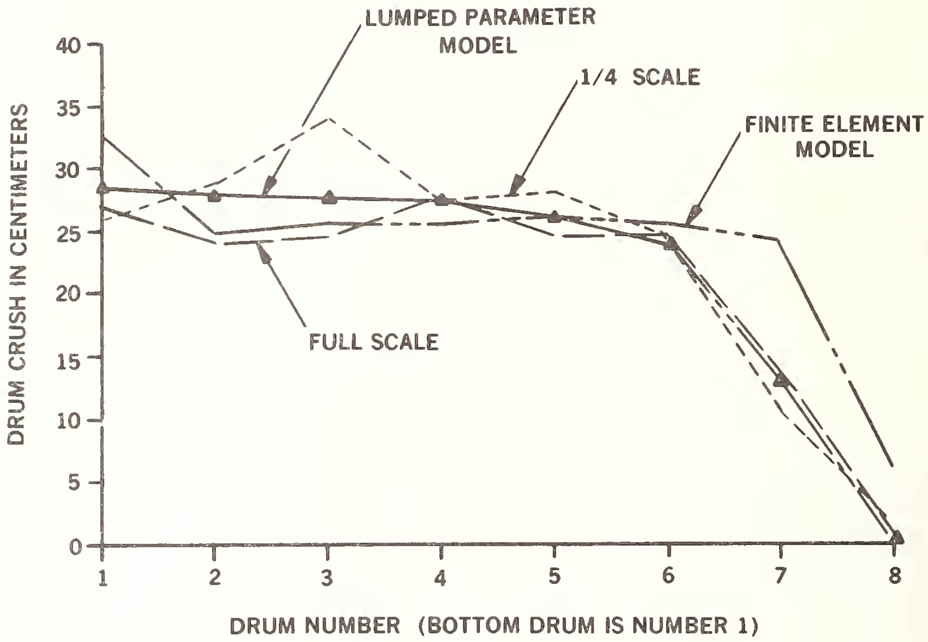

Figure 5. Comparison of Full and 1/4 Scale Test Results to Analytical Results; Eight High Stack, 9 m drop

CONCLUSIONS

Based on the tests that have been conducted, it is felt that the structural response of low-level waste drums to impact environments can be adequately predicted using analytical and subscale models. Both the finite element and the lumped parameter models give results that are expected to be in the range of behavior of the full scale drums. The failure of the drum closure cannot be directly observed from the subscale test results or predicted by the computer analyses, but can be inferred from experimental data.

Results of tests and analyses indicate that a large number of drums would be opened in a hypothethical accident situation. This is caused primarily by the drums near the front being crushed by those in the rear of the multiple drum arrays. The damage to the drums could be reduced by providing partitioning to separate the drums thus eliminating the damage caused by the inertial effects of the rear drums.

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DESIGN OF PACKAGING FOR TRANSPORTING TRANSURANIC CONTAMINATED WASTES*

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Abstract: Contact-handled transuranic (CH-TRU) waste continues to be generated and temporarily stored at a number of locations in the United States as a by-product of national defense programs. The Transportation Technology Center at Sandia National Laboratories has assumed the lead lab responsibility for development of safe, efficient, licensable, and cost-effective transportation systems to be used in the management of this waste. The TRansUranic PACkage Transporter (TRUPACT), a Type B packaging, will be transported by rail or truck and will be compatible with Type A packagings used by waste generators, interim storage sites, and repositories. Developing an efficient interface with each facility is being given a high priority

CH-TRU waste is typically packaged in steel drums, fiberglass reinforced plywood (FRP) boxes, or a variety of steel boxes. The waste typically consists of plutonium contaminated metal scraps, sludge, paper, filters, and other materials resulting from production of weapons grade nuclear materials and from reprocessing operations. The activity of the waste is low with the maximum container surface dose rate being less than 200 mrem/hr.

TRUPACT is required to be a Type B packaging since larger than Type A quantities of some radionuclides (particularly plutonium) may be transported. Waste packagings inside of TRUPACT are not required to be Type A and could be any standard industrial container that facilitates handling. The TRUPACT must provide structural and thermal protection

*Work sponsored by the U. S. Department of Energy under Contract DE-AC04-76-DP00789.

to the waste in accident environments as specified in DOE 5480.1 Chg. 3 (5/1/81), DOT regulations 49 CFR 173 and NRC regulations 10 CFR 71. No radiation shielding or internal heat removal mechanism is necessary for this waste.

The TRUPACT consists of inner and outer steel sheathed tubular frameworks separated by polyurethane foam. Stainless steel plate is used internally to facilitate decontamination. The inner containment structure has a hinged door that uses elastomeric seals. The polyurethane foam in the walls and ends of the package provides impact energy mitigation and fire protection. The outer tubular framework of the TRUPACT is covered with carbon steel plate and has a steel door. Stainless steel puncture plates are included on the inner surface of the outer tubular steel framework to prevent puncture of the inner containment cavity.

Preliminary design of the packaging has been completed and effective methods have been employed to prevent failure under both the normal handling and hypothetical accident conditions. This paper will describe design criteria adopted, mechanical and physical features of the packaging, and packaging features included to meet regulatory test conditions. In addition, analyses that have been conducted will be summarized, scale model tests that have been performed will be discussed, and the program schedule through delivery of first production units will be outlined.

Key words: Nuclear waste, contact-handled waste, transportation, Type B packaging, packaging, overpack, truck transport, rail transport, structural analysis, thermal analysis, puncture analysis, testing, scale model testing.

Introduction

The Contact-Handled Transuranic (CH-TRU) waste transportation system under development at Sandia National Laboratories is known as the TRUPACT (TRansUranic PACkage Transporter). The TRUPACT is a Type B packaging concept which is being developed to transport a variety of low-level, transuranic contaminated waste forms.

Type A packagings are normally used to transport low concentrations of radionuclides and must be shown to withstand, without loss of integrity, test conditions which simulate environments encountered in normal transportation as specified in 49CFR173.398, Section b. A Type B packaging, in addition to satisfying the Type A packaging requirements, must also be shown to withstand, without loss of package integrity, the hypothetical accident conditions of transportation given in Section c of 49CFR173.398.

A Type A packaging may be used for containing and transporting a Type A quantity or less of radionuclides. Greater than Type A quantities may only be transported in a Type B packaging. The Type A and Type B quantities of a particular nuclide are determined by the nuclides transport group. Specific information concerning transport groups and Type A and B quantities is given in 49CFR173.389 and 49CFR173.390.

Type B packaging is required for transporting CH-TRU waste since greater than Type A quantities of some radionuclides (particularly plutonium) may be present in individual or groups of CH-TRU waste containers. The TRUPACT is being designed to meet the existing Type B requirements. An awareness of possible changes in the regulations will be maintained; the effect of such changes on the TRUPACT design will be examined to determine if design alterations are required.

The TRUPACT conceptual design is currently a large metal box which will be available in both rail and truck transported versions. The packaging dimensions and capacities are:

<u>Rail Configuration</u>

Overall Dimensions

Length: 23 ft, 2 in. (7.1 m)
Width: 10 ft, 0 in. (3.0 m)
Height: 10 ft, 11 in, (3.3 m)

Inside Dimensions

Length: 17 ft, 2 in. (5.2 m)
Width: 7 ft, 8 in. (2.3 m)
Height: 8 ft, 7 in. (2.6 m)

Weights

Empty Packaging: 38,000 lb (17.2 tonne)
Cargo Capacity: 32,000 lb (14.5 tonne)
Gross Weight: 70,000 lb (31.7 tonne)

<u>Truck Configuration</u>

Overall Dimensions

Length: 24 ft, 8 in. (7.5 m)
Width: 8 ft, 0 in. (2.4 m)
Height: 9 ft, 0 in. (2.7 m)

Inside Dimensions

Length: 18 ft, 8 in. (5.9 m)
Width: 6 ft, 0 in. (1.8 m)
Height: 7 ft, 0 in. (2.1 m)

Weights

Empty Packaging: 28,000 lb (12.7 tonne)
Cargo Capacity: 17,000 lb (7.7 tonne)
Gross Weight: 45,000 lb (20.4 tonne)

Truck Transport Design Basis

The transport semi-trailer and tractor are being designed to operate as a legal system throughout the United States, i.e., at a Gross Vehicle Weight (GVW) of 73,280 lb (33.3 tonne) or less. The weight distribution will be such that the limiting loads of 32,000 lb (14.5 tonne) for a tandem axle, and 9000 lb (4.0 tonne) for a single steering axle are not exceeded. Most states now have GVW limits of 80,000 lb (35.3 tonne) or more; however, the practical limit for a tandem axle semi-trailer with a three-axle tractor is about 78,000 lb (35.4 tonne). In states that use "bridge formulas", the distance between the steering axle and the rear axle of the trailer is used to limit the allowable GVW.

The most restrictive limits for a five-axle tractor and semi-trailer combination according to a 1979 chart published by the Truck Trailer Manufacturing Association are

Length - 55 ft, 0 in. (16.8 m)
Width - 8 ft, 0 in. (2.4 m)
Height - 13 ft, 6 in. (4.1 m)

The position of the packaging on the trailer will be adjusted to uniformly distribute the load between the tractor and semi-trailer axles. The Type B package will be attached to the trailer bed using standard container mounting hardware (as recommended by the International Organization for Standardization (ISO)) which has been installed into the trailer frame to provide rigid attachments. The tiedown hardware will be designed to provide a static strength equal to 2, 5, and 10 times the maximum weight of the loaded packaging in the vertical, transverse, and longitudinal directions, respectively (10CFR71.31 (d)). Design loads for packaging analysis will be determined using the test conditions discussed in the introduction material.

Rail Transport Design Basis

The railcar transported TRUPACT option will provide a system that can be used throughout the U.S. without restrictions. To permit unrestricted interchange of railcars, the Plate B Envelope dimensions imposed by the American Association of Railroads (AAR) have been selected. The railcars used will have a minimum GVW of 200,000 lb (90.8 tonne) and a minimum cargo capacity of 140,000 lb (63.5 tonne), i.e., two rail TRUPACTS loaded to their maximum rated capacity.

The TRUPACT packagings will be attached to the railcar using moveable ISO tiedowns located in the railcar flatbed surface. The tiedown hardware will be designed to provide a static capacity of 2, 5, and 10 times the maximum weight of a loaded packaging in the vertical, transverse, and longitudinal directions, respectively, (10CFR71.31 (d)). The overall length of the railcar will be approximately 60 ft (18.3 m). Design loads for packaging analysis will be determined using the normal and hypothetical accident test conditions referenced in the introduction.

Reference Design Description

The TRUPACT concept is illustrated in Figure 1. The structure consists of inner and outer tubular-steel frameworks separated by rigid polyurethane foam and sheathed with steel plate. Package weight is minimized by using a ductile high strength, low alloy steel in both frameworks.

The outer framework is covered with thin mild steel or stainless steel sheet. The packaging cavity is lined with stainless steel because of its ductility and ease of decontamination. The inner structure is made secure by closing and bolting into place a hinged 5 in. (13 cm) thick door which also has an internal tubular steel framework and is filled with foam. This door is sealed with dual elastomeric seals. An exterior door is attached to the outer framework with hinges, is 36 in. (90 cm) thick, utilizes a single elastomeric seal, and is also bolted in place for transport.

There is approximately 36 in. (91 cm) of foam in the packaging ends and 14 in. (36 cm) and 12 in. (30 cm) in the walls of the rail and truck configurations, respectively, to provide thermal protection and to absorb impact and puncture energy. Puncture protection is provided by placing ductile stainless steel plates on the inside surface of the outer framework to distribute the penetrator energy over a large area of low strength polyurethane foam.

Cargo loading and unloading will be accommodated by using either a rolling pallet or roller conveyor system on the floor of the TRUPACT. Heavy handling equipment cannot be used inside the packaging because the floor covering is thin steel sheets supported on and attached to widely spaced tubular steel framework members and the rigid foam between the framework has a relatively low compressive strength. Figures 2 and 3 show the rail and legal weight truck versions, respectively, of TRUPACT loaded with 2 x 3 arrays (six-packs) of 55-gallon drums.

Failure Prevention Design Details

The four primary assaults which could cause package failure addressed during the TRUPACT preliminary design are impact, puncture, thermal, and pressure. Secondary conditions possibly leading to package failure not addressed in this paper are vibration and immersion. Vibration and immersion can generally be handled primarily by selection of proper materials or fabrication methods.

Impact failures which might occur during normal or hypothetical accident test conditions are prevented (1) by selecting steel materials with high ductility, (2) by surrounding the inner structure with rigid polyurethane foam having a low density (6 lbs/ ft^3 or 96 kg/m^3), and (3) by configuring the framework geometry to distribute impact loads and retain the original structure's geometry particularly in the closure seal area. The inner framework is lined with stainless steel sheet having a thickness of 0.13 in. to 0.20 in. (.3 cm to .5 cm) which is a highly ductile material that can absorb very large deformations without loss of integrity. Similarly, the frameworks are of high-strength low-alloy steel having a yield strength of 80 Ksi (550 MPa) which when continuously supported by the inner or outer steel covering can absorb large deformations without failure. The inner and outer frameworks are keyed into the foam during foam placement thus alleviating the need for additional mechanical connections between the frameworks. Shear strength in low-density foam is low but the keying action of the frame members causes a tremendous amount of foam to be loaded during a dynamic event and eliminates large concentrated loads on the inner frame. The frameworks are also configured to redistribute any concentrated loads to other portions of the frames by using a triangular framing pattern.

Achievement of puncture resistance in the TRUPACT presents a very severe design problem. The hypothetical accident condition for puncture requires that the packaging retain its integrity when dropped 40 in.(1 meter) onto a 6 in.(15 cm) diameter rigid steel pin. The reference design for preventing failure during a puncture event has continuous sheets of stainless steel on the inside surface of the outer frame. During the puncture event these stainless steel plates are displaced into the rigid foam, which backs the plates, and spreads the load over a large area of low compressive strength foam. The soft foam alters the puncture load to change it from an impulsive load to one of long duration, thus, lowering the magnitude of the peak force. The outer sheet of steel and thin layer of foam material on the outside of the stainless steel puncture plates also tend to blunt the punch edges and increase the effectiveness of the puncture plates.

The effects of the severe thermal environment presented by the totally engulfing pool fire test at 1475°F (800°C) for 30 minutes will be negated by careful foam selection. The 12 to 14 inches (30 to 36 cm) thickness of foam in the TRUPACT walls provide a large amount of insulation. One dimensional thermal analysis has indicated that an 8 in. (20 cm) thickness of the foam type currently being considered will prevent significant temperature rise on the inner surface after one hour of exposure to the engulfing fire.

Changes in external pressure, upward to 1.7 atmospheres or downward to 0.5 atmosphere, must also be analyzed to evaluate their effect on the structure. The pass/fail criteria for the packaging response to the pressure change is that the containment vessel will suffer no loss of contents. The walls of the TRUPACT are flat panels but the manner of construction produces a sandwich panel with excellent rigidity. The foam core is covered on the inner and outer surfaces with the metal plates and framework members. Deformations predicted by analyses indicate that yielding may occur but failure is not apparent.

In summary, the key to the success of the TRUPACT design is the mutual interactions of the steel covered framework members, the puncture resistant steel plate on the inside of the outer framework and the foamed-in-place polyurethane foam in the package walls. The rigid foam is an essential element in this design and, as indicated in the above paragraphs, must be selected carefully to perform a variety of functions.

TRUPACT Analyses Performed

Compliance of the TRUPACT concept with federal regulations must be demonstrated by analysis, testing (either full-scale or with scale models and components), or an appropriate combination of each. In this development program evaluation by analysis was emphasized since the TRUPACT is a relatively large and complex structure and procurement of full-scale and, to a lesser extent, quarter-scale models of the package for test purposes is prohibitively expensive and time consuming. A fairly extensive test program, however, was pursued concurrently with the analysis of the package to confirm the validity of numerical predictions as they were obtained. Some results of the analyses are presented immediately below while prominent features of the test program are discussed in the subsequent section. Presently only the rail version of the package has been analyzed but the truck version is similar enough in concept such that similar results are expected when it is finally investigated.

The analysis of the TRUPACT was limited principally to estimating the effects of the following four assaults to the package:

1) impact of the fully loaded package on a rigid target following a 30 ft (9 m) free fall;

2) impact of the fully loaded package on a rigid steel pin following a 40 in (1 m) free fall;

3) exposure of the fully loaded package to a totally engulfing 1475°F (800°C) fire for 30-minutes, and

4) subjection of the package to a 1.7 atm. overpressure.

According to the regulations, the accumulated damage sustained by the package following sequential application of the first 3 assaults must be determined. In the analyses, however, the assaults were considered individually. The results of the analyses and tests suggested that the extent of accumulated damage would not be substantially more severe than that obtained by each individual assault alone.

The impact following the 9 m free fall was considered in the first analyses of the TRUPACT. Three different impact orientations (end-on, side-on, and center of mass over corner) were investigated numerically. Initially, hand calculations utilizing a simple rigid mass model of the TRUPACT's inner frame and cargo were performed to estimate foam characteristics and thickness requirements needed in the package's ends and sides to ensure the inner frame was well protected in end-on and side-on impacts. Following these scoping calculations, consecutively more complex three-dimensional finite element analyses were performed using the ADINA structural analysis code. Some features of these various analyses are indicated in Table 1.

The most sophisticated (as well as most meaningful) analyses of the TRUPACT for the three impact orientations were done utilizing the finite element model which employed 3 dimensional truss structures for the inner and outer fameworks, 3 dimensional solid elements for the foam, and either truss elements or concentrated (nodal) masses for the cargo. Some results obtained in the analysis of the model are given in Table 2.

In addition to estimates for deformation, the ADINA analyses indicated strain levels which might be achieved during the impacts. The strain levels reached in the inner frame of the TRUPACT are of particular interest since, if these strains are limited, it is probable that the integrity of the inner structure will be maintained. The analyses predicted slight yielding in some inner frame members for the end-on and side-on impacts and significant yielding in the corner impact. Apparently this result is an artifact of the conservative finite element model analyzed since no inner frame yielding was evident in quarter-scale drop tests (which are described below) and inner frame yielding ceased when the skin on the inner frame of TRUPACT was included in the computer model as thin shell elements.

TABLE 1. TRUPACT Analyses Series for Three 9 m Drop Impact Orientations

SIMULATED IMPACT	ANALYSIS	MODEL COMPONENTS: Outer Frame	Foam	Inner Frame	Cargo
END-ON	1	-	Nonlinear Spring	Rigid Mass	Rigid Mass
	2	-	Truss Elements	3-D Truss Structure	Nodal Masses
	3	-	Truss Elements	3-D Truss Structure	Truss Elements
	4	3-D Truss Structure	3-D Solid Elements	3-D Truss Structure	Truss Elements
SIDE-ON	1	-	Nonlinear Spring	Rigid Mass	Rigid Mass
	2	-	Truss Elements	Nodal Mass	Truss Elements
	3	3-D Truss Structure	3-D Solid Elements	3-D Truss Structure	Truss Elements
CORNER	1	-	Nonlinear Spring	Rigid Mass	Rigid Mass
	2	3-D Truss Structure	3-D Solid Elements	3-D Truss Structure	Nodal Masses

TABLE 2. Numerical Predictions for Foam Crush in Impact Situations

Impact Orientation	Foam Yield Stress, psi (MPa)	Amount of Foam Crush: in (cm)	Amount of Foam Crush: % of foam available
END-On	50 (0.345)	5.2 (13.2)	14
SIDE-ON	100 (0.690)	1.7 (4.2)	21
CORNER	100 (0.690)	21 (53)	51

The numerical evaluation of the puncture resistance of the TRUPACT was done using both finite element analyses and hand calculations employing an approximate theoretical treatment of the problem. Of the two, the latter approach was emphasized since the finite element analyses were expensive and the hand calculations gave excellent results particularly when substantiated by a parallel test program.

In the approximate theoretical treatment of the puncture problem fully plastic circular stainless steel plates supported on a foam foundation were considered. Expressions were developed for the amount of strain energy in the plate and energy expended in deforming the foam as the punch deflected the system. By equating the sum of these energy expressions to the total change in gravitational potential energy of the system during the puncture test, it was possible to obtain an estimate of the peak deflection and strain levels obtained in the plate as a function of plate thickness and yield stress of the steel and foam. Based on this theory and correlations with the concurrent test program, it was estimated that a puncture plate thickness of about 0.5 in (1.3 cm) would be sufficient to prevent penetration and unacceptable deformation in a fully loaded rail TRUPACT punch-drop test.

Thermal analyses have been performed using the Aerotherm Charring Material Thermal Response and Ablation (CMA) Computer Code.* Foam samples have been tested for a number of candidate materials to obtain input parameters for the CMA code and to screen the materials by comparing thermal degradation characteristics. Samples have been tested in furnaces and in a wall panel simulation fixture. Data are still being collected and analyzed but suitable materials appear to be obtainable. The requirements for the foam are foam-in-place fabrication, minimum density, low thermal diffusivity, high char yield, fine grain char, and dimensional stability. There are also additional considerations related to impact and puncture performance that must be considered. Preliminary analyses indicate that foam materials of the type being tested will limit the interior surface temperature rise in the 1/2 hour regulatory fire to about 40°F if the foam thickness is 4 inches (10.2 cm) or 0°F if 8 inches (20.3 cm). As indicated previously, the TRUPACT wall thickness exceeds 8 inches (20.3 cm).

The structural response of the TRUPACT under a 1.7 atm. overpressure was evaluated using one of the more sophisticated ADINA finite element models analyzed in the impact situations but modified to allow beam type response of the frame members. The results of the calculation indicated that localized yielding may occur in a few frame members but the overall deformations sustained would not compromise the package's integrity.

*User's Manual: Aerotherm Charring Material Thermal Response and Ablation Program (Version 3), Vol. 1, Report No. UM-70-14, Aerotherm Corporation, Mountain View, CA, 1970.

At this time, the TRUPACT response to a 0.5 atm. underpressure has not been determined but it is believed this loading is not as severe as the overpressure environment.

TRUPACT Tests Conducted

An extensive test program has been pursued during the TRUPACT development with the goals of validating the concept, obtaining data for comparison with numerical analyses of the package, and providing visual evidence for public information purposes. Many impact and puncture tests have been performed with quarter scale models of the TRUPACT which include the predominant structural features of the package. A series of drop tests were conducted with individual and arrays of eighth-, quarter-, and full-scale Type A waste containers to determine their response to impacts and to aid in developing parameters for modeling cargo loadings in the numerical analyses. Numerous puncture tests with circular plates were also performed to enable correlations to be made between test results and theoretical predictions.

The quarter scale TRUPACT models had linear dimensions one fourth those of the corresponding full-scale values and weighed, when loaded with scaled waste containers, about one-sixty-fourth the weight of a full-scale fully loaded package. Each model was subjected to one or more 9 m drop-impact tests in different orientations including end-on, side-on, center of mass over corner, end-edge, side-edge, corner slapdown, and end-edge slapdown. Although the models sustained appreciable exterior damage, particularly those subjected to three or more impacts, virtually no distortion of the package interiors was evident. This result suggests that the TRUPACT exterior structure provides sufficient protection for the inner structure in impact type situations. In addition, excellent comparisons between theoretical predictions and experimental results were obtained. For instance, one end-impact test of a model resulted in 1.3 in (3.2 cm) crush-up of the package end which corresponds to 5.0 in (12.8 cm) full scale. The numerical prediction for this same impact situation was 5.2 in (13.2 cm) full scale.

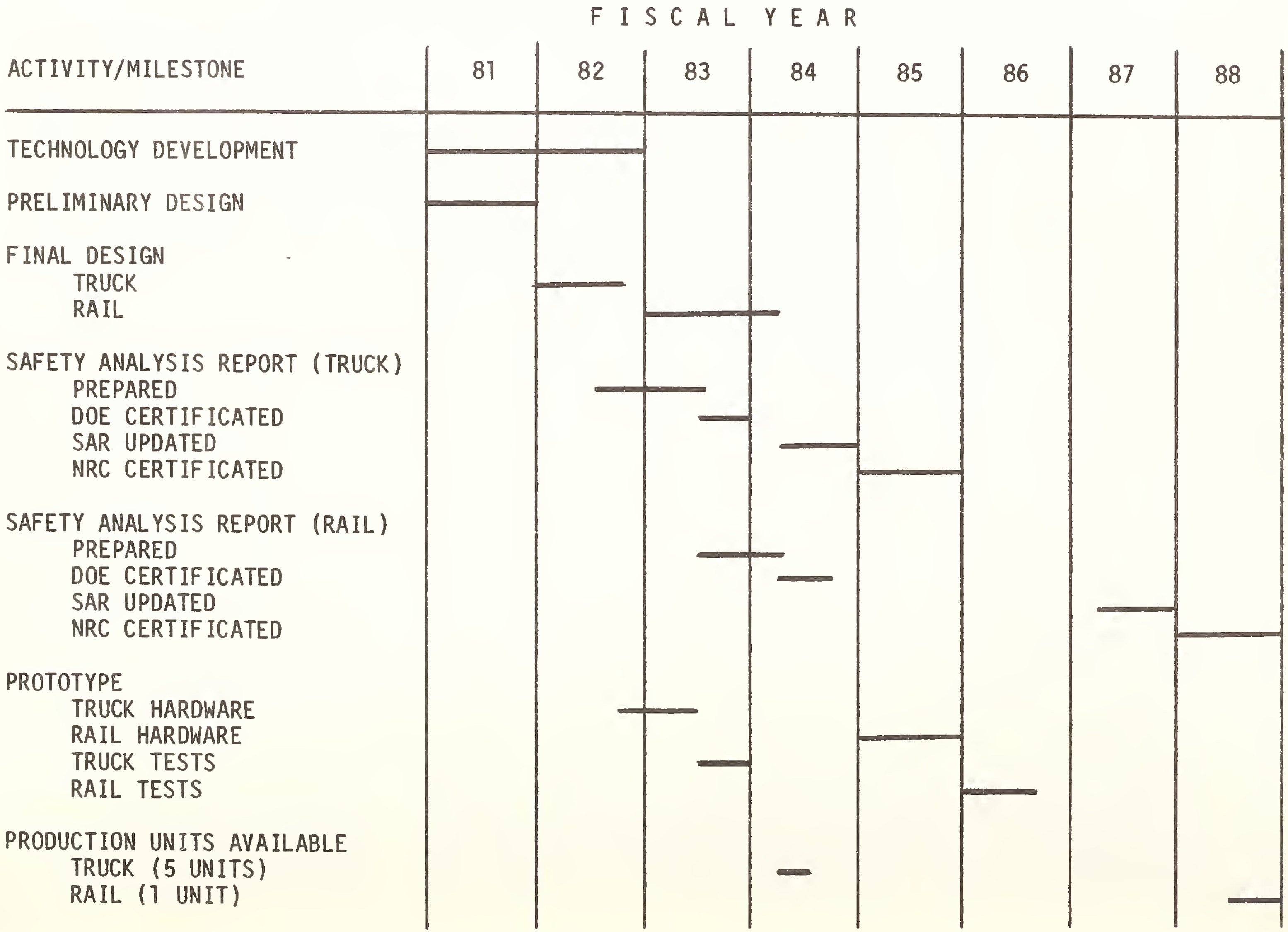
The first phase of the puncture test series included dynamically punch loaded clamped edge, 9 in (23 cm) and 18 in (46 cm) diameter, plates both with and without foam backing. These tests verified that scaling laws apply to the puncture phenomena. The approximate theoretical treatment of the puncture event was also found to predict the results of the punch tests reasonably well. Based on these tests and associated analyses, it was possible to estimate the puncture plate thickness required in the TRUPACT to prevent penetration. Puncture plates were included in two of the TRUPACT models and subsequent drop-puncture tests with these models demonstrated that a satisfactory puncture resistant design was obtainable with about 0.5 in (1.3 cm) thick stainless steel plates.

Thermal tests have been conducted on a number of commercially available rigid polyurethane foam materials. Screening tests were performed to measure response of a foam cube exposed to a uniform temperature for a specified time interval. Test objectives were to identify degradation characteristics (especially the tendency to melt), identify physical characteristics of the charred foam, and establish performance rankings of candidate foams (char yield and integrity). In these tests eleven candidates were tested and three were recommended for wall-fire testing. The objective of the wall-fire tests is to simulate a pool fire environment using radiant lamps to heat candidate foams in a stainless steel enclosure and to develop one-dimensional heat and mass transfer data. These tests are currently being conducted and will be reported on at a later date.

Program Schedule

The program schedule is currently being revised to emphasize final design of the truck configuration and provide a test prototype in FY 83 and five production units in late FY 83 and early FY 84. Final design of the rail configuration will begin after the truck version is completed and will be finished in FY 84 with prototype hardware available in FY 86 and production units available about FY 88 or 89. A program schedule for TRUPACT is shown in Table 3.

Table 3. TRUPACT Program Schedule



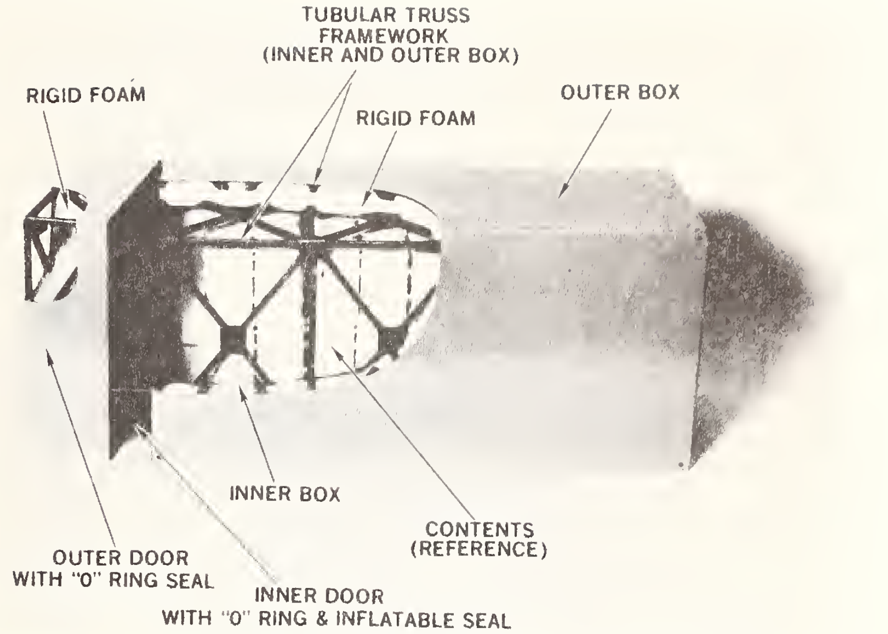


Figure 1. Photograph of a TRUAPCT Model, Partial Cutaway Showing Internal Structure

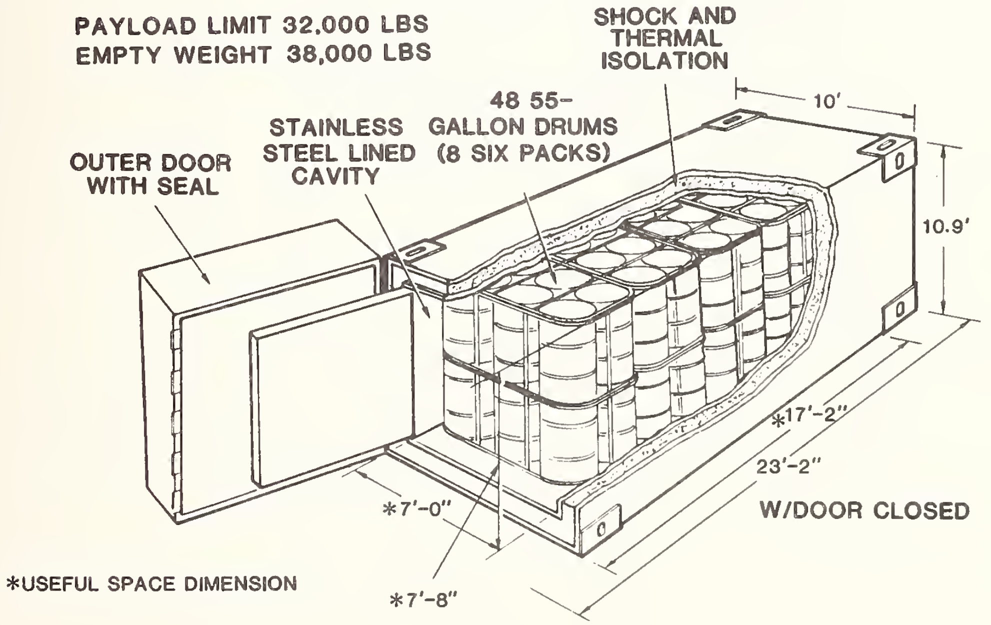


Figure 2. TRUPACT Containers (10 Ft Wide) Showing Railroad Configuration With Six Packs

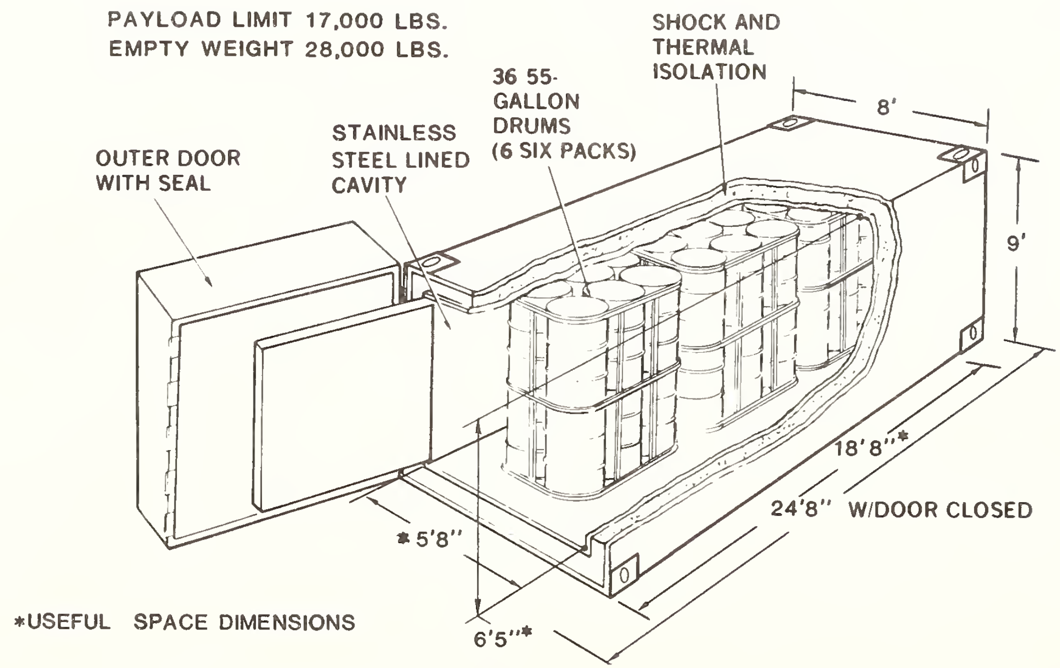


Figure 3. TRUPACT Container Showing Truck Configuration with Six Packs

SESSION VI

HIGHWAYS

CHAIRMAN: J. C. SHANG
GENERAL AMERICAN RESEARCH DIVISION

PLANNING TO PREVENT FAILURE IN THE TRANSPORTATION OF HAZARDOUS MATERIALS

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Abstract: The prevention and mitigation of consequences in the transportation of hazardous materials are discussed. Efforts in the mitigation, or "emergency response" aspects, are overemphasized at the expense of preventive efforts.
Planning for the prevention of incidents requires two areas of effort: knowledge of past failures, and a preventive outlookin design, construction, operation, and maintenance.
There is a general lack of data base for preventive efforts. Incident informationis usually used solely for training emergency response personnel.
An example is discussed, that of the George Washington Bridge incident of August 7, 1980. Why a valve failed. Why it took a $4 plumber's test plug--by purest chance and coincidence--to solve such a potentially catastrophic situation. An example of how little questioning there was of valves or other components of vehicles or guideways, in all the transport modes.
The problems of moving hazardous materials in bulk form, a major problem on the railroads. Derailment, turnover, puncture as a common sequence. The application of limited solutions: shields as crutches. Ignoring the systems approach.
The need for a system-wide approach ("Transportation System Safety") in planning, based on feedback from construction, testing, operations (incidents) and maintenance. The need for original, specific designs for each special purpose.

Key Words: Hazardous materials; transportation of hazardous materials; transportation system safety.

In safety engineering, there are only two approaches to improving safety: that of attempting to prevent incidents from happening; and that of mitigating, or reducing the severity, of the accidents which surely will happen.
In the activity of transporting hazardous materials, more effort (and money) goes into the mitigation aspect, in this case 'emergency-response' activities, than into prevention. At the same time, however, because of the unsystematic, scattergun approach taken to emergency-response training, overall efforts even here are inadequate.
Prevention involves planning, and planning ahead requires knowledge of what has happened in the past. In this respect

there is little knowledge, little 'data base' available concerning how accidents have happened in the past. What informaterion has been collected on past 'failures' of the system, past incidents, is used primarily for training emergency response personnel.

This is a laudable use of data, but greater efforts in prevention should reduce the need for emergency response efforts.

In an incident on the George Washington Bridge, on August 7, 1980, a vehicle carrying liquefied petroleum gas sprang a leak. The leak was eventually stopped and the incident caused no injuries or damage.

However, in considering the 'system' of transporting such hazardous materials on public highways, a number of questions can be raised:

--Why has no study been made of how and why the GWB incident occurred, and of how it can be prevented in the future?

--Why did a specific valve fail?

--Why did it require the extraordinary coincidence of a policeman, who moonlights as a plumber's helper, recognizing that a $4 test plug (which he promptly purchased) would stop the leak?

The answer to all these questions is: because of lack of planning. The National Transportation Safety Board, which investigates all major accidents in all transportation modes, would not investigate an accident which did not result in death or major injury. Therefore the lessons to be learned from a potential disaster go unlearned. The National Highway Traffic Safety Administration, which has had responsibility for improving the design of vehicles, is undergoing a loss of many of its powers, as weak as they were. The solution to the spcific problem on the Bridge was left up to Coincidence, or Providence, or whatever.

In considering how to prevent failures in the transportation of hazardous materials, engineers in recent years have fallen back on concepts of 'system safety', which is as much a planning tool as it is an engineering tool. The interest in system safety concepts reached a peak with the work done at the Polytechnic Institute of New York from 1976 to 1980 on creating a methodology called 'transportation system safety', with applicability to all transportation modes.

The first task in applying this approach is to define the system being dealt with, and all its parts, subsystems, and components. In the case of defining the transportation of hazardous materials by highway, the basic system is composed of man as driver; machine, the vehicle; and the environment, composed of the road, the roadside, signs and markings, other traffic, rules and regulations, weather and light conditions, etc.

The difficulties with a system as complex as that of moving hazardous materials by highway immediately surface: there are few standardized vehicles; every driver has different physical capabilities, training competence, and psychology; and the 'environment' includes a host of variables, each of which has an impact on safety.

And yet, by making the identification process as detailed as possible, we can see immediately where there is a lack of needed data, or knowledge, or standards of design or operation; where there is duplication or lack of training. For instance, fire fighters were never taught that you don't put out a fire that involves a tank vehicle carrying LPG. The common name for this condition is "blevey", for Boiling Liquid Expanding Vapor Explosion. Until a number of firefighters were killed, the profession did not learn that if the tank is venting, you don't put it out, you let it burn. If it is not venting, you use a fog spray. You don't approach the tank from either end, because the ends are its weakest points.

Is it possible that engineers and chemists didn't know this 30 years ago? Obviously they did. No one asked them, so they didn't tell the right people.

On the railroads, most incidents involving hazardous materials relate to derailments. Derailments relate directly to poor maintenance of both track and vehicles. Maintenance problems represent poor planning. Maintenance records are always used as an example of a source for planning new units, new components, new subsystems, and new systems. But did the designers of the BART system consult the operators of the New York City Subway System? The example of constant feedback, of learning from the errors and failures of the past, is often used but rarely followed.

On the highways problems of highway maintenance are compounded by problems of vehicle design and operation and by the inadequacy of training of drivers and materials handlers.

Our dedication to 'reacting' to emergencies means that we do little planning to prevent the emergency situation. And then we don't prepare our emergency crews sufficiently anyway.

The Federal Government publishes lists of hazardous materials but the manufacturers must tell the constructors of the special-purposes vehicles what their requirements are, and they must also have a hand in driver training, the training of the people who load and unload materials onto and from trucks, and the emergency response personnel who must be trained in the proper handling of thousands of different chemical compounds which can escape and burn or explode.

Some of this is done, of course, by the largest of chemical companies, which own their own fleets of trucks and train their own drivers. But most trucks are owned by owner-drivers, and there were more than 27,000 hazardous-materials incidents involving trucks last year in the U.S.

The systems approach applied to this problem means true planning--thinking ahead to what <u>might</u> happen and what <u>could</u> happen; of what the probabilities <u>are</u> of failure of a component, a vehicle, or a human being, and eliminating the hazard in the design process. The procedure of looking for accident cause, and patching up a tank or a valve or changing a highway route, under the assumption that another accident of the same type will be avoided, is completely incorrect. No two accidents or incidents ever occur in exactly the same way.

Systematic thinking--and planning--is the only solution to preventing failures in the complex activity of transporting hazardous materials on the roads.

CARGO RIDE EVALUATION ON A ROAD SIMULATOR

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Abstract: A combination vehicle such as a tractor-semi-trailer is a dynamic system which responds to road surface irregularities. When the natural frequencies of the system coincide with the road frequencies, near resonant conditions exist which might result in cargo damage or driver fatigue or both. A technique is presented in this paper where a road simulator is used to compare the effects of road surface, speed, and suspension on ride and trailer floor accelerations.

Fruehauf Road Simulator is a tire actuated electro-hydraulic system where real roads can be simulated under controlled conditions. The spindle accelerations are measured on the road. The transfer function between the tires and spindles are measured on the road simulator. Then, by a compensation technique, the effective road profile can be calculated. When the trailer is excited with this effective road profile, the response of it will be similar to that on the road.

Once the effective road profile of a representative road surface is obtained, it could be used to study the effects of speed, loading conditions, suspension design and wheel base on ride and cargo accelerations.

At present, there is no universal specification for maximum acceleration on cargo. The reason for this may be because there are infinite variations in loading locations, speed, road surface and cargo mounts. As an indication of the dynamic environment to which the cargo will be subjected, the PSD of the vertical accelerations of the floor at various locations are calculated from acceleration response. Again, the effects of modification, including load location, can be easily studied.

Thus, the road simulator could be effectively used to compare various design changes in terms of cargo accelerations under controlled conditions in the laboratory.

Key words: Effective road; frequency domain; frequency response function (FRF); power spectral density (PSD); spatial PSD; system resonances; temporal PSD; time domain.

One method for studying the response of a heavy duty vehicle subjected to dynamic road inputs is reproducing those inputs in the laboratory. The Fruehauf Corporation Road Simulator is composed of four tire actuated electro-hydraulic actuators and a fifth wheel fixture (composed of two electrohydraulic actuators) which takes the place of a tractor (see Figure 1). The methodology presented here is aimed towards heavy duty trailers, but is applicable to most size vehicles.

A technique has been developed whereby an "effective" road is input to the vehicle through vertical actuators such that the dynamic response will be similar to that obtained on the road (1)*. Once this effective road has been developed, a number of options are available for using it which will be discussed later. First, some insight into effective road development is in order.

Several steps are required to develop an effective road as shown in Figure 2. The vehicle is driven over a selected route at constant speed and the spindle and fifth wheel accelerations are recorded on analog tape, which represents the desired response to be reproduced on the road simulator. Accelerations at floor level or other locations may be recorded as well, which provide another means of verifying dynamic response in the laboratory. The field data is then digitized using analog to digital converters and a digital computer. Once digitized, the data is then edited to remove erroneous data such as tape drop outs or large transients.

The next step is to take the same vehicle used to record the data and determine its Frequency Response Function (FRF) on the road simulator. Simply stated, an FRF is a measure of the gain and phase characteristics between an input (e.g., vertical tire actuator displacement) and output (e.g., acceleration response at spindle) as a complex function of frequency (see Figure 3). It should be noted that an FRF is a description of a linear system. Since vehicle components such as leaf springs and tires are non-linear, the inputs used to excite the vehicle for FRF determination should be composed of roughly the same frequencies and amplitudes as the measured road inputs. Hence, the input used for FRF determination will be random in nature, but statistically similar to the road profile being developed.

For a multi-input, multi-output system there would be one FRF for every input-output combination, which is best handled in matrix form as shown in Figure 4. The $Y_i(f)$'s in Figure 4 represent spindle response, the $X_i(f)$'s rep-

*Refers to reference at end of paper

resent actuator drive signals, and each $H_{ij}(f)$ represents the FRF between the respective signals.

In matrix notation, this may be written as

$$Y(f) = H(f)X(f)$$

Through a series of matrix manipulations, X(f) can be solved for, thus

$$X(f) = H(f)^{-1}Y_D(f)$$

where $Y_D(f)$ is the Fourier transform of the desired response and X(f) is being sought (1).

This calculated drive signal, X(f), based on the system transfer function and desired response, can then be used to excite the vehicle on the road simulator and the actual spindle responses can be recorded simultaneously. If the actual response of the vehicle is not suitably close to the desired response, the drive signal can be iteratively modified until the actual and desired responses are in agreement.

In the effective road development technique just described, the test vehicle was used for obtaining the road profile. Once the road profile is derived and stored on digital tape in the form of vertical actuator stroke inputs, the test vehicle is no longer required to reproduce the effective road. Any vehicle similar to the test vehicle may be excited by the effective road and its response will be similar to the response obtained if it were driven over the same road as the test vehicle.

A convenient way for summarizing random time history response is through the use of power spectral density (PSD). A PSD statistically resolves a time history into its frequency components, with the ordinate being squared amplitude per temporal frequency, and the abcissa temporal frequency. [Temporal frequency is cycles per second or hertz (Hz). Unless stated otherwise, the use of the term PSD will indicate temporal PSD throughout this paper.] A PSD is actually the mean square value of all the amplitudes occurring within a narrow frequency band, or bandwidth, divided by the bandwidth. Hence, by indexing the bandwidth throughout the entire frequency spectrum of interest, a continuous curve is obtained.

Referring now to Figure 5, the solid curve is a PSD of vertical floor acceleration at the front of a trailer at floor level being driven down a fairly rough asphalt road at 40 mph. Note that the amplitude is in g's squared per hertz. The dashed curve is a PSD for the same trailer being excited by the corresponding effective road on the road simulator. The PSD's are nearly identical up to approximately 35 Hz. The deviation is due to the physical limitations of the actuators. However, the major structural modes occur below 20 Hz so that this becomes inconsequential.

Figures 6a, b, and c are time history plots for three different one-second time intervals from which the PSD's of Figure 5 were obtained. These are presented to illustrate the difference between time domain and frequency domain data. Additionally, these three plots indicate the accuracy with which dynamic responses can be reproduced in the laboratory. (It should be noted that the plots presented in this paper are in no way conclusive; rather, they are presented as illustrations of possible effects due to different vehicles, loads, road surfaces, etc., on cargo ride.)

It can readily be seen that the effects on cargo ride due to different vehicles, as well as design changes and load variations within the same vehicle, can be compared by exciting with the same effective road and recording the corresponding responses at floor level.

As an illustration of the preceeding statement, the following examples are presented. A 45 foot van trailer with a concentrated 50,000-pound load at the center of the trailer was excited on the road simulator. The input was the effective road of the fairly rough asphalt road at 40 mph as previoulsy mentioned. Vertical floor accelerations were recorded and the PSD's calculated. The load was then uniformly distributed and the test repeated. The two curves in Figure 7 are PSD plots with the solid curve representing the first load condition and the dashed curve representing the second load condition. The several major peaks occurring before 7.5 Hz represent the trailer free body modes, i.e., roll, bounce, pitch, etc. The highest peak which occurs at 5.2 Hz for load condition one is lowered to 4.6 Hz for load condition two, and the amplitude is reduced as well. However, the peak at 2.8 Hz remains at the same frequency for both load conditions, and the amplitude becomes larger for the second load condition. Hence, depending on cargo critical frequencies, the second load condition may or may not be desirable.

Since a PSD is a squared function, it is often convenient to obtain a linear plot so that a linear comparison may be made. This can be accomplished by multiplying the PSD amplitude by the bandwidth and taking the square root. The result is an rms plot as shown in Figure 7b.

A second example is presented in Figure 8 which compares the difference in vertical floor acceleration at the rear of the trailer between two different trailer suspensions (same trailer in both cases). The solid curve represents the response obtained with a conventional leaf spring suspension, and the dashed curve is the same for an air suspension. The curves indicate the reduction in acceleration levels obtained with an air suspension.

Up to this point, the effective road discussed has been a time history reproduction of a particular stretch of road at one speed. Chronological events including transients such as chuck holes, railroad tracks, chatter bumps, etc., can be reproduced. However, if general road surface characteristics alone are of interest, it is advantageous to develop effective roads which statistically represent particular classes of roads. This is accomplished by using a frequency domain effective road development procedure, which is similar to the time domain effective road development technique previously discussed. The resultant "frequency domain" effective road used to drive the actuators is still a time history, but it does not reproduce the road inputs in chronological order. Rather, the order is random, with amplitude and frequency content similar to the road being developed. Hence, one frequency domain effective road could represent smooth concrete roads, another represent rough asphalt roads, etc. In this manner, the effects of road surface roughness can easily be studied.

Road surfaces are most commonly categorized by spatial PSD's of profilometer recordings. A spatial PSD is given in terms of spatial frequency and spatial spectral amplitude (e.g., cycles per foot and feet squared per cycle per foot) as opposed to temporal PSD which is in terms of temporal frequency and temporal spectral amplitude (e.g., cycles per second and feet squared per cycle per second). For a given constant velocity, a spatial PSD may be converted to a temporal PSD by multiplying the spatial frequency by the vehicle velocity, and dividing the spatial spectral amplitude by the velocity, and vice versa.

Shown in Figure 9 are spatial PSD's of a rough asphalt road (solid curve) and a smooth concrete road (dashed curve). Note that the trends are similar (i.e., high amplitude at low frequency and low amplitude at high frequency) despite the individual differences. These plots were obtained by calculating the temporal PSD's from effective road actuator stroke inputs and then converting to spatial PSD's. Due to nonlinearities (which are compensated for by the effective road development technique), these will differ slightly from spatial PSD's of profilometer recordings of the actual roads, but the mean square values will be approximately the same. (The mean square value is equivalent to the area under the PSD.) As a demonstration of cargo ride effects due to road surface roughness variation, both of these frequency domain effective road profiles were used to excite the same trailer under the same loading condition. Corresponding trailer vertical floor acceleration PSD's for both of these inputs are shown in Figure 10. The curves indicate that cargo ride is less severe on smooth roads at most frequencies.

However, this is not always the case. Depending on load distribution and speed, the vehicle pitch mode can be excited due to tire resonances on a smooth road. A rough road tends to wash out the tire resonances thus eliminating the severe pitching.

Two additional advantages of a frequency domain effective road are that the wheelbase of the test vehicle and the speed of the effective road may be changed.

To test two vehicles of varying wheelbase with the same effective road, the phase relationships between actuator inputs can be mathematically adjusted to reflect the corresponding change in wheelbase. To do this, the PSD of the frequency domain effective road drive signal is calculated, then it is "filtered" by the appropriate time delay matrix. The effective road drive signal is then regenerated based on the shape of the filtered effective road PSD. (Correction for wheelbase is also possible with a time domain effective road.) The effects on cargo ride due to a 5% change in wheelbase are dramatized in Figure 11.

As with wheelbase correction, to change the speed of a frequency domain effective road, the temporal PSD is first calculated. Then, based on the original speed of the effective road, the temporal PSD is converted to a spatial PSD. The spatial PSD is then converted back to a temporal PSD based on the desired new speed, and the effective road drive signal regenerated. Due to the mathematics involved, decreasing the effective speed results in loss of high frequency data, addition of low frequency data, and the effective frequency resolution increases. On the other hand, increasing the speed results in just the opposite effects, i.e., the addition of high frequency data, loss of low frequency data, and a decrease in the effective frequency resolution.

Shown in Figure 12 are PSD's of the same effective road surface input before and after a 5 mph speed increase. From the plots it can be seen that increasing the speed increases the frequency of a given peak, while at the same time the amplitude is decreased. The advantage of being able to change the speed of an effective road is that critical vibration speeds may be determined. In effect, a speed sweep test may be conducted by successively changing the speed of the same effective road, exciting the test vehicle, recording the response, and thereby, determine the speeds at which the system resonances are at a maximum. Figure 13 graphically illustrates the results of such a test.

In summary, the value of a road simulator to define dynamic cargo environment has been outlined. Except for measuring accelerations on the road, all testing is conducted in a stationary environment where vehicle dynamic response is easily observed and measured. The effects of cargo load variation, test vehicle component variation, and different test vehicles of the same wheelbase may be studied by using a time domain or frequency domain effective road. To determine the effects of road surface roughness, wheelbase, or speed variations, a frequency domain effective road is preferred. For each variation tested, the vertical floor accelerations may be recorded and analyzed using PSD techniques.

A final step, which has not been discussed, would be to utilize the vertical floor accelerations, which define dynamic cargo environment, for component cargo testing. Since cargo configurations, as well as the impedance at the cargo-trailer floor interface are so variable, the cargo of interest could be excited on a shaker table using the vertical floor acceleration PSD's recorded on the road simulator to define the input. The dynamic response of the cargo itself could then be recorded and analyzed.

Reference

(1) B.W. Cryer, P.E. Nawrocki, and R.A. Lund, "A Road Simulation System for Heavy Duty Vehicles", paper 760361 presented at the SAE Automotive Engineering Congress, Detroit, Michigan, February 23-27, 1976

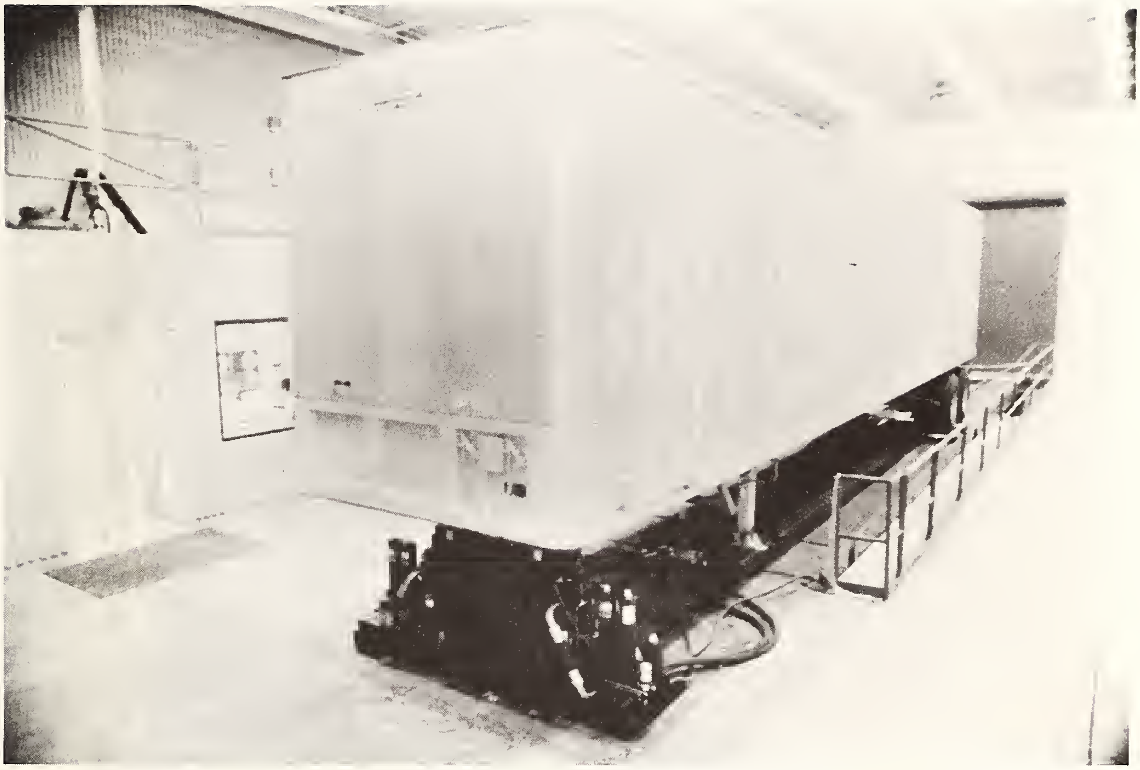

FIG. 1 - FRUEHAUF CORPORATION ROAD SIMULATOR

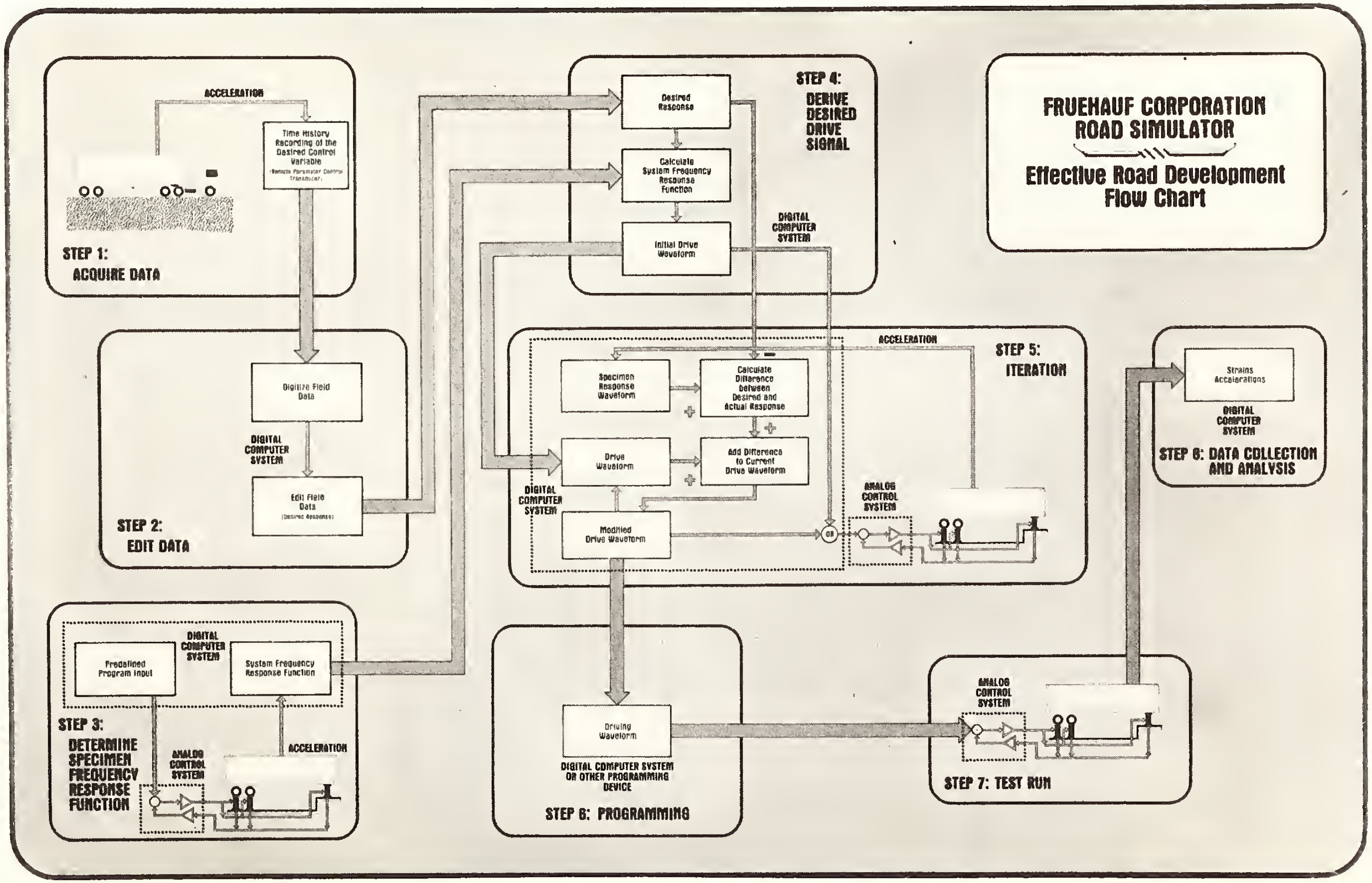

FIG. 2 - EFFECTIVE ROAD DEVELOPMENT FLOW CHART

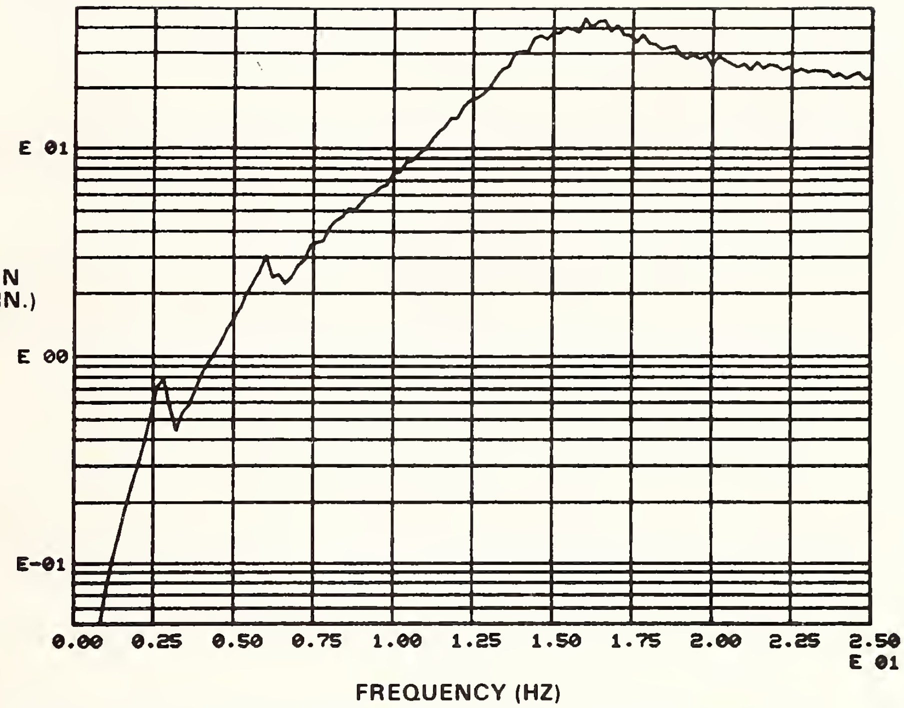

FIG. 3a - FRF GAIN BETWEEN TIRE INPUT DRIVE SIGNAL AND AXLE SPINDLE RESPONSE

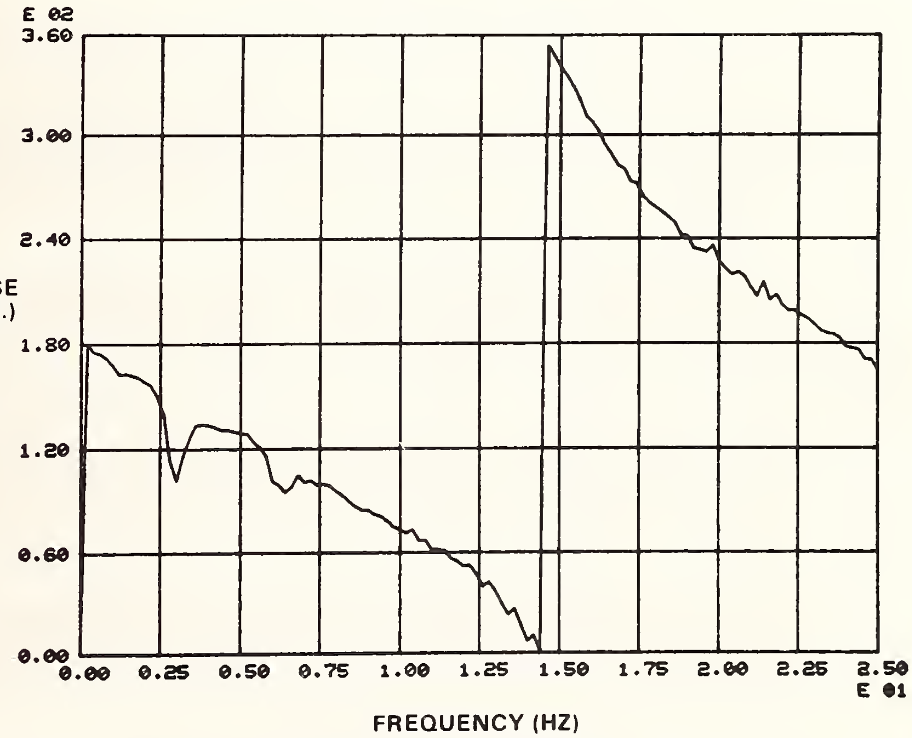

FIG. 3b - FRF PHASE BETWEEN TIRE INPUT DRIVE SIGNAL AND AXLE SPINDLE RESPONSE

$$
\begin{bmatrix} Y_1(f) \\ Y_2(f) \\ \cdot \\ \cdot \\ \cdot \\ Y_n(f) \end{bmatrix}
=
\begin{bmatrix} H_{11}(f)_1 H_{12}(f) & \cdot\ \cdot\ \cdot & H_{1n}(f) \\ H_{21}(f)_1 H_{22}(f) & \cdot\ \cdot\ \cdot & H_{2n}(f) \\ \cdot & & \\ \cdot & & \\ \cdot & & \\ H_{n1}(f)_1 H_{n2}(f) & \cdot\ \cdot\ \cdot & H_{nn}(f) \end{bmatrix}
\begin{bmatrix} X_1(f) \\ X_2(f) \\ \cdot \\ \cdot \\ \cdot \\ X_n(f) \end{bmatrix}
$$

FIG. 4 - FRF MATRIX EQUATION

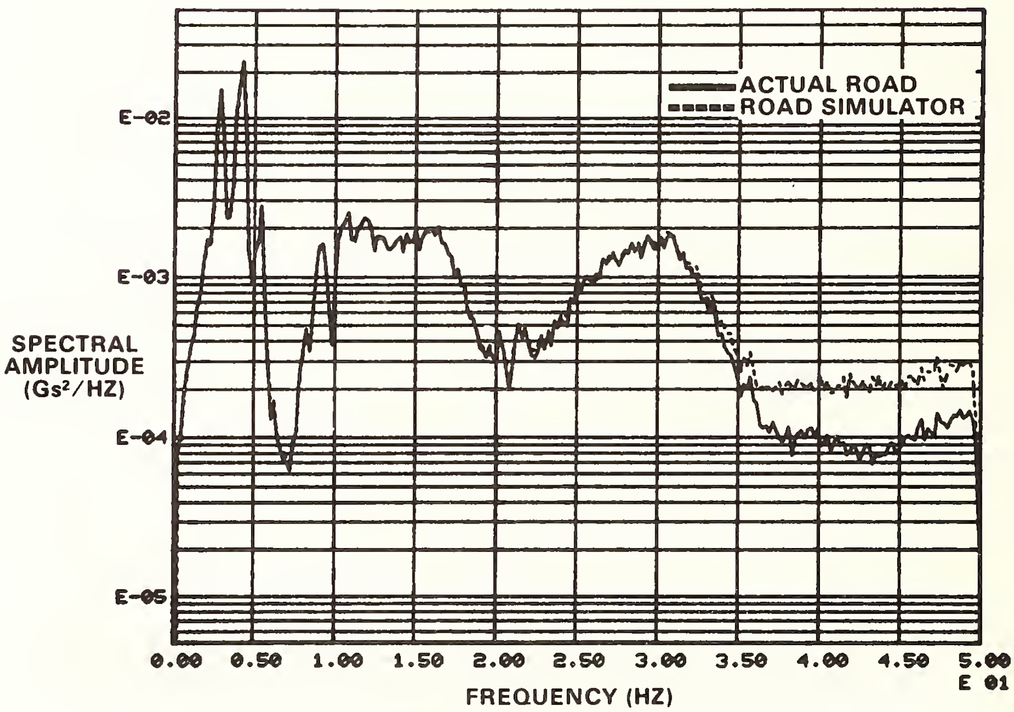

FIG. 5 - PSD's OF ACTUAL ROAD RESPONSE VS. ROAD SIMULATOR RESPONSE

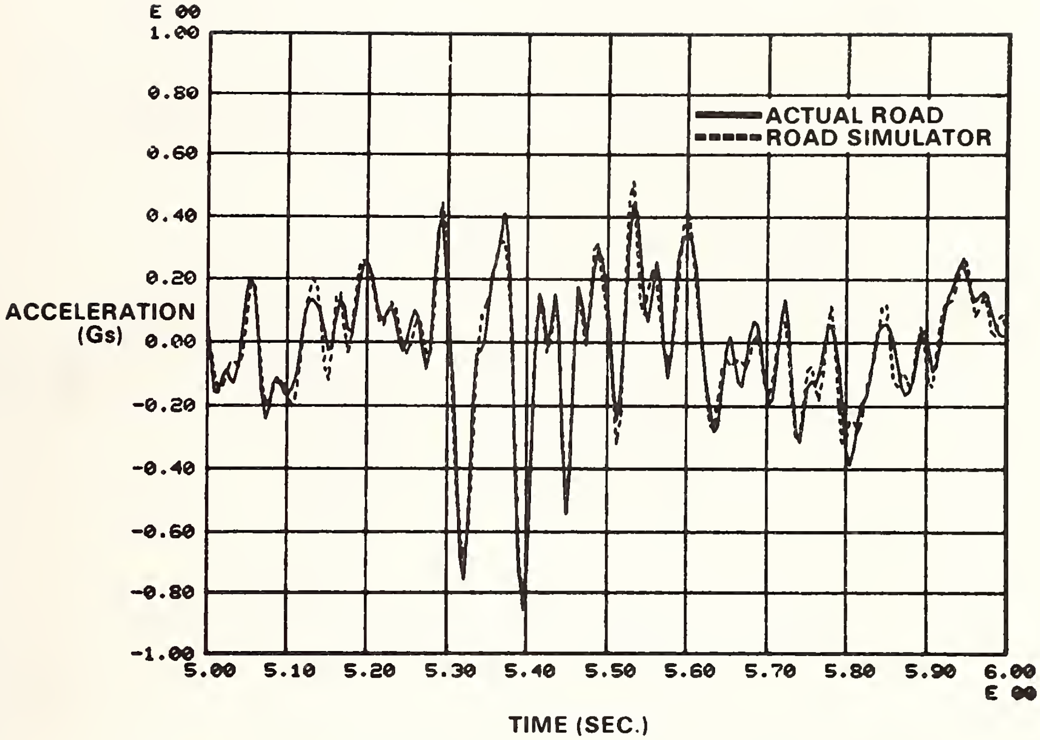

FIG. 6a - TIME HISTORY OF ACTUAL ROAD RESPONSE VS. ROAD SIMULATOR RESPONSE

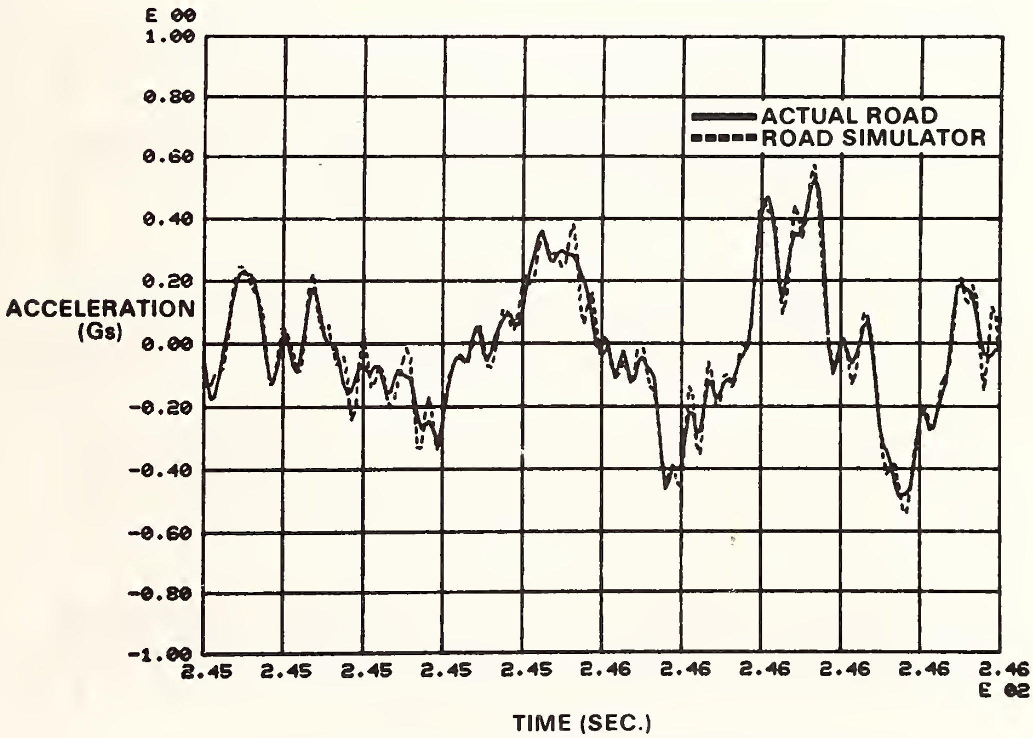

FIG. 6b - TIME HISTORY OF ACTUAL ROAD RESPONSE VS. ROAD SIMULATOR RESPONSE

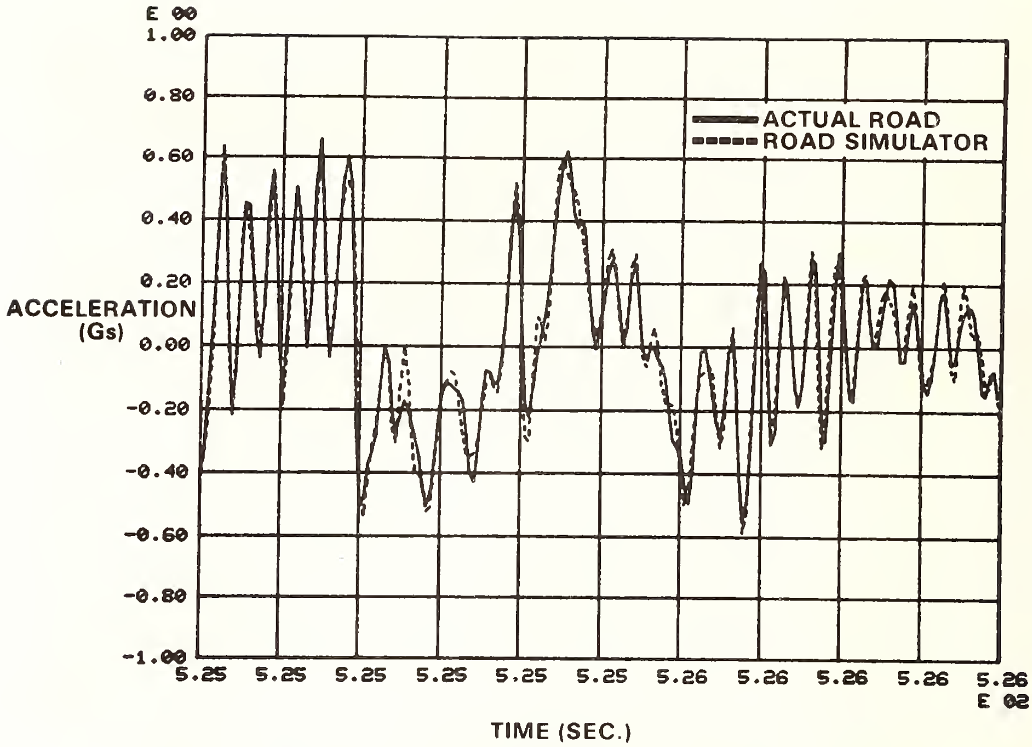

FIG. 6c - TIME HISTORY OF ACTUAL ROAD RESPONSE VS. ROAD SIMULATOR RESPONSE

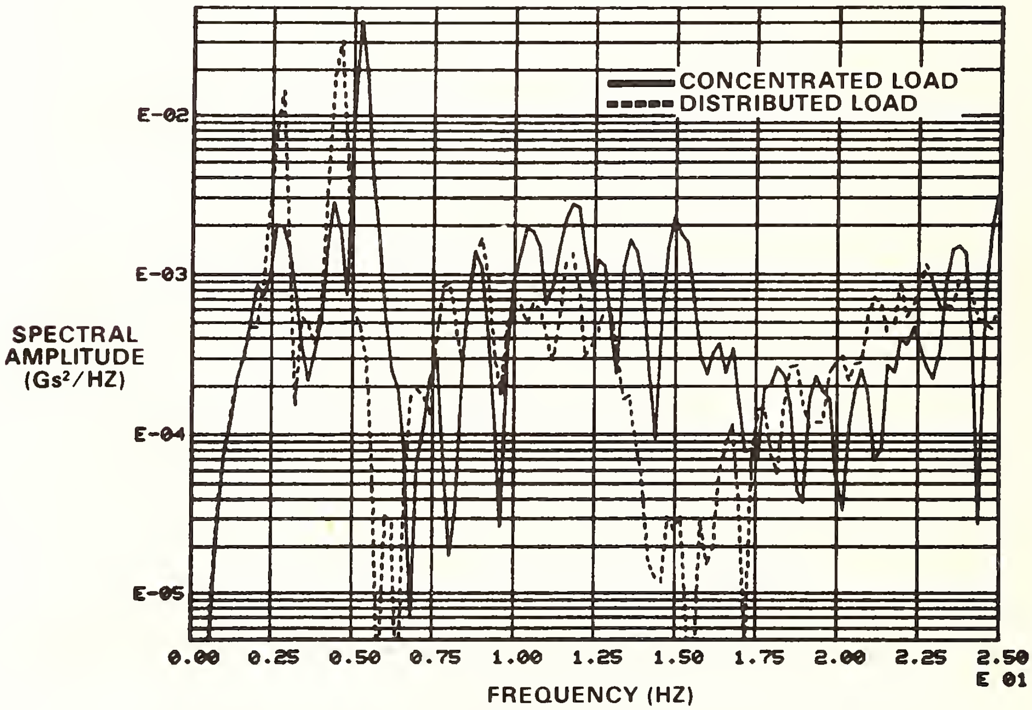

FIG. 7a - PSD OF VERTICAL FLOOR RESPONSE — LOAD VARIATION COMPARISON

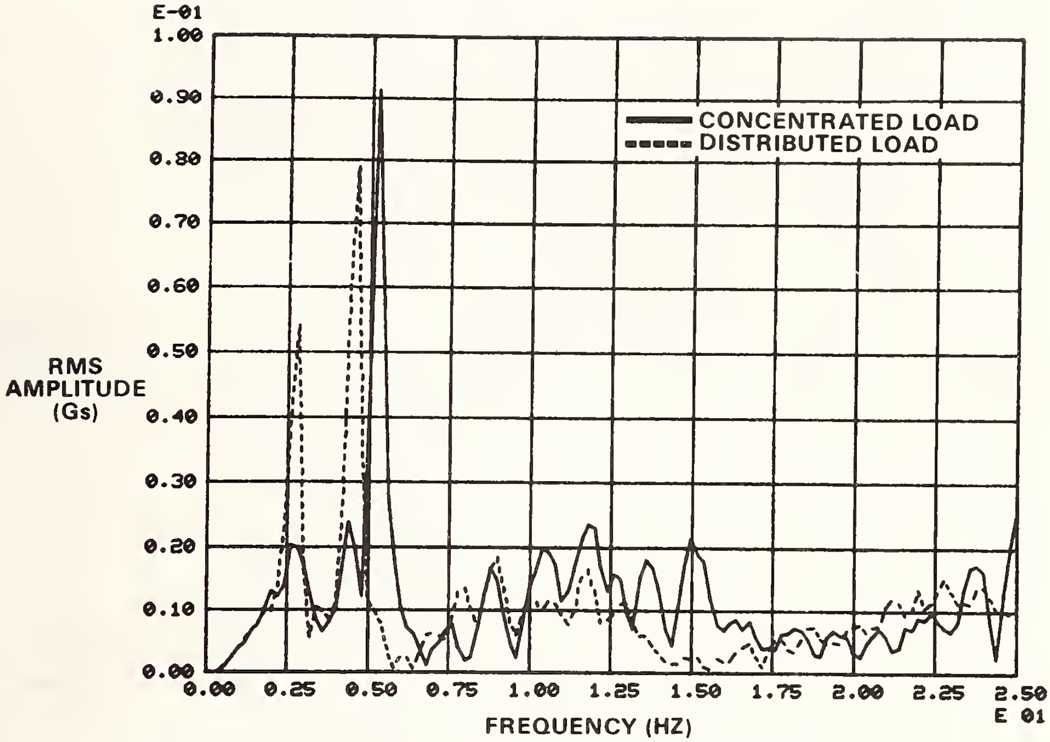

FIG. 7b - RMS OF VERTICAL FLOOR RESPONSE — LOAD VARIATION COMPARISON

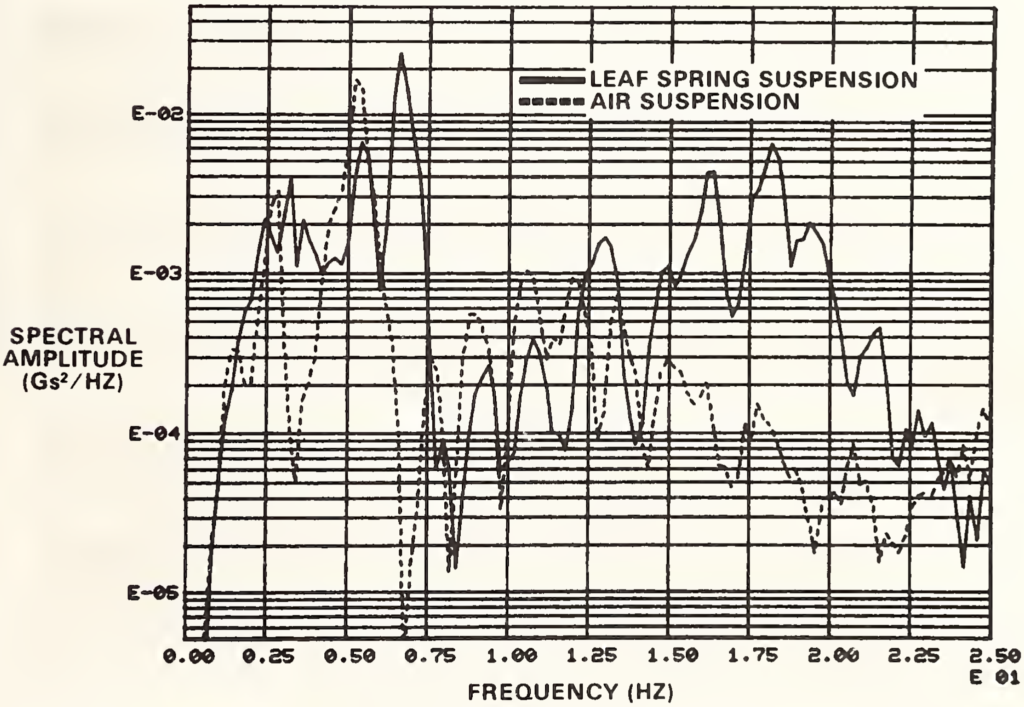

FIG. 8a - PSD OF VERTICAL FLOOR RESPONSE — TRAILER SUSPENSION COMPARISON

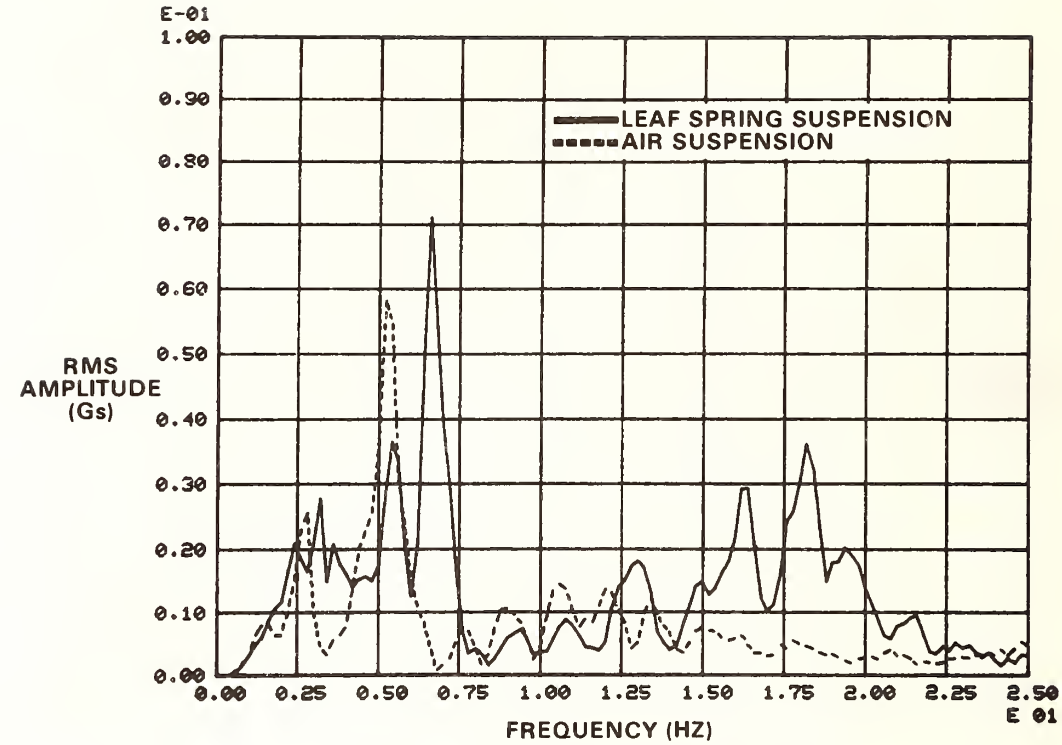

FIG. 8b - RMS OF VERTICAL FLOOR RESPONSE — TRAILER SUSPENSION COMPARISON

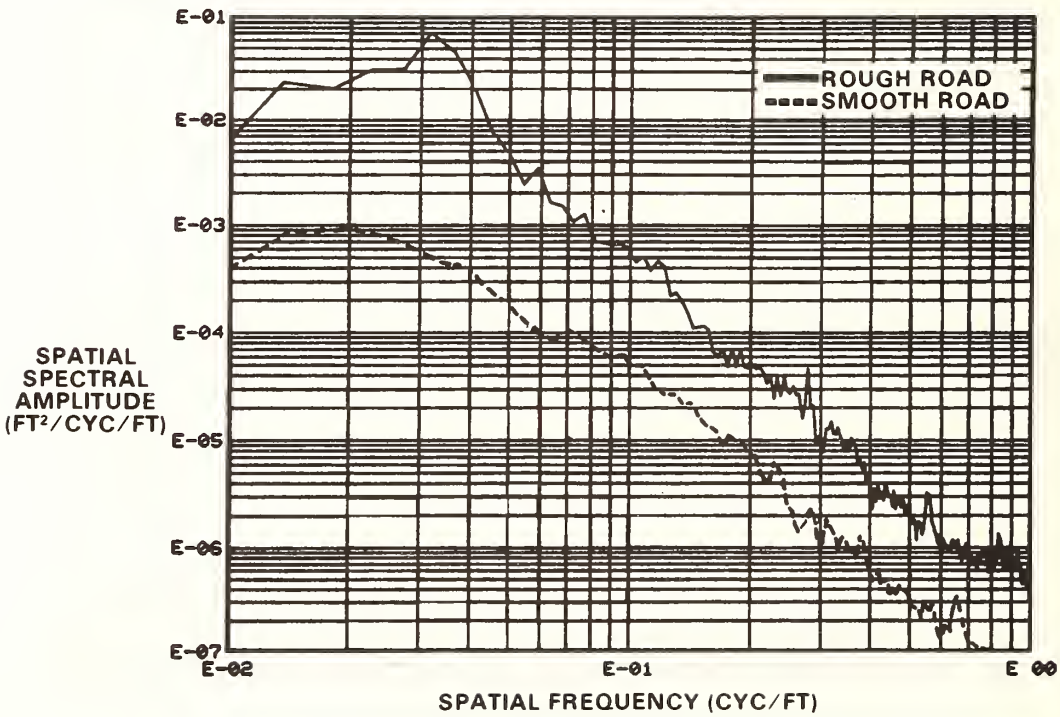

FIG. 9 - SPATIAL PSD's OF EFFECTIVE ROAD INPUTS

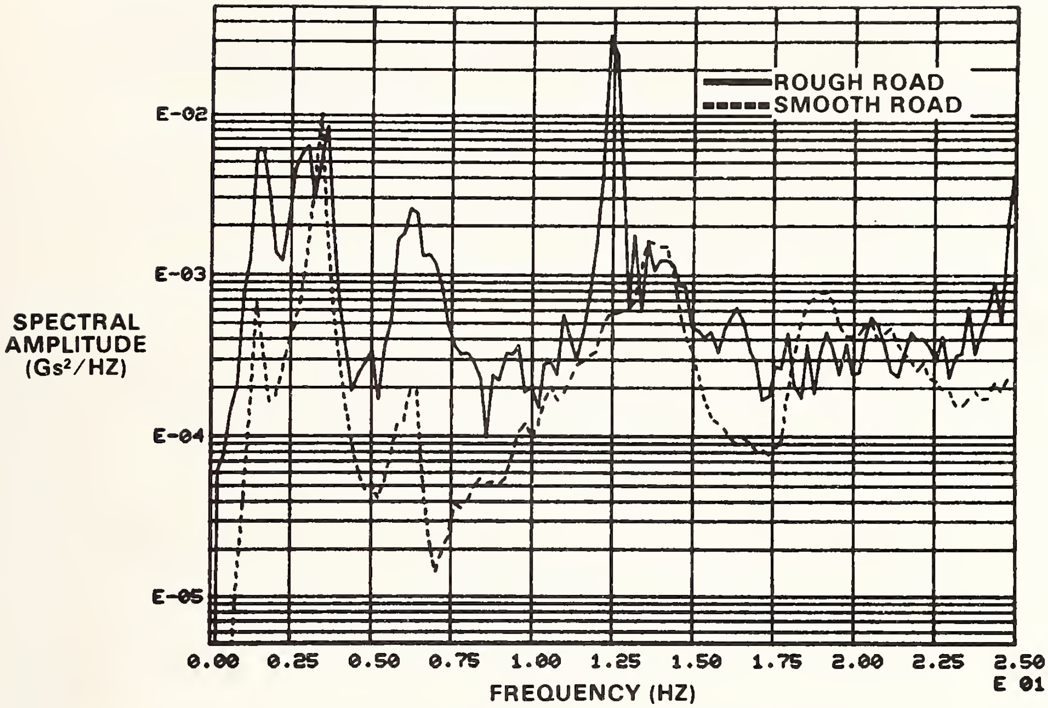

FIG. 10 - PSD OF VERTICAL FLOOR RESPONSE — ROAD SURFACE COMPARISON

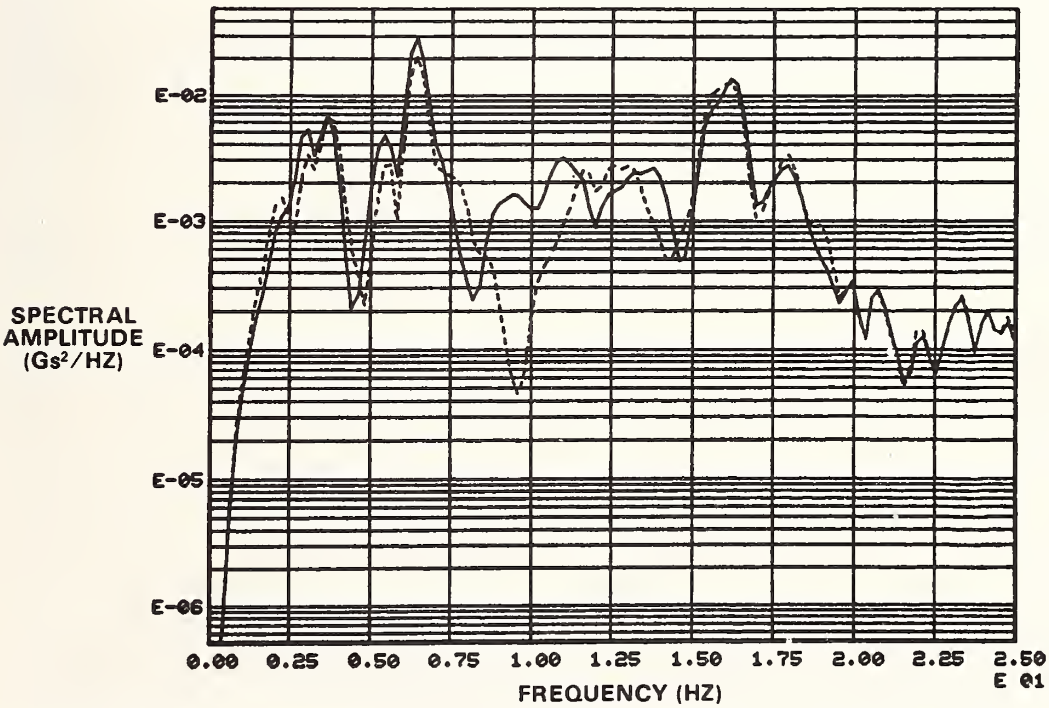

FIG. 11 - EQUAL INPUT AMPLITUDE INPUT PHASE DIFFERENCE (EQUIVALENT TO 5% CHANGE IN WHEELBASE)

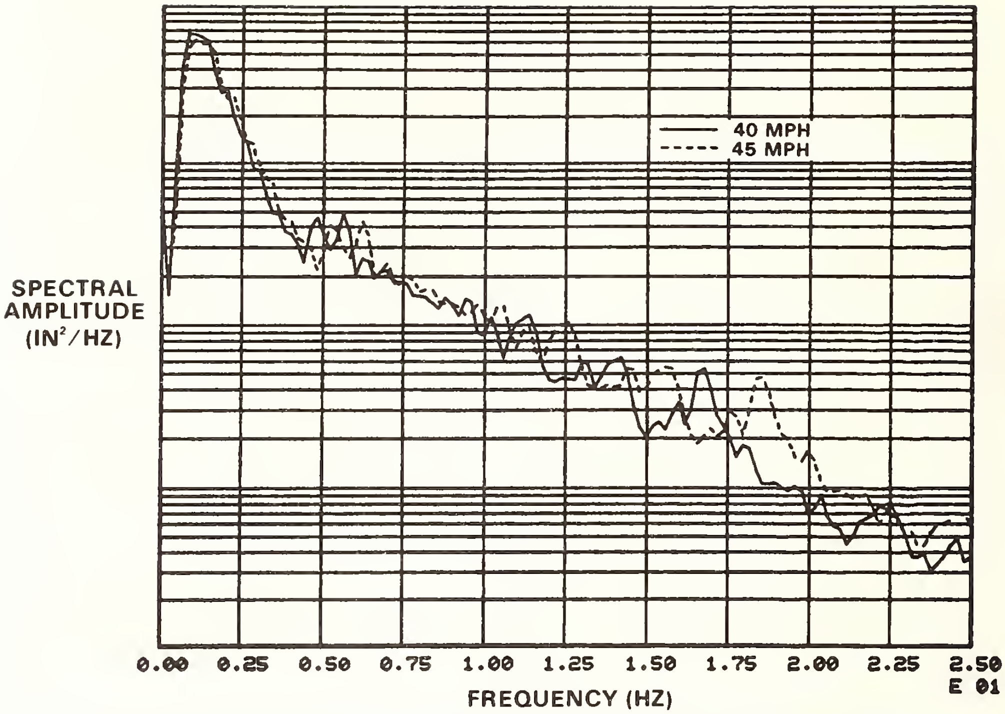

FIG. 12 - SPECTRAL DENSITY OF SINGLE ROAD SURFACE AT DIFFERENT VEHICLE SPEEDS

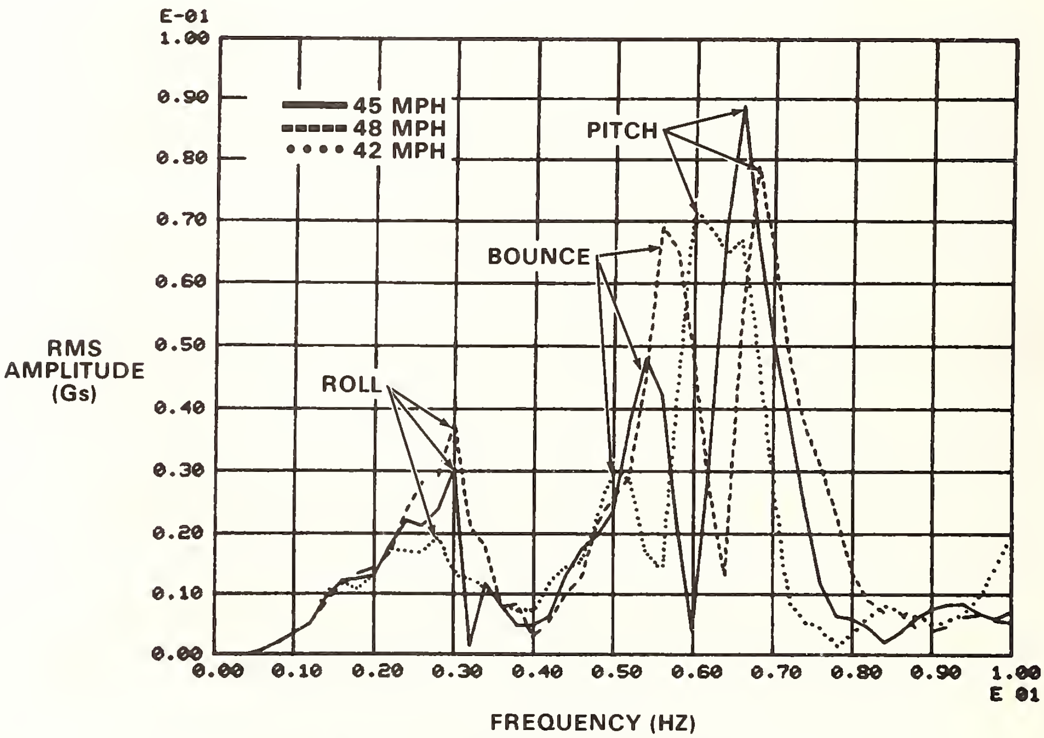

FIG. 13 - RMS OF VERTICAL FLOOR RESPONSE — SPEED VARIATION COMPARISON

HIGHWAY DESIGN AND MAINTENANCE FOR TOMORROW'S TRAFFIC

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Abstract: The condition of the roadway surface plays a significant role in preventing damage to cargo and to increasing safety in the shipment of goods by ground transportation. Increasing traffic volumes and loads have led to changes in pavement design concepts to adequately handle the increased stresses occurring in the roadway. Additionally, because of these increased stresses, proper preventive maintenance has become a major concern of highway engineers. This problem is complicated by the rapid rise of inflation and declining revenues available for pavement maintenance which has resulted in the highway system deteriorating at an alarming rate in recent years. It will be shown that programmed maintenance that includes timely resurfacing before rapid deterioration begins will extend the pavement life more economically than in the case where pavement resurfacing is delayed. The consequences of deferred action are increased fuel costs, increased operator costs, increased damage to cargo, increased safety problems, rerouting of commercial traffic and general user dissatisfaction. A pavement management system is essential if we are to obtain the most efficient use of the highway dollar and establish those roads in immediate need of repair.

Key words: Asphalt concrete; California Bearing Ration (CBR); DAMA; design period; Equivalent Axle Load (EAL); highway system; pothole; rehabilitation; Resilient Modulus (M_r).

The shipping of goods and services to every corner of the country and the individual freedom of mobility Americans enjoy can only be possible with an efficient, well-maintained system of roads. The total vehicle transportation industry -- including moving of goods, services and people and the construction and maintenance of the road system -- represents more than 18 percent of the nation's Gross National Product (GNP). Forty percent (40%) of the world's passenger cars and trucks are found in the United States. To say that a good highway system is vital to our economy is an understatement since 65% of the country's communities are without rail service and must rely on highway transportation for goods and services. Additionally, we see more and more miles of track abandoned each year primarily due to poor track condition which is the result of years of deferred maintenance.

To adequately and efficiently handle the 154 million vehicles that are registered in the United States, the roadway surface plays a significant role in preventing damage to cargo and to increased safety in the shipment of goods. Included in this fleet of vehicles are 32 million trucks hauling goods and services to virtually every corner of the nation. Additionally, it should be noted, there are over 3.5 million trucks sold each year in the United States and sales of large heavy duty trucks has been increasing.

For a pavement to perform its function, it must be properly designed and built to avoid the highway engineer's nemesis the "pothole." With increases in both the volume and weight of traffic, design concepts have also changed from granular bases to Full-Depth asphalt designs. And methods of pavement design are going from an empirical approach to a more rationale approach that is based on a mechanistic model that considers material properties and failure criteria. The Asphalt Institute has developed a procedure based on current knowledge of multilayered elastic theory, fundamental material properties, and realistic failure criteria. The method utilizes the computer program DAMA to develop design curves that determine proper pavement thicknesses. As with other pavement design procedures, the factors traffic, subgrade, environment and pavement materials are analyzed.

The traffic analysis determines how many vehicles of different types are expected to use the pavement. Traffic considerations then include present traffic, the design period, and a growth factor. Knowing the types and numbers of vehicles expected to use the road, we then express that data in a single number known as the EAL, Equivalent Axle Load. The total number of Equivalent Axle Load application contributed by a single vehicle is called its truck factor. By determining the truck factor of all the vehicles of all types expected to use the road during the design period we arrive at the design EAL.

Next, we must consider the strength of the subgrade that will support the pavement and traffic. The strength of the subgrade is characterized by the subgrade's Resilient Modulus, M_r. M_r is a ratio between stress and strain -- the stress exerted on the subgrade soil and the resulting strain produced in the subgrade. If other more familiar soil strengths are known, such as the CBR (California Bearing Ratio) they can be used to estimate the soil's Resilient Modulus.

After considering traffic and subgrade characteristics we must consider the environment that will influence the pavement. The computer program allows for all types of climatic conditions and the design charts in manual consider environments where frost is a factor which is one extreme condition.

The final factor that must be considered is that of the pavement materials to be used in the construction of the road. Several different materials are considered for the surface and base which include asphalt concrete, emulsified asphalt mixtures, and untreated aggregate.

Utilizing these factors as input design, charts are presented that allows one to design an asphalt concrete pavement to support the anticipated loads. The design chart is a log-log plot with design EAL values running horizontally and subgrade Resilient Modulus values running vertically.

The Asphalt Institute's design procedure does afford the design engineer the tools to design a pavement for the anticipated traffic and soils conditions that will last for the design period. However, it should be recognized that the usual thickness design period is selected for about twenty years and that if more or heavier traffic loads use the road than were expected, the road wears out sooner than the design period. If we arbitrarily extend the pavement life beyond its design period by not maintaining the road properly and if we experience heavier loads than expected, our system will change from the finest-in-the-world road system to a two-million-mile obstacle course of ruts, bumps, cracks and potholes.

Bad roads waste billions of gallons of fuel every year. They damage vehicle tires, brakes, and suspension systems and take a heavy toll in injuries and deaths. Inferior roads swell transportation costs, which raises the price of everything at the market place. They lead to millions of hours of costly delays that interfere with both work and leisure time. Unfortunately, the problem is not limited to isolated areas, but rather we see signs all over the United States.

To appreciate the scope of the situation, consider that there are approximately two million miles of paved roads in the United States -- a little more than half of the country's total road mileage. Of those two million miles, 31 percent carry 87 percent of the nation's traffic. This 31 percent includes America's major highways -- the interstate system, the arterials, and the collectors. The Federal Highway Administration took a look at these roads in 1975 and found that 100,000 miles were already in "poor" condition and required immediate attention. The report also noted that each year 22,000 miles of highway in "fair" condition deteriorate into "poor" condition and 4,000 miles in "good" condition fall into "fair" condition. From the report a goal of rehabilitating 26,000 miles each year was established. That goal was nearly achieved in 1976 but by 1980 less than half of the goal was reached adding more mileage to the backlog. Remember, these figures represent only 31 percent of the paved roads in the United States.

The local service roads comprise the other 69 percent and are generally in worse condition than the major highways.

Part of the problem is tied to our inflationary spiral with tremendously increasing costs for repair. Highway construction and repair costs have risen 50 percent faster than general wholesale prices. Another part of the problem is highway financing. In terms of constant dollars, the funding available for rehabilitating our major highways has declined drastically.

Another part of the problem is how pavements are affected by vehicle axle loads. As stated earlier, if more or heavier traffic loads use the road than were expected, the road will wear out sooner than the design period. The weight and configuration of the axles along with the number of repetitions is what determines the pavement life or time to first maintenance. For example, 2,500 cars put the same load on a road as one legally loaded truck and one overloaded truck may be as harmful as two or three legally loaded trucks. Pavements usually fail due to fatigue which means that it takes many applications of loads before failure. As either the amount of the load or the number of axle loads increase, the pavement life decreases. As the pavement thickness increases, its ability to carry more and heavier loads also increases. Damage to the pavement increases very rapidly as the load is increased. For example, one 18,000 lb. load causes almost ten times more damage to the pavement than does a 10,000 lb. load. To illustrate, we will assume that one 18,000 lb. load on a single axle with four tires causes one unit of damage. Then a semi-trailer with a 10,000 lb. front steering axle and two single 18,000 lb. load axles loaded to maximum loading would have 2.1 damage units with a pay load of 16,000 lbs. If the same truck was overloaded to 20,000 lbs. the pay load would increase 25 percent but pavement fatigue damage would increase almost 50 percent to 3.1. Taking it one step further, if the same truck was loaded to 24,000 lb. axle loads it would cause almost three times the damage as the 18,000 lb. axles and increase the pay load by only 75 percent. Going the other way, research has shown that a 42,000 lb. evenly loaded tri-axle will cause only one unit of damage.

Another form of pavement failure is permanent deformation or rutting due to loading in the wheel paths causing an overloading of the subgrade. The thicker the pavement the less load on the soil. Therefore, weaker soils require a thicker or stronger pavement.

Recognizing that pavements deteriorate with loading conditions and age the lifecycle of a pavement shows a general decline for its design life. During the first 8 to 10 years change in pavement condition is minimal. Only after 15 or more years does a typical pavement cross the line from good condition to fair. At that point the pavement has gone through 75 percent of its design life and its condition has declined about 40 percent. To return the pavement to good or like - new condition, some rehabilitation is required. If that rehabilitation is deferred, deterioration accelerates, cracks quickly become potholes and surface irregularities degenerate into rough surfaces. Within a short time, pavement conditions drops another 40 percent from fair to poor or very poor. At this point the pavement requires complete reconstruction. This means that each dollar you're saving by deferring rehabilitation will cost you $4 to $5 a short time later. Obviously, deferred maintenance costs more.

Conversely, timely maintenance or rehabilitation saves money in the long run and extends the design life of the pavement. A hot-mix asphalt concrete overlay placed at the right time can increase the pavement life by more than 60 percent. Therefore, a timely rehabilitation program should be established to extend the pavement life which will result in significant savings to the road users.

A rehabilitation schedule should be planned around the design life of each pavement, with rehabilitation done before the pavement condition crosses the line between good and fair. Once the schedule has been established it should be communicated to decision makers so that adequate funding will be provided. The consequences of deferred action are increased fuel costs, increased safety problems, rerouting of commercial traffic, damage to cargo, and general user dissatisfaction. Also deferring rehabilitation costs more. For example, compare two jurisdictions with average pavement in the good to fair range. One opts for an overlay program immediately and begins construction of 100 miles per year while the other defers action for 5 years in order to build up treasury reserves and with the thought that the pavement would last that long. When jurisdiction number two does begin to upgrade their system the costs are substantially more per mile and a 10-year summary would indicate that jurisdiction number one actually upgraded 1000 miles at one-half the cost that jurisdiction number two expended to upgrade 500 miles.

When people hear the term "worn roads" they immediately think of potholes. Unfortunately, potholes are a sympton of a problem almost beyond repair. A crack in the road is the same as a crack in the roof. In both cases, one should repair the minor damage before it becomes more extensive and more expensive to correct. We should maintain our system of highways in the United States to provide the most economical use to the highway-user.

THE FAILURE OF MANHOLE COVERS ON TANK TRAILERS DURING ROLLOVER

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Abstract: Accidents involving the rollover of bulk petroleum transport tankers in Michigan have frequently involved the wholesale failure of the manhole cover assemblies mounted at the top of the tank. Such failures result in the rapid spillage of large quantities of liquid and, often, are the occasion of large fires. This paper describes a research effort by which the pertinent accident experience was defined and full-scale rollover experiments conducted to establish the mechanisms by which manhole cover failures occur. Results show that the internal fluid pressures prevailing inside the tank vessel at the moment of ground impact are three to five times higher than the pressure level currently specified in federal regulations which mandate the proof testing of manhole cover assemblies. Recommendations are made for a type of proof test which would assure the level of manhole cover strength needed to prevent failure.

Key words: Fire; fuel spillage; manhole cover; rollover; tank trucks.

The spillage of fluid product in highway accidents involving petroleum transport tankers derives primarily from tank failures induced in rollover impacts. Indeed, data gathered in a recent two-year period in the State of Michigan show that only one in 25 product spill incidents occurred due to a non-rollover accident. In a research study [1] conducted by the Highway Safety Research Institute of The University of Michigan, a review was made of tanker accidents reported by the Michigan Fire Marshall's Office to identify mechanisms of tank failure occurring in rollover accidents. Upon observing that the most common failure mechanism involved the manhole covers which are mounted on the top of each tank compartment, a set of experiments was then conducted to establish the impact pressure conditions to which manhole covers are subjected in a rollover.

Review of Tank Failure Data: Using the State of Michigan Fire Marshall's Incident Reports for 1977 and 1978, 33 accidents were identified for study in which the tanker overturned with the possibility of damage to the tank. Selection criteria included (1) vehicle overturn, (2) tank capacity in excess of 5,000 gallons, (3) a single-trailer articulated vehicle or a "doubles" vehicle. Straight, local delivery vehicles were not included. For each accident, the damage to the involved tank was determined through either direct examination of the vehicle, interviews with trucking company staff, or from State Police incident reports.

In 25 of the 33 instances, the tank was found to have been significantly damaged. The level of damage ranged from gouges and dents to cave-in of an entire side of a doubles train.

Twenty-three of the vehicles had suffered loss of product by means of the following failures:

- 13 cases - manhole cover dislodged or leaking
- 5 cases - split in tank weld seam
- 4 cases - puncture in tank wall
- 1 case - both split seam and puncture

Manhole Cover Failures: Manhole cover failures ranging from total dislodging of the cover to a sustained leakage after rollover have been observed under a variety of rollover impact conditions. Manhole cover failures were experienced with tanks which struck rigid pavement as well as snow banks. Shown in Figure 1, for example, are tanks having

Figure 1. Flattened sidewalls of a double tanker which rolled over on the roadway.

the distinctly flattened sidewalls which result from rollover onto a rigid pavement, the situation most likely to dislodge manhole covers.

Shown in Figure 2 is a photo taken at the accident scene of a burned tanker which has lost two of its manhole covers after rolling over on a Detroit expressway. One of the covers can be seen laying on the ground near the tank.

Figure 2. Accident scene showing gasoline tanker that has suffered dislodgement of two manhole covers, with ensuing fire.

Although the anecdotal information illustrated that a problem exists, a statistically-sound basis for inferring the role played by manhole-cover failure is not possible, since the available data are so sparse and since reporting of specific types of manhole-cover failure was not done consistently. Nevertheless, the complete dislodging of manhole covers appears to constitute a quite hazardous mode of tank failure since the fluid load is released at such a high flow rate that a "river" of fuel is created. The "river" flows rapidly along the natural drainage profile of the roadway, posing a particular threat to other vehicles either involved in, or stalled by, the accident. If the fuel is ignited, of course, the rapid flow of the "river" implies a rapidly advancing flame front. Fuel releases involving tank seam split, puncture, and the like, appeared to result in more containable fires and to pose less risk to persons since the spillage rates of fuel were much less than those experienced in the event of manhole cover dislodgement.

Study of the Integrity of Manhole Covers: Shown in Figure 3 is a closeup photo of a dislodged manhole cover having a design such as has been popular in Michigan, and in much of the U.S., for use on tankers transporting gasoline and fuel oil. The manhole cover assembly is normally situated on top of each tank compartment, between two protective rails. The device shown in the photo serves as a covering over the compartment's manway and has installed a "filler cap" for conventional gravity loading of the tank. Also, the assembly includes devices to limit internal pressure and to provide for the recovering of vapors when the tank is loaded through the bottom.

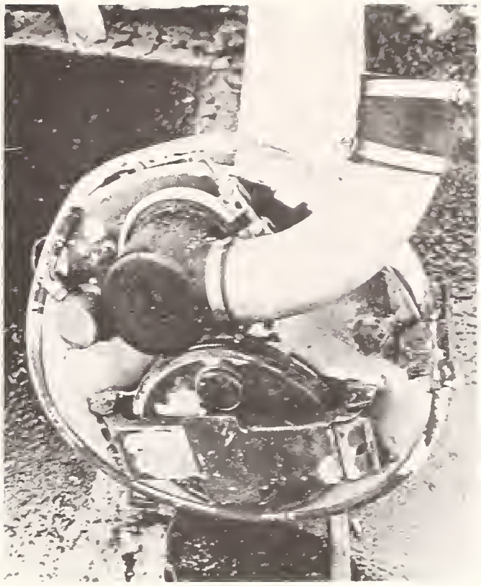

Figure 3. Typical manhole cover assembly.

The key design feature which has led to concerns for manhole cover integrity involves the clamping band by which the cover plate is fastened to the flanged opening to the compartment. The photo, Figure 3, shows distortion in the clamping band by which this manhole cover became detached from the tank. The integrity of such manhole-cover assemblies and their attachment hardware is currently addressed by the Code of Federal Regulations #CF49-178.341-3 which states:
(a) Each compartment (of a tank vehicle of type MC-306) in excess of 2,500 gallons capacity shall be accessible through a manhole of at least 11 x 15 inches. Manhold and/or fill opening covers shall be designed to provide secure closure of the openings. They shall have structural capability of withstanding internal fluid pressures of 9 p.s.i.g. without permanent deformation. Safety devices to prevent the manhole and/or fill cover from opening fully when internal pressure is present shall be provided.
The accident experience cited in the preceding section led to the hypothesis that the 9 psi internal pressure requirement of the regulation was inadequate for assuring that manhole covers would not become dislodged in rollover impacts.

Experimental Program of Rollover Impact: In order to test the stated hypothesis, a set of experiments was planned to determine the maximum level of internal fluid pressure which is likely to occur in a gasoline tanker which suffers a rollover impact against a flat paved surface. A set of four individual experiments was conducted using full-size gasoline tank trailers.

Experiment #1 - The first test was conceived as representing a severe rollover event such as would derive if a rapid steering maneuver was attempted such that the tank trailer could gain a large level of roll velocity before its tires came off of the ground. Computerized simulation was employed to predict the roll and elevation trajectory of the vehicle. The final impact attitude is shown in Figure 4, indicating that in a very severe rollover event of this type, the tank arrives at a rather steep roll attitude by the time ground contact is made. The impact condition further entails a final angular velocity of 2.6 rad/sec and a vertical velocity at the mass center of 16.2 ft/sec.

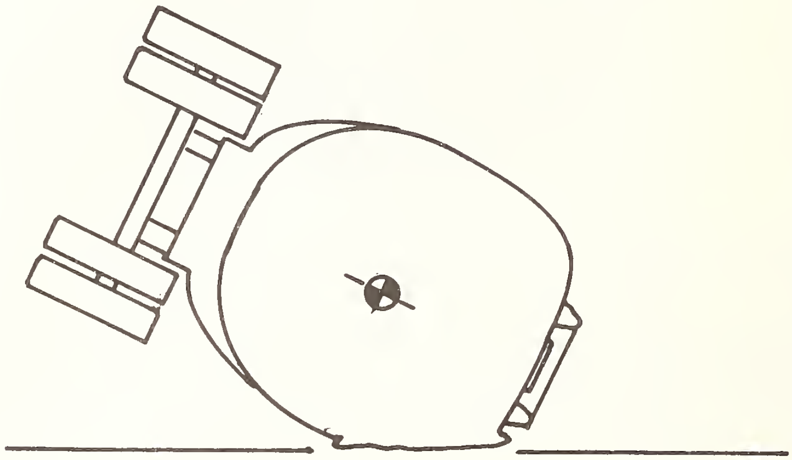

Figure 4. Tanker orientation at moment of impact in computerized simulation of severe rollover event.

A full-scale kinematic equivalent of this impact condition was achieved through use of the test rig shown in Figure 5. By this arrangement, the vehicle was set up to achieve a free roll motion about a selected longitudinal axis which was near the center of the tanker's axles. The figure shows certain auxiliary vehicles which were used to hold down or rigidly locate elements of the pivot mechanism. The vehicle is perched upon two upright poles which support the tank by means of rollers set into box-like fittings at each end of the tank. Each roller, which is fastened to the pole, bears against an inclined plate such that the vertical load of the tanker tends to push the pole away from the tank. Upon suddenly releasing tension on the overhead cable, the poles are both pushed out and the tanker falls in a free rotation about the pivot. Since the tank strikes the ground with a contact force which passes very nearly through the center of percussion of the tanker, residual motions following impact are insignificant.

The test vehicle itself is a 7,700-gallon, 10-ga steel-shell tanker comprising the full trailer of a typical Michigan double. The tank has

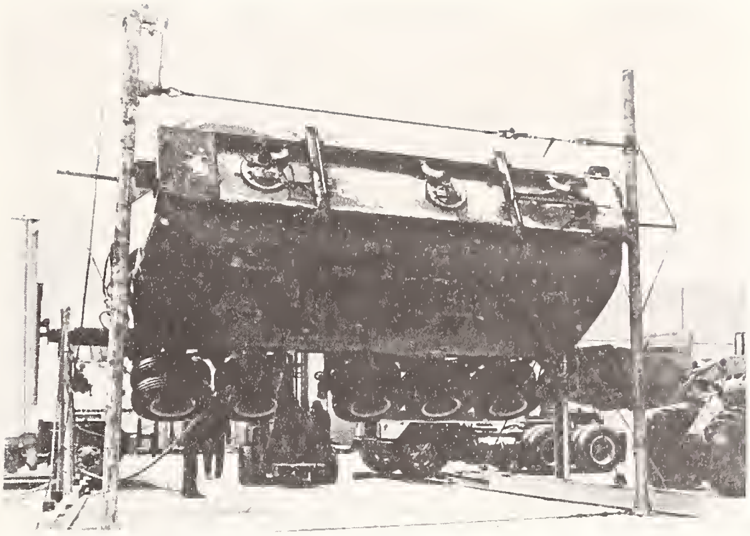

Figure 5. Test vehicle set up for severe rollover experiment.

three compartments, of capacities 3,550 gallons, 1,450 gallons, and 2,700 gallons, numbered 1, 2, 3, respectively, from left to right. For the first experiment, the 1st and 3rd compartments were nominally filled with water, leaving a three-percent ullage volume, or free space, in both loaded compartments. This loading condition was chosen since it yielded the design gross load of the vehicle, 65,000 lbs. The manhole covers were all of the clamping-band type shown earlier. The manhole cover in the filled #1 compartment was left in its as-designed state while the #3 cover was mechanically blocked to prevent its dislodging. Transducers were installed to record two fluid pressures adjacent to the manhole in both the 1st and 3rd compartments, two acceleration signals normal to the impact point, and the instantaneous angular position of the vehicle during its fall. Data were recorded on an FM tape recorder, as well as on oscillographic pen-chart recorders.

Shown in Figure 6 is a photo of the tanker striking the pavement. In Figure 7 we see the "river" that was created within two minutes following the impact-dislodgement of the #1 manhole cover. Subsequent inspection showed that the bulkhead between compartments #2 and #3 had failed in the vicinity of the impact deformation. Further, following completion of the second impact experiment using the same test vehicle, it was discovered that an artificial cushioning of the internal fluid pressures had taken place due to partial collapse of air tanks which existed inside each of the three compartments. The air tanks had been installed inside the compartments during a previous research study [2] for purposes of displacing fluid so that a water load would yield the same inertial parameters as a reference loading of gasoline. Thus, the fluid pressure data taken in experiments 1 and 2 were looked upon as

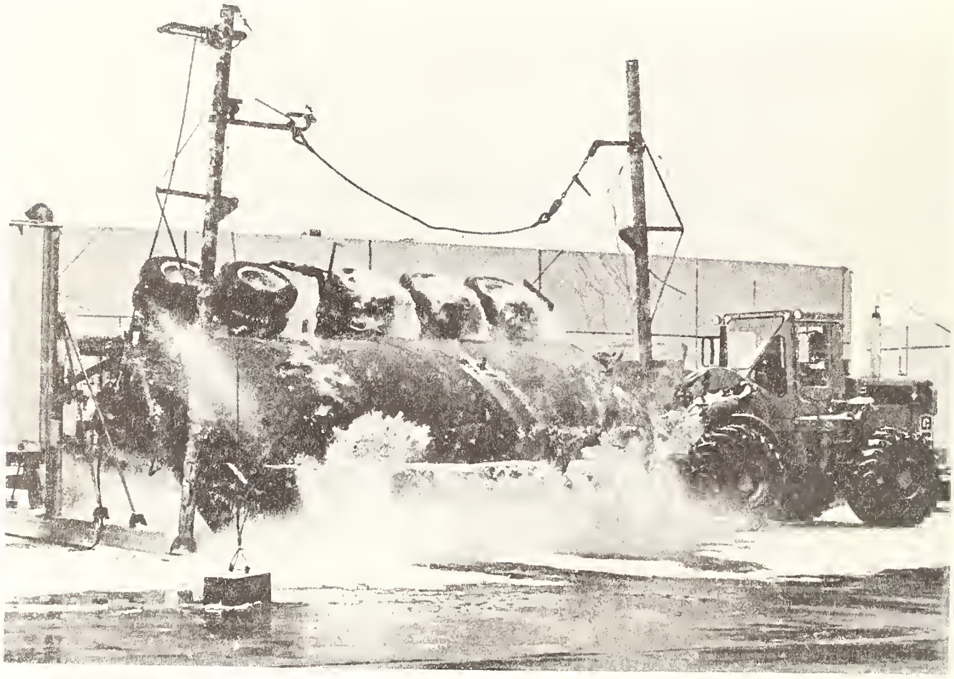

Figure 6. Test vehicle striking pavement.

Figure 7. "River" of fluid is spilled from open manhole. Note dislodged manhole cover some 15 ft to the right of tanker.

representing an unknown degree of underestimation of pressures actually pertaining to such rollover impact situations. Nevertheless, the results of these experiments do aid in obtaining an understanding of the mechanisms contributing to internal pressure.

Shown in Figure 8 are the recorded time histories from pressure transducers and accelerometers installed on the vessel. From the top, we see the two pressure signals from compartment #1, next the two pressure signals from compartment #3, and finally, the two acceleration signals (which show signal A_1 limiting at 20 g's while the broader range accelerometer, signal A_2, continues to yield a measurement). The pressures in compartment #3 pertain to a condition in which the manhole cover is blocked to prevent dislodgement. The pressures in compartment #1, on the other hand, provide a measure of the pressure causing one specific manhole cover to dislodge. Although we see variations in the noise content of the four pressure signals, the nominal response entails a low frequency ($\tilde{}$ 10 Hz) peak in the range of 20 to 25 psi and high frequency (> 100 Hz) spikes from 30 to 50 psi. Interestingly, the signals obtained in the "blocked" compartment are virtually the same as those obtained in the compartment in which the 20-inch diameter orifice was opened, upon having dislodged the manhole cover. This result was initially surprising since it had been expected that failure of a manhole cover would act to relieve the pressures experienced in the involved compartment. The observation can be explained, however, on the basis of the very limited time period over which peak pressures are sustained, as contrasted with the time needed to accelerate a sufficient quantity of fluid out of the manhole opening to provide a pressure-relieving effect.

The accelerometer signal, A_2, provides a continuous recording of accelerations normal to the ground plane, showing a noisy, and relatively low-level, response for the first 30 milliseconds, followed by a rather sustained pulse of approximately 25 g's amplitude. Double integration of the A_2 signal, by graphical means, yields a net deflection of the tank wall which agrees well with the nominal 8-inch maximum deformation measured on the vessel.

Experiments #2 Through #4 - The subsequent rollover tests were of the simple "tipover" variety by which the tanker was brought just beyond the point of unstable equilibrium by means of a laterally-affixed tow cable, and then tipped over to strike the ground at a 90° roll angle. Again, all manhole covers but one were mechanically blocked to prevent dislodgement. In each case, the nominal result of the test was as depicted in Figures 9 and 10. Namely, the rollover impact caused an extensive spray of fluid from the low-pressure vents on each manhole cover while the "unblocked" manhole cover, at the right in these figures, was dislodged.

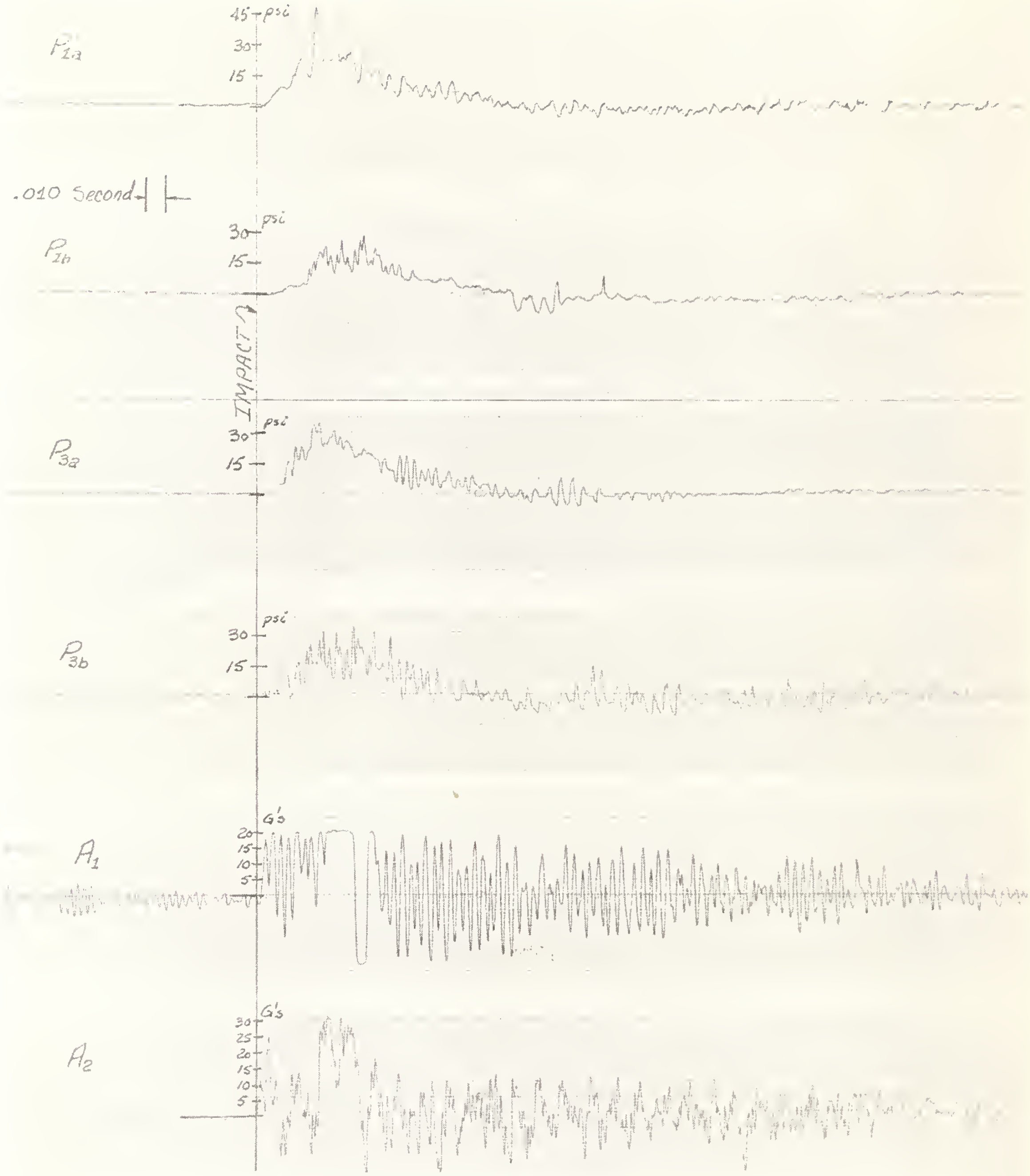

Figure 8. Impact response data gathered in severe rollover experiment.

Figure 9. Fluid spraying through low pressure vents at impact, while large plume at right derives from dislodgement of manhole cover.

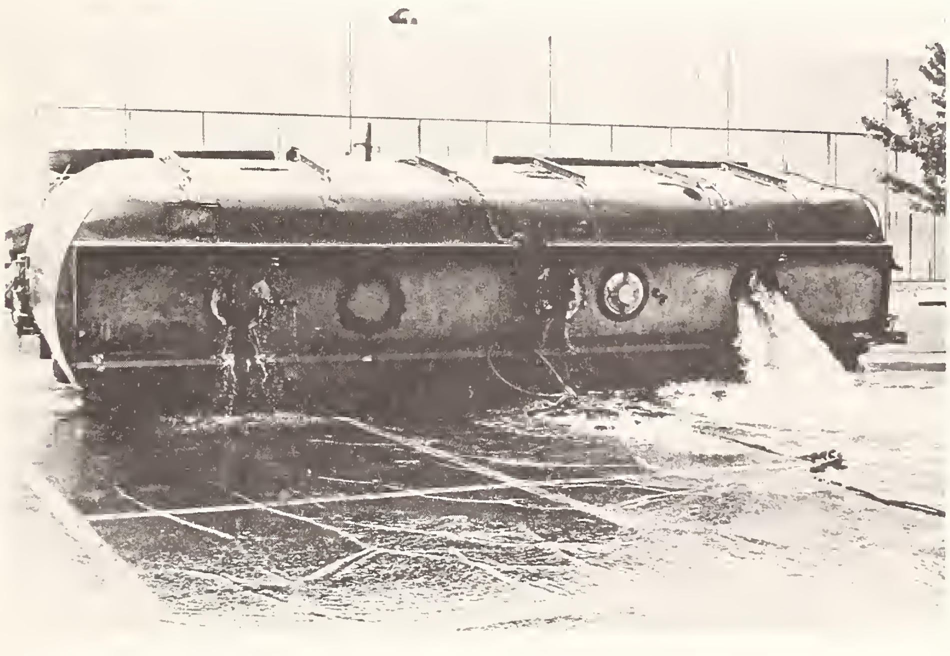

Figure 10. Leaking low pressure vents at left, manhole cover dislodged at right.

In the last two of the tipover experiments, a more extensive set of pressure measurements were made inside the vessel and the initial ullage or vacant volumes were made equal to 10 percent and 3 percent, respectively, of the total volume in each compartment. An increase in the ullage space (to 10 percent) was of interest since examination of the tank deformations resulting from the previous (3 percent ullage) tests showed that the volume displaced through crushing of the tank was approaching 3 percent of the tank volume. Thus a substantial fraction of the peak level of internal pressure would appear to have derived from simple compression of the air space inside the vessel. According to the Ideal Gas Law, of course, one would expect that fluid pressure levels would reduce if the initial ullage space is increased and if the "crushed volume" remains constant. The results illustrate that, while this hypothesis is qualitatively correct, the numbers indicate that internal pressure is actually determined from a fairly complex set of mechanisms. Data from the tipover test run with 10 percent ullage volume show a low frequency peak pressure of 21 psi, while the corresponding test with 3 percent ullage produced a low frequency measure of peak pressure equal to 27 psi. Pressure spikes reached an average of 41 psi for the 10 percent ullage case, and 54 psi for the 3 percent ullage case. Accordingly, for this simple index in ullage volume, it is clear that compression of the trapped air is a significant mechanism contributing to the internal pressure which causes dislodgement of manhole covers.

The second primary mechanism involves the pressure gradient induced by acceleration of the bulk fluid mass. It can be shown that the pressure value, p, experienced at any point which is h (ft) below the surface of a liquid of density, ρ, (lb/ft^3) whose total bulk volume is subject to a normal acceleration having amplitude A_n (g's) is given by:

$$p = \rho \frac{A_n h}{144} \quad \text{(psi)}$$

For the nominal average values of acceleration experienced during the rollover impacts (approximately 6 to 10 g's), average pressure levels equal to approximately 10 to 16 psi at the center of the manhole will derive from the "bulk acceleration" mechanism. Higher levels of short-lived pressure pulses presumably result from the nonuniform acceleration condition and from acoustic, or shock, phenomena within the fluid.

Another mechanism contributing to fluid pressures at the manhole derives from the horizontal component of the impact acceleration vector. This component is produced when the tank strikes the ground because there exists a horizontal component of velocity which becomes reduced toward zero upon generation of frictional forces between the tank shell and the pavement. The horizontal acceleration level can approach or even exceed 1 g since high normal forces are created by the high normal acceleration levels such as were illustrated in Figure 8. Over the length of the water column measuring 8 feet along the original vertical axis of the tank section which was tested, for example, a 10-g normal acceleration at impact will produce a 1-g horizontal acceleration and a 3.5 psi additional pressure increment at the manhole cover, presuming a 0.1 frictional coupling between shell steel and pavement.

Conclusions Drawn from the Rollover Experiment: Insofar as the fluid employed in the foregoing tests was water, whose density is 8.3 lb/gal compared to 6.1 lb/gal for gasoline, the test measurements of the impact pressure applied to manhole covers are somewhat conservative, that is, somewhat higher than would derive if the load had been gasoline. Given that more than one mechanism contributes to impact pressures applied to manhole covers, however, the degree of conservatism cannot be easily determined. For example, if the only mechanisms giving rise to internal pressure, at impact, were those associated with bulk acceleration of the fluid mass, the pressures to be expected from the same volume of gasoline would be equal to the ratio of the fluid densities of gasoline versus water times the pressure level measured using water as the test fluid (viz., gasoline pressure = (6.1/8.3), or 74 percent of the water pressure level). In actual impacts, however, the gasoline pressure would be higher than the indicated 74 percent of water pressure levels since, clearly, the compression of the ullage space due to tank deformation constitutes a major mechanism for generation of internal pressure at impact. Obviously, the ullage space compression mechanism is insensitive to the density of the liquid product.

Accordingly, it was concluded that the measured pressure levels provide a reasonable, and somewhat conservative, estimate of impact pressures bearing upon manhole covers in the rollover of actual gasoline-transporting tankers of the type examined. The worst condition expected for simple tipover onto a rigid pavement entails, approximately, a 27-psi pressure pulse lasting on the order of 50 milliseconds, with 2-millisecond pulses reaching as high as 60 psi. On the strength of these findings, recommendations for improved gasoline tankers in the State of Michigan included a specification of 50 psi as the steady-state pressure containment capability of manhole covers.

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APPENDIX

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DAMAGE PREVENTION IN THE TRANSPORTATION ENVIRONMENT

34th Meeting

October 21-23, 1981

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U.S. DEPT. OF COMM. BIBLIOGRAPHIC DATA SHEET (See instructions)	1. PUBLICATION OR REPORT NO. NBS SP 652	2. Performing Organ. Report No.	3. Publication Date April 1983

4. TITLE AND SUBTITLE

Damage Prevention in the Transportation Environment
Proceedings of the 34th Meeting of the Mechanical Failures Prevention Group held at NBS, Gaithersburg, MD, October 21-23, 1981

5. AUTHOR(S)

T. Robert Shives, Editor

6. PERFORMING ORGANIZATION (If joint or other than NBS, see instructions)

NATIONAL BUREAU OF STANDARDS
DEPARTMENT OF COMMERCE
WASHINGTON, D.C. 20234

7. Contract/Grant No.

8. Type of Report & Period Covered

Final

9. SPONSORING ORGANIZATION NAME AND COMPLETE ADDRESS (Street, City, State, ZIP)

National Bureau of Standards, Washington, DC 20234
Office of Naval Research, U.S. Department of the Navy, Arlington, VA 22217
Naval Air Systems Command, U.S. Department of the Navy, Washington, DC 20360
NASA, Goddard Space Flight Center, Greenbelt, MD 20771

10. SUPPLEMENTARY NOTES

Library of Congress Catalog Card Number: 83-600521

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)

These proceedings consist of 27 submitted entries (25 papers and 2 abstracts) from the 34th Meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards, Gaithersburg, Maryland, October 21-23, 1981. The subject of the symposium was damage prevention in the transportation environment. Areas of special emphasis included packaging for the transportation environment, research in the railroad industry, damage prevention in the railroad industry, designing for the transportation of hazardous materials, and highways.

12. KEY WORDS (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

Airline environment packaging; hazardous materials transportation; highway design; marine environment packaging; packaging techniques; railroad environment packaging; shipping; transportation; transportation damage prevention; trucking environment packaging.

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14. NO. OF PRINTED PAGES

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15. Price

$8.00

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