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DEVELOPMENT OF SMALL SCALE IMPACT SIMULATION TECHNIQUES

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FINAL REPORT

Prepared for

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Federal Railroad Administration

by

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TABLE OF CONTENTS

LIST OF FIGURES

EXECUTIVE SUMMARY

In derailments and switchyard operation, tank cars carrying hazardous materials frequently experience impacts that result in head punctures causing loss of lading material or other damage. It is not practical from a cost viewpoint to conduct full scale testing to verify that a particular tank car design will perform satisfactorily in all possible impact situations. Hence there is a need for reliable small scale testing procedures to simulate the response of full scale tank cars in impact situations. The objective of this task, therefore, is to investigate the critical parameters in tank car impacts, and to develop a methodology for small scale testing so as to evaluate existing and improved tank car designs. The effects of such things as tank head design, amount of lading, environmental conditions, impact speed, etc. were investigated.

An extensive literature search was conducted to examine past work regarding failure modes; quantitative estimates of impact parameters such as impact velocity, impact energy, tank car size and weight, temperature and material characteristics; effects of impacts on structures; analysis methods; scaling laws; small scale test facilities; and instrumentation equipments. The results of the literature search were then used to recommend a small scale testing methodology for testing current and new designs for tank cars. This methodology could be used to accurately predict the performance of full scale tank cars thereby resulting in significant cost saving.

Some of the more important findings from the literature search are summarized as follows:

a. The threshold impact velocity for puncture of some conventional size cars (112A340W car for example) can be seen in the range of a 9 to 13 mph depending on weights, lading, etc. The performance standard by DOT specifies tank cars should be capable of impact speeds up to 18 mph depending on the size of the impact car and impacted/backup car configuration. Hence it appears that a small scale testing facility should be capable of simulating impact velocities of up to about 20 mph.

b. Internal pressure decreases the threshold velocity for puncture. A completely full car (no outage) reduces the threshold impact velocity even further. Hence the small scale facility must be capable of simulating lading and internal pressure up to about 100 psig.

c. Temperature change can substantially alter the impact energy that a tank car steel specimen absorbs in the standard Charpy V-notch test. This indicates that tank head puncture resistance may decrease quite substantially when temperature is lowered beyond the transition temperature of the steel used. Tests to identify the effect of temperature have not been located in the literature searched.

d. Impact forces of the order of 10^6 pounds and associated indentations of the order of 10 to 15 inches can be experienced in ductile materials. Under high pressure and little outage, primary failure mode is shear along the periphery of coupler area.

e. In head on collisions where gravity effects can be neglected, equal velocity scaling can be used provided material properties are identical between model and prototype. Strain rate effects on material behavior can be a complicating matter which should be considered in the testing methodology.

f. A vertical drop testing facility is a low cost means of screening new head designs and validating theoretical models but does not adequately simulate lading effects.

g. A horizontal test configuration using about 1/5 scale models appears to be a good test configuration capable of simulating impact velocity, internal pressure, lading, wheel friction, etc. However a cost study should be conducted to optimize the scale size.

h. A head shield significantly increase threshold velocity for head puncture (up to 20 mph without excessive damage).

i. The test facility should be capable of simulating other types of protective devices such as foam mitigators.

j. There are analysis methods available that appear to be semi-empirical in nature. They do not adequately account for lading, strain rate effects, and impact location.

Based on the findings from the literature search, a testing methodology as illustrated in Figure 1 is recommended. The DOT Performance Test specified in the Federal Register Volume 42 No. 179, 1977, (DOT Docket No. HM-144) Page 9 is recommended as a standard for simulation by a small scale test facility. However, before committing newer or modified tank head designs to simulation testing, a screening procedure is recommended to evaluate the potential of the tank car design of surviving the DOT standard test. It is proposed that the head design be evaluated by 1/5 scale vertical drop test to estimate energy absorption capability, and to obtain measurements of force-deflective characteristics of the head. If the head design looks promising, a theoretical analysis will be conducted using finite-element methods to estimate force-deflection (including strain rate effects). The model will be validated by comparison with the experimental data from the vertical drop tests. Knowing the force-deflection characteristic of the heads, a dynamic analysis of the collision process will be made to estimate the peak impact force and maximum indentation as a function of impact velocity. A suitable failure criteria will be developed to estimate the extent of indentation before puncture occurs. If the theoretical results indicate that the tank head has a high confidence of surviving the 18 mph impact velocity required by the DOT standard test, then the tank head can be committed to detail design and fabrication for testing in the 1/5 scale model arrangement shown in Figure 2. The test arrangement proposed simulates coupler to tank head impacts at relative car speeds of 18 mph when

(a) The full scale weight of the impact car is at least 263,000 pounds.

(b) The impacted tank car is coupled to one or more backup cars which have a total full scale weight of at least 480,000 pounds and the hand brakes are applied on the first car.

(c) The impacted car is loaded with water at 6% outage with an internal pressure of at least 100 psig. Other outage and pressures can be used as desired.

A test is considered successful if there is no visual leakage from the standing car.

The test arrangement will be suitably instrumented to measure impacted force versus time, indentation versus time and motion of the cars as a function of time. The results will then be compared with the previous theoretical analyses to validate the theoretical models and to assess the importance of such things as strain rate. Preliminary estimates of full scale performance will be made using the scaling laws discussed in the report. However, the scaling laws may not adequately account for inelastic behavior of the heads; particularly when strain rate effects are important. That is why the theoretical models are so important. Once the theoretical predictions are validated by comparison with the 1/5 scale DOT standard impact tests, then the behavior of the full scale tank cars can be reliably predicted using the theoretical models. The model can also be used for sensitivity studies to determine threshold velocity for puncture as a function of such things as head thickness, tank car weight, etc.

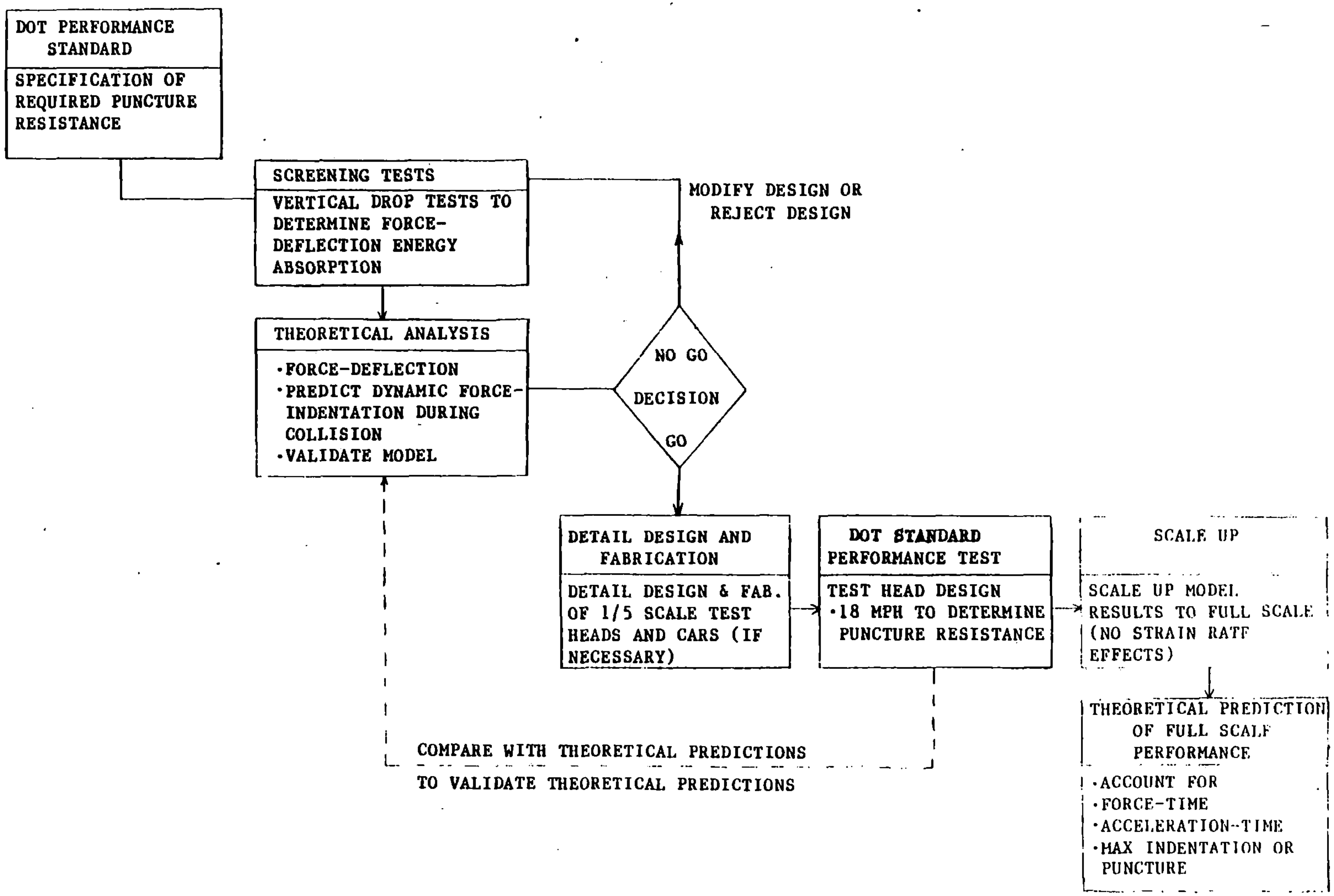
1. INTRODUCTION

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 practical from a cost viewpoint to conduct full scale testing to verify that a particular tank car design will perform satisfactorily in all possible impact sitations. Therefore, there is a need for reliable small scale testing procedures to simulate the response of tank cars in impact situations.

The following report contains the result of a study conducted to develop and validate small scale test procedures. An extensive literature search was conducted to identify and analyze small scale impact test facilities, test results, scaling laws, analysis techniques and special instrumentation requirements. Based on these results a small scale testing methodology was developed to predict the effect of impacts on tank car structures.

FIGURE 1

ILLUSTRATION OF METHODOLOGY FOR SMALL SCALE TESTING
TO EVALUATE PUNCTURE PERFORMANCE OF TANK CAR HEAD DESIGNS



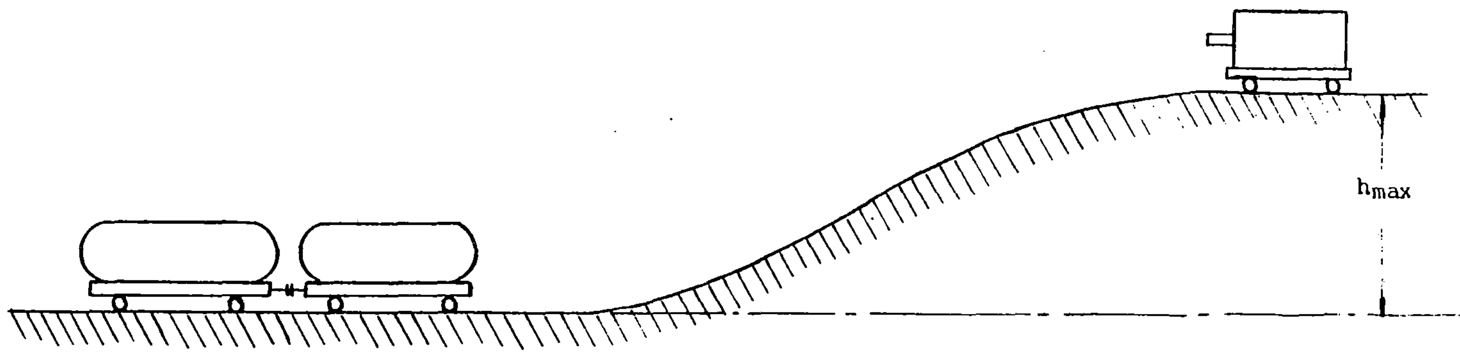


Figure 2 Impact test setup

2. STATEMENT OF THE PROBLEM

In derailments and high speed impacts of railroad tank cars, couplers may become disengaged and one coupler may override the other. The shell head of tank cars carrying hazardous material have been punctured by the coupler of an adjacent car, or in some cases by some other equipment (projectile). Figures 2-1 through 2-4 (from Reference 1) illustrate four possible switchyard impact scenarios for tank puncture which have been investigated in considerable detail. Analysis of accident statistics had previously brought to light a correlation between certain initial conditions and the occurrance of major switchyard accidents. Briefly, the critical initial conditions appeared to be creation of a standing consist during train makeup, in which the last car is an unloaded car that has failed to couple to the remainder of the consist (Figures 2-2 and 2-3). This loose light car can place one or both of its couplers above sill height by a combination of pitching and vertical motions, after it is struck by the next car to be fed into the consist. The likelihood of such an override event is increased when the striking car is heavy, and head punctures can occur if either or both of the cars adjacent to the loose light car are loaded tank cars.

Fig #'s wrong

A protective shield (Reference 2), as illustrated in Figure 2-5, has been under development since 1970 in an effort to protect the lower half of the head. Switchyard tests (Reference 2) of the shield indicated that the shield does not absorb any significant amount of impact energy, but effectively blunts the coupler thereby distributing the impact energy over a greater area. Hence the tank head is able to survive a higher impact velocity.

3. OBJECTIVE

The objective of this effort is to investigate the critical parameters in tank car impacts, and to develop a methodology for small scale testing so as to evaluate existing and improved tank car designs. The effects of such things as tank car design, type lading, environmental conditions, impact speed, etc. are considered.

4. RESULTS OF LITERATURE SEARCH

The following contains the results of an extensive literature search on the general subject of tank car impacts, model tests, analysis methods, etc. Other than the literature referred herein, those relevant to the topic other than the referred ones are also included in the reference.

4.1 Failure Modes of Tank Car Heads

The heads of tank cars are generally ellipsoidal in shape with a nominal diameter of 120 inches and thicknesses of the order of 7/16 inch to 15/16 inch thick depending on whether or not they are pressurized. The commonly used material of TC-128-B steel is ductile under normal conditions. However, some TC-128-B steels are brittle at lower temperatures encountered in railroad revenue service. Shang and Everett (Reference 3) discuss vulnerability of these type tank car heads in impacts. The tank car heads are either indented or punctured by couplers, side sill, end sill or other flying objects during derailments. They investigated the effect of internal pressure, lading, and percent outage on the indentation, and puncture process.

Indentation is generally defined as the permanent deformation created by the force of the coupler without producing puncture into the tank car head. For a large head, the indentation can be of the order of 10 inches or more if the car is unpressurized and is not completely full (some outage). On the other hand, puncture implies complete piercing by the projectile. Indentation will generally be produced by a projectile such as a coupler at low or intermediate impact velocity. Since the head thickness is generally less than one inch for an unpressurized head, the impact energy will be absorbed by relatively large elastic and plastic behavior of the head as illustrated in Figure 4-1(a). The application of internal pressure (Figure 4-1(b)) can provide considerable resistance to the overall deformation of the tank head. As Shang points out, the deformation is much more localized and the tank head would fail with less overall tank head deformation and at a lower energy level as compared with the case of non-pressurized tank heads. Complete filling of the tank with lading (no outage) has the same effect as increased pressure. The primary mode of failure is

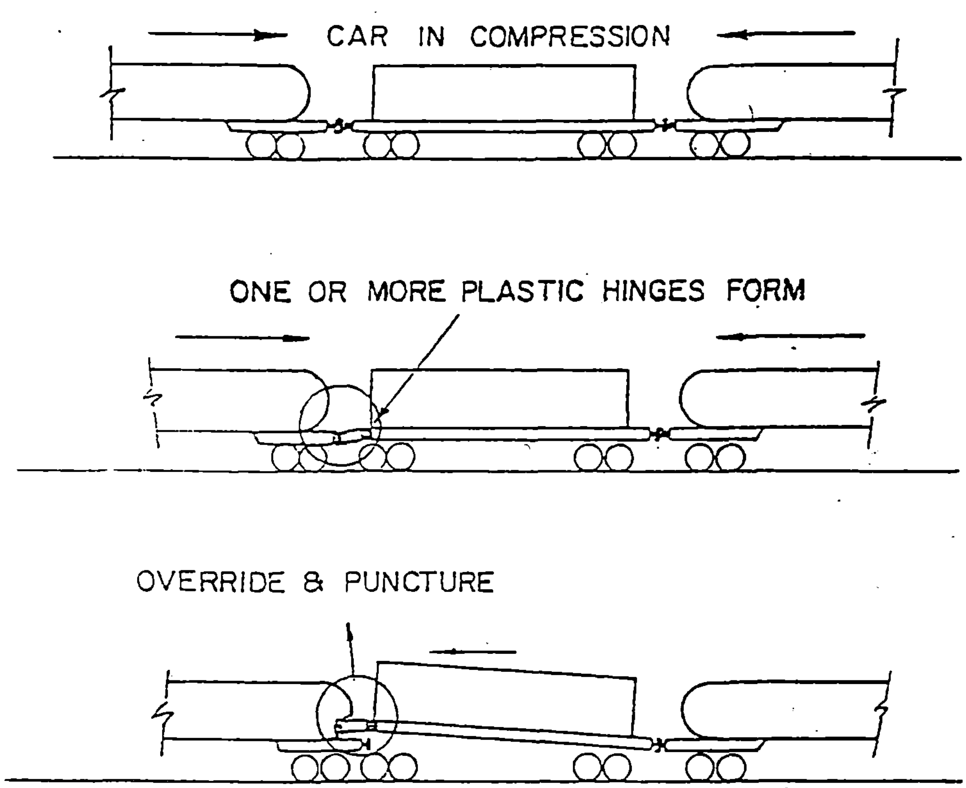


Figure 1-1 Dynamic squeeze mechanism

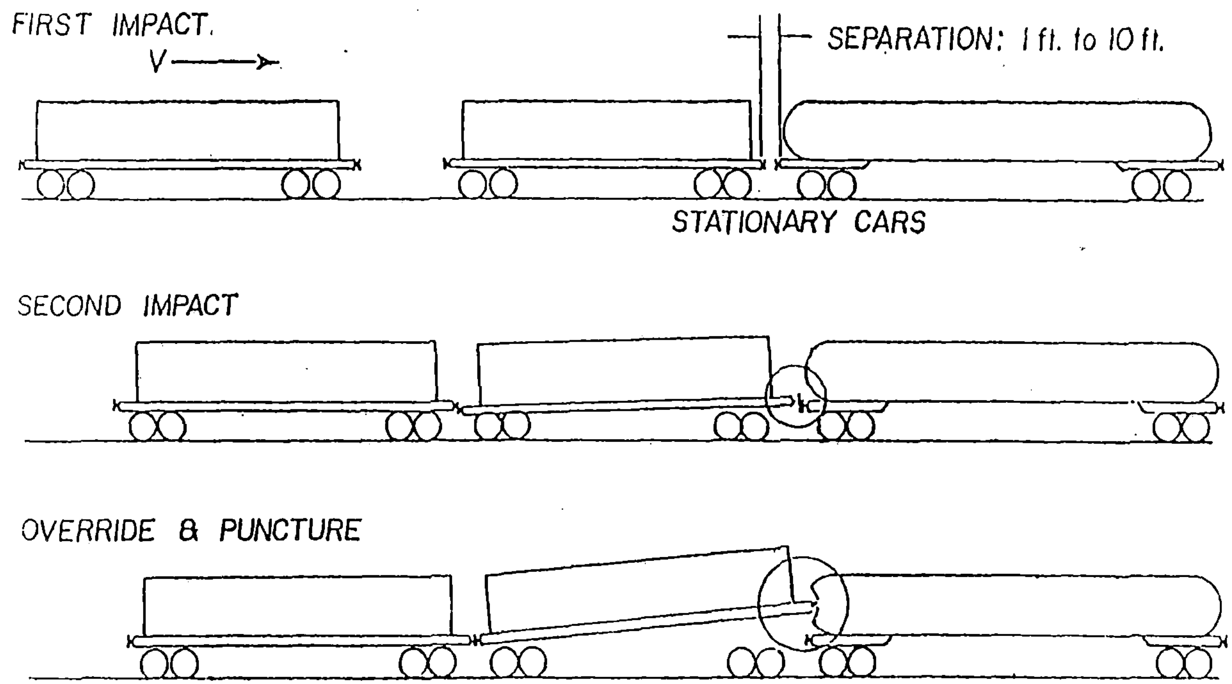


Figure 1-2 Double impact override mechanism

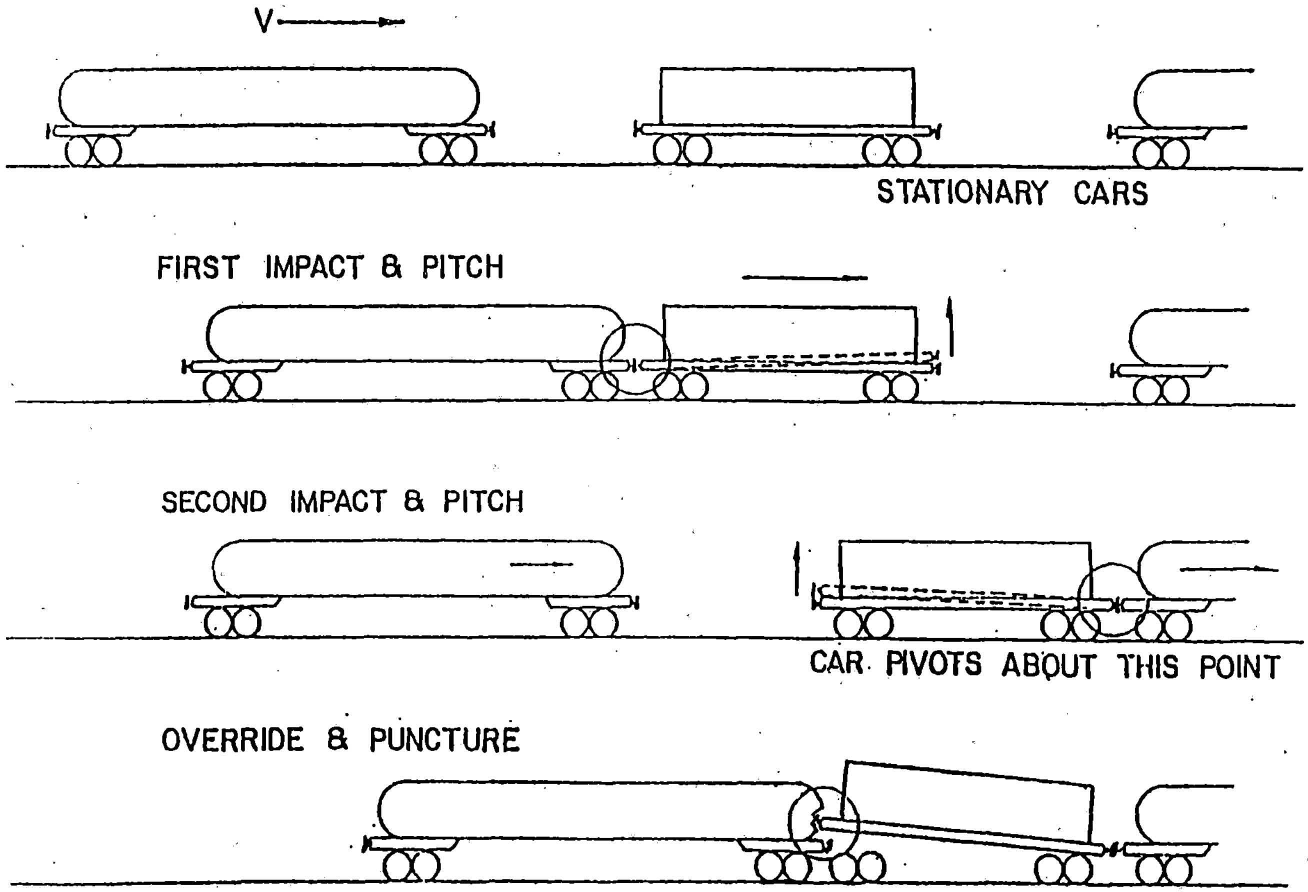


Figure 1-3 Triple impact override mechanism

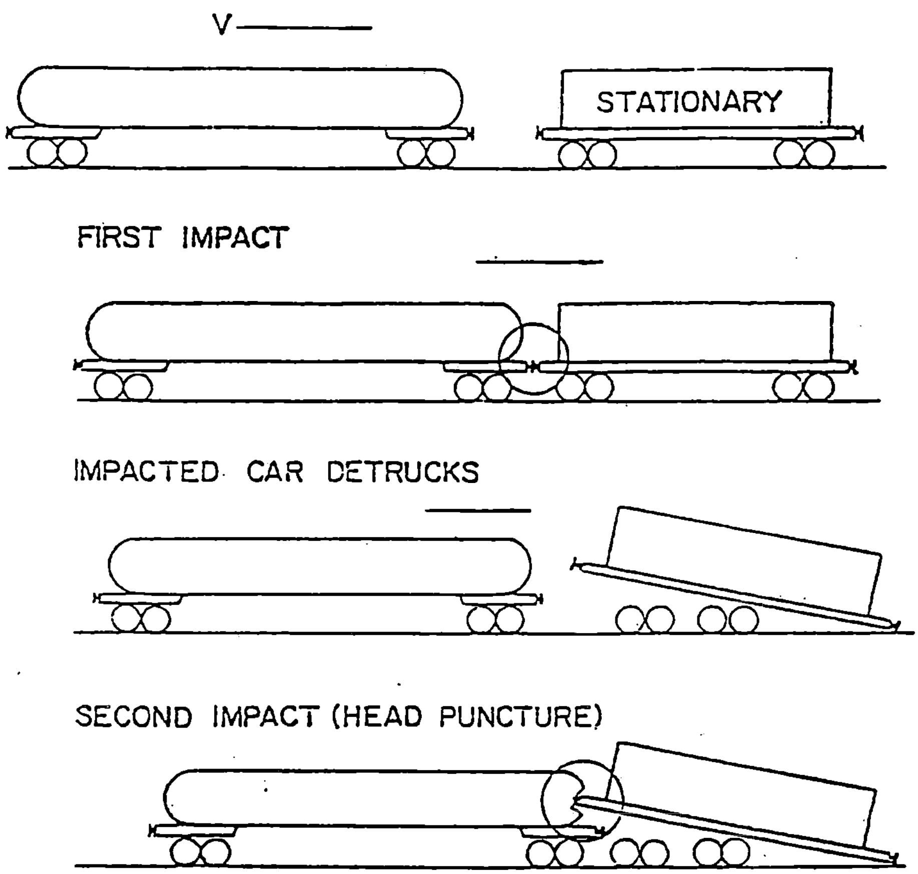


Figure 1-4 Detruck or derail override mechanism: theory

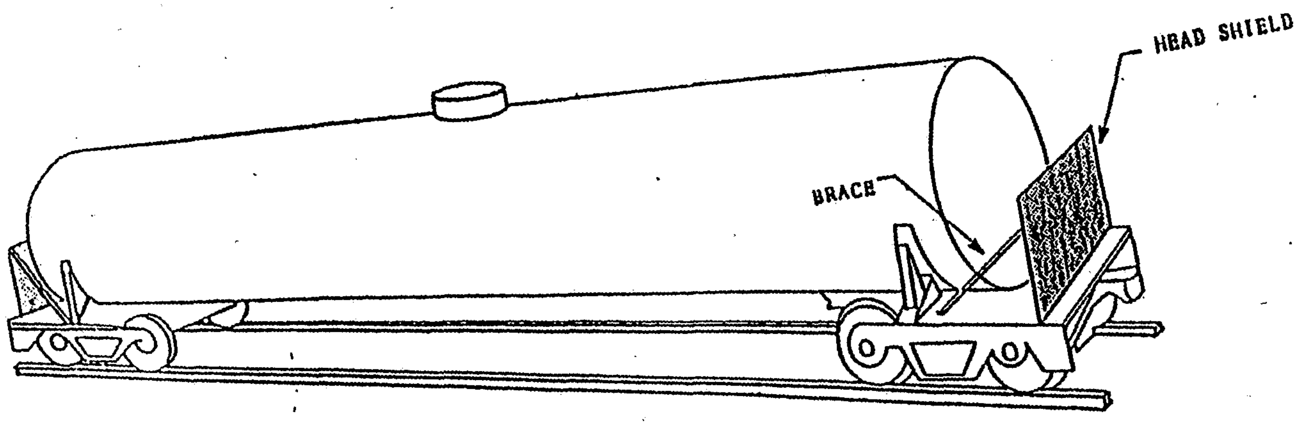


Figure 1-5 Head shield protection concept

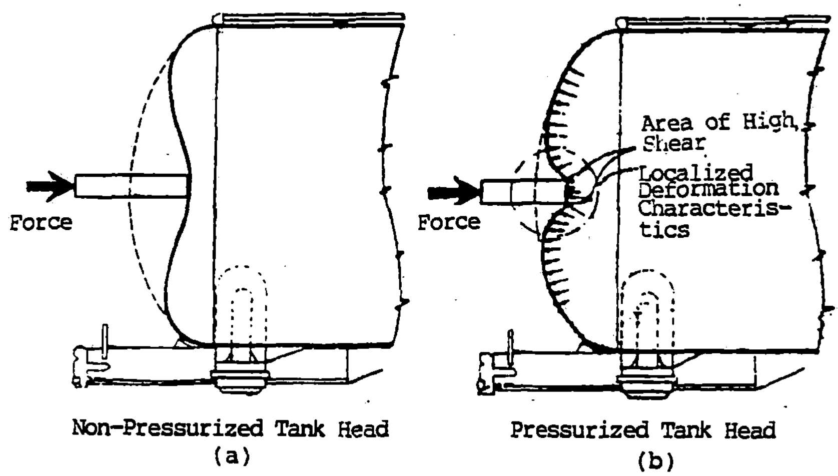


Figure 4-1 Schematic views of tank head deformations

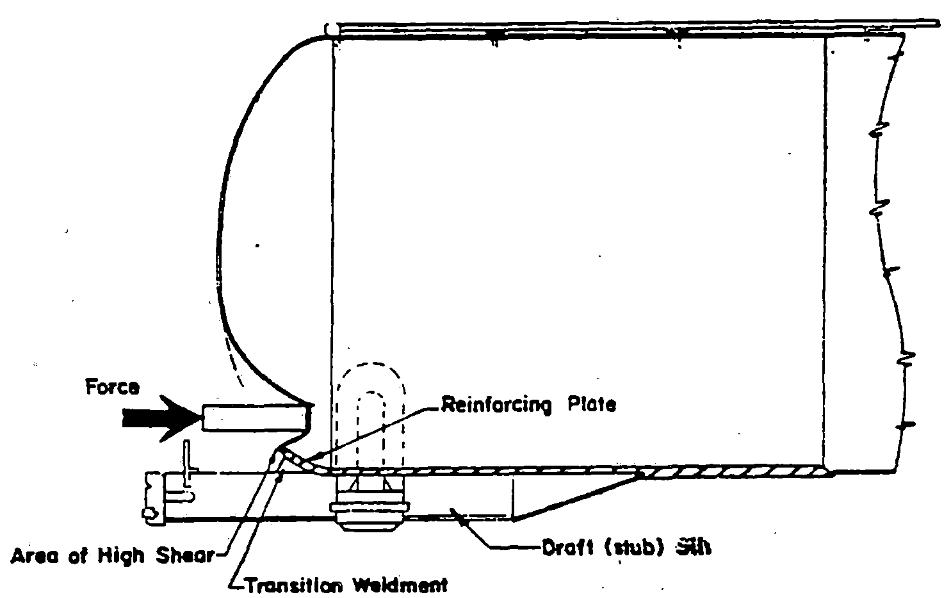


Figure 4-2 Schematic view of knuckle deformation

attributed to the plug formation resulting from the great magnitude of shear force which is applied along the edge of the coupler impact impression.

For the case of impacts near the knuckle in the head, the deformation characteristics are substantiatly different from the case of central impact as illustrated in Figure 4-2. Generally the knuckle area is reinforced and considerably less deformation occurs before ruptures. The energy absorbed is therefore somewhat less.

The previously discussed failure modes have been confirmed in a number of small scale and full scale tests. The Association of American Railroads (AAR) in cooperation with Railroad Progress Institute (RPI) sponsored a series of head car impacts at 1/12 model scale (Reference 4). The objective of this test program was to gain preliminary insight to the phenomena of railroad tank car head punctures as caused by adjacent cars. These tests involved dropping a 1/12 scale coupler backed up by various weights from various heights on a 1/12 scale models of 2:1 ellipsoidal heads. A deficiency in the tests was the inability to offset gravity and hence the inability to accurately scale the results up to full scale. However, they did provide useful information on failure modes. All heads were impacted at their centers.

With the 1/12 scale factor, the model heads simulated a 120" diameter x .718 inch thick head as a full scale car. Curves relating kinetic energy absorbed and dent depth are given in Figure 4-3. The results verified that:

. An empty non-pressurized tank head is susceptible to deeper dents than one backed up with liquid and pressurized.

. An empty non-pressurized tank head is less susceptible to puncture than one backed up with liquid and pressurized.

. A tank head backed up by liquid and a pressurized vapor space is less susceptible to puncture than a head backed up by liquid only (no vapor space) and pressurized.

Appendix C of Reference 4 summarizes impact tests on full scale riveted heads by General American Transportation Corporation's test center. The ram car was a 10,000 gallon car. The striking coupler was instrumented with a dynamometer (type E) to measure the striking force. Some tests were made using a 1/2" thick head shield. Some general observations quoted from Appendix C as regards to failure modes are as follows:

. The primary mode of failure is direct shear, initiating at the top and bottom edges of the impression made by the coupler knuckle. Blunting of the knuckle contour at these edges would decrease the probability of puncture by a small, though unknown, extent. In cases where this shear progressed far enough (puncture in the limiting case), a secondary mode of tension failure occurred around the periphery of the striking coupler outline.

. An empty or filled tank becomes more susceptible to puncture with an increase in tank pressure.

. Puncture vulnerability increases as the impact point moves from the center of the head toward the lower knuckle region.

. An empty non-pressurized tank is more susceptible to puncture than a loaded non-pressurized tank when struck in the lower knuckle region.

. A shell full (no outage) pressurized tank is much more susceptible to puncture than one containing 2% outage at the same pressure.

. A jacket head of 1/8" thick steel four (4) inches away from the head and filled with glass wool is an unacceptable head shield.

. An acceptable amount of damage (no lading loss) occurs at an impact speed of 10.5 mph when the tank is loaded to 2% outage, pressurized to 40 psig, backed up by three cars, and struck 1/3 of the way down from the center of the head. The threshold puncture speed is about 12.5 mph for these conditions.

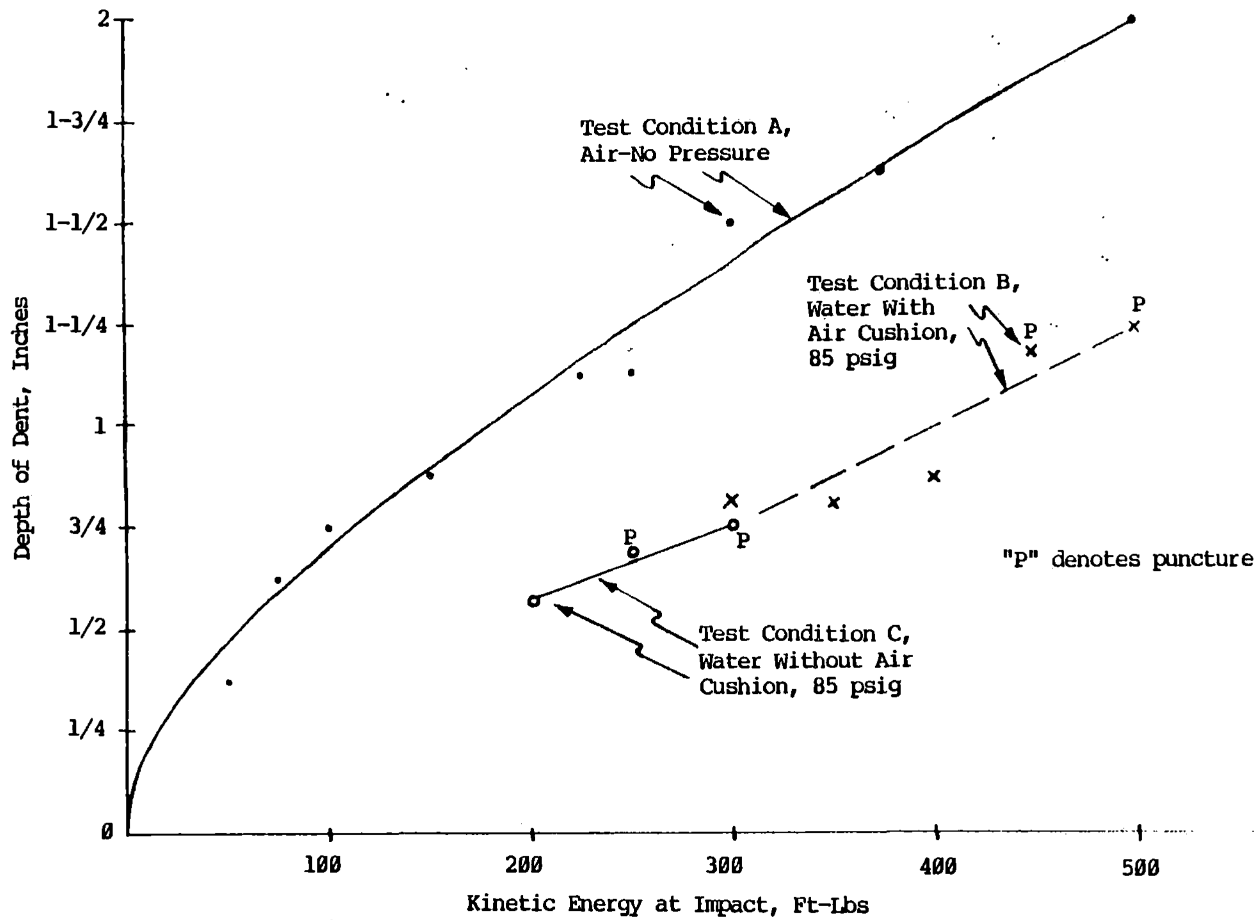


Figure 4-3 Kinetic energy vs. dent depth 1/12 scale impact tests

. For the conditions described above, a 1/2" head shield successfully protects the head from puncture at a 17 mph impact speed.

Tests at MIT by James J. Gorman (Reference 5) using a 12 ounce "Coke can" as a model were conducted to examine such things as penetrator shape, internal pressure, lading etc. on failure modes. A four wire pendulum was constructed to deliver the blow to the end of the can held in a horizontal position. The results indicated that:

. The character of the failure was greatly dependent upon the indenter (impactor) shape. In the case of a sharp edged indenter, failure was due to the shearing of a plug with the same diameter as the indenter. A rounded indenter produced tensile failures at the head of the bulge formed in the can end. The effect of indenter shape on puncture energy appear to vary with rate of loading (strain rate effects). As the rate of loading increased from essentially zero to that in the impact tests, the influence of the indenter shape decreased dramatically.

. The pressure of a nearly incompressible fluid behind the head decreases the puncture energy significantly.

. The change in puncture energy with increasing pressure was small in these tests and inconsistent.

In summary, the failure modes observed in both the model tests and full scale tests appear to give consistent qualitative results. Hence, except possibly for strain rate effects, the failure modes between the model and full scale should be quite similar. Strain rate effects are constitutive behavior and failure criteria will be considered in the methodology developed in cases where it is expected to be important.

4.2 Quantitative Estimates of Impact Parameters

This section deals with quantitative estimates of impact parameters that tank cars are subjected to in derailments and switchyard operations.

4.2.1 Impact Velocity

The detrimental effects of tank car impacts are strongly dependent on impact velocity since in a very simple analytical model the kinetic energy of the impacting car depends on velocity squared.

A study of switchyard impacts (Reference 1) indicated a probability of impact speeds as shown in Figure 4-4. While the average impact speed is only 4.7 mph, an effective car design should protect against higher speeds which might be expected. Head puncture occurrances in classification yards are usually caused by overspeed impact. The maximum safe speed, below which coupler override (and therefore head puncture) is unlikely to occur is approximately 8 mph.

Orringer and Tong (Reference 2) state that override and tank car head impact can occur at impact speeds as low as 12 mph. Occurrances of 112A/114A tank car head punctures at impact speeds as low as 4 mph in switchyards have been reported (Reference 6). Occurrances at such low speeds apparently involved kinematic situations, and the reported occurrances may have involved effects of low temperature on the ductility of tank head materials.

Full scale tests of a new 112A340W car conducted by General American Transportation Corporation's test center indicated a threshold of puncture occurrance at a speed of between 9.3 and 12.7 mph (Reference 4 Appendix F page F-3). The ram car consisted of two cars coupled together with a solid draft gear to simulate a heavy single car (350,000 pounds total). The test car weighed 337,400 pounds. The striking location was 24 inches above the top of the sill. An acceptable amount of damage occurred at 15.5 mph when the tank head was protected by a 1/2 inch steel head shield.

Based on the results of these tests, it appears that a model test facility should be capable of simulating impact velocities of up to 20 mph.

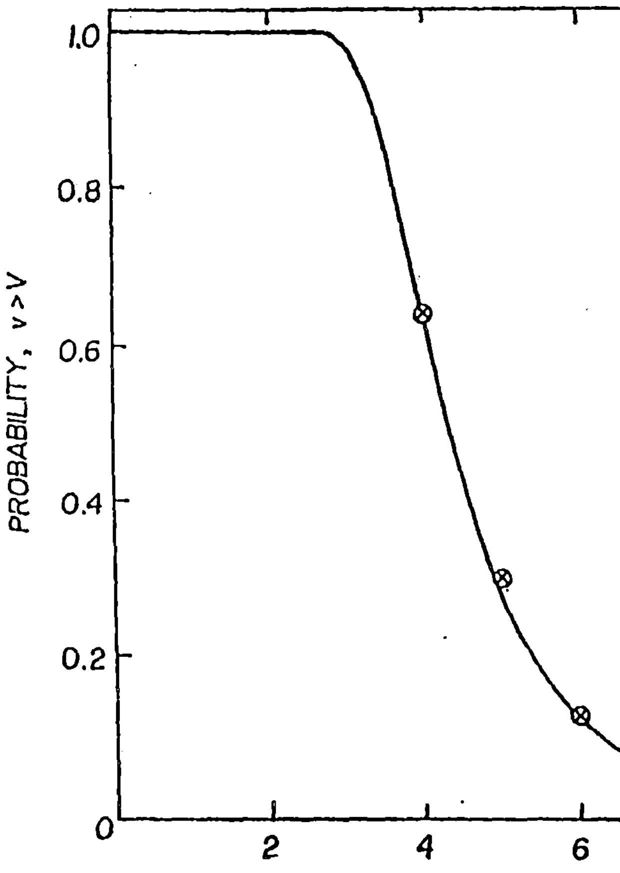
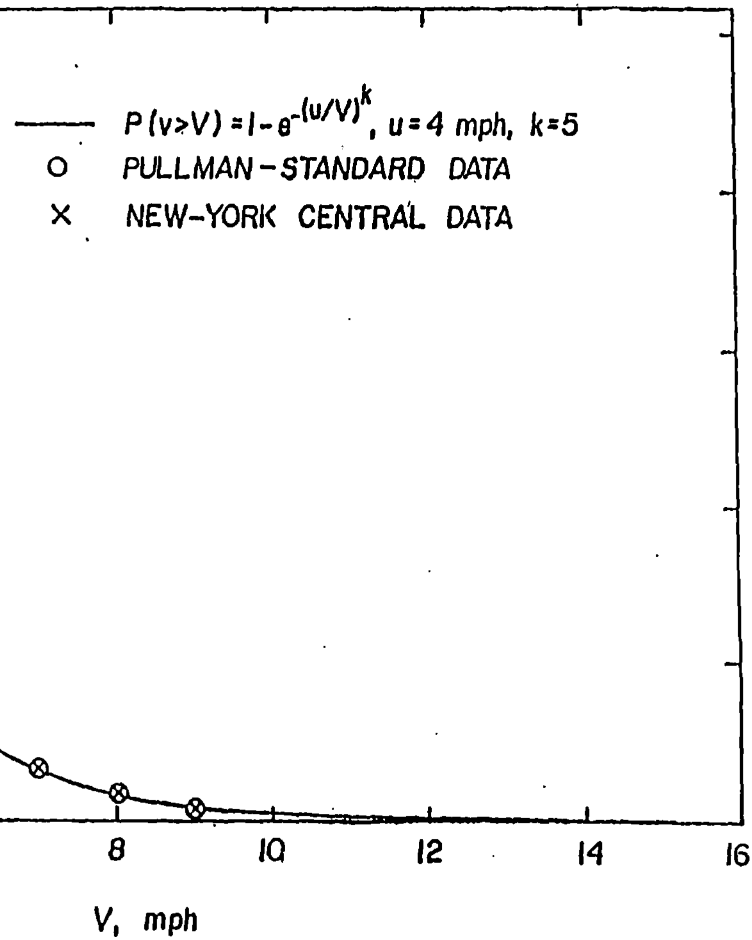


Figure 4-4



Probability of impact speeds

4.2.2 Energy Available for Tank Car Structural Damage

One of the critical parameters in the collision of tank cars is the amount of energy available for head puncture. The energy available for tank car structural damage is highly dependent on relative impact velocity and the collision scenario. Considerable work (for example Reference 1) has already been done to estimate the dynamics of tank car collision and the energy available for head puncture. A preliminary "quick look" indicates that in a single impact of a 273,000 pound car at 20 mph, the maximum possible energy available is 3,650,000 ft-lbs. Considering a momentum balance and loss of energy in structural deformation, it is reasonable to expect that a 2,500,000 ft-lbs is available for head puncture.

For the collision scenarios involving a loose car and multiple impacts (Figures 1-2, 1-3, 1-4), the available impact energy is highy dependent on car spacing between the light car and the first back up car. Figure 4-5 (from Reference 1) illustrates a total energy balance for various impact speeds. The dashed lines give the total energy available for each impact speed. The solid lines give the energy required to override. Therefore the difference in the two curves is the energy available to puncture. If we assume that as little as 1.2×10^6 foot-lbs is necessary to puncture, by substituting this value we obtain the shaded area which is the excess puncture energy. It is apparent that a puncture protecting device such as a head shield or other mitigator should be at least capable of doubling the energy required to puncture the head. Fig #'s

If humping is not limited to a single car, then the problem become much more severe. For example in multiple car humping, tests have shown that only the first 3 cars participate in the override mechanism. Hence for a 20 mph impact, the maximum impacting energy is roughly (3 x 3,650,000) ft-lbs or 11×10^6 ft-lbs. About half of this (5.5×10^6 ft-lbs) is available for head punctures. Figure 4-6 (Reference 1) shows the percentage of head punctures which could be prevented in multiple humping as a function of design impact energy for the head/shield combination. A design value of 5.5×10^6 ft-lbs would prevent all switchyard punctures.

Based on this preliminary quick look, it appears that any small scale testing method should be capable of simulating a full scale energies of about 2.5×10^6 ft-lbs for single car humping and up to 5.5×10^6 ft-lbs for puncture during multiple car humping in the switchyard. Hence scaling laws must be developed to appropriately scale these energies down to values achievable within the laboratory.

4.2.3 Impact Force

In order to design a small scale test facility, we must have some estimate of the impact force expected and the impulse. The full scale tests described in Appendix F of Reference 4 indicated threshold velocities for puncture between 9.3 and 12.7 mph for a striking car weight of 348,900 lbs and weight of 3 backup cars of 382,750 lbs. Figure 4-7 shows the measured impact force and car displacements as a function of time. Coupler full scale forces of at least 700,000 pounds can be expected and should be planned for in a test facility.

4.2.4 Tank Car Design Parameters

The design of the tank car head can have a significant effect on its impact resistance. Appendix E of Reference 4 summarizes the results of tests conducted by ACF Industries using 1/5 scale models. The objective of the tests was to study head puncture phenomena on class DOT 112A340W pressure cars, specifically with respect to the influence of internal pressure, head thickness, head material, and steel head shields. Figure 4-8 from Reference 4 summarizes the results of the tests without head shields.

The effect of increasing the strength of the head material is derivable from the results. From Figure 4-8 the head puncture threshold speeds for materials of different strengths, with parameters of thickness and pressure held constant, are:

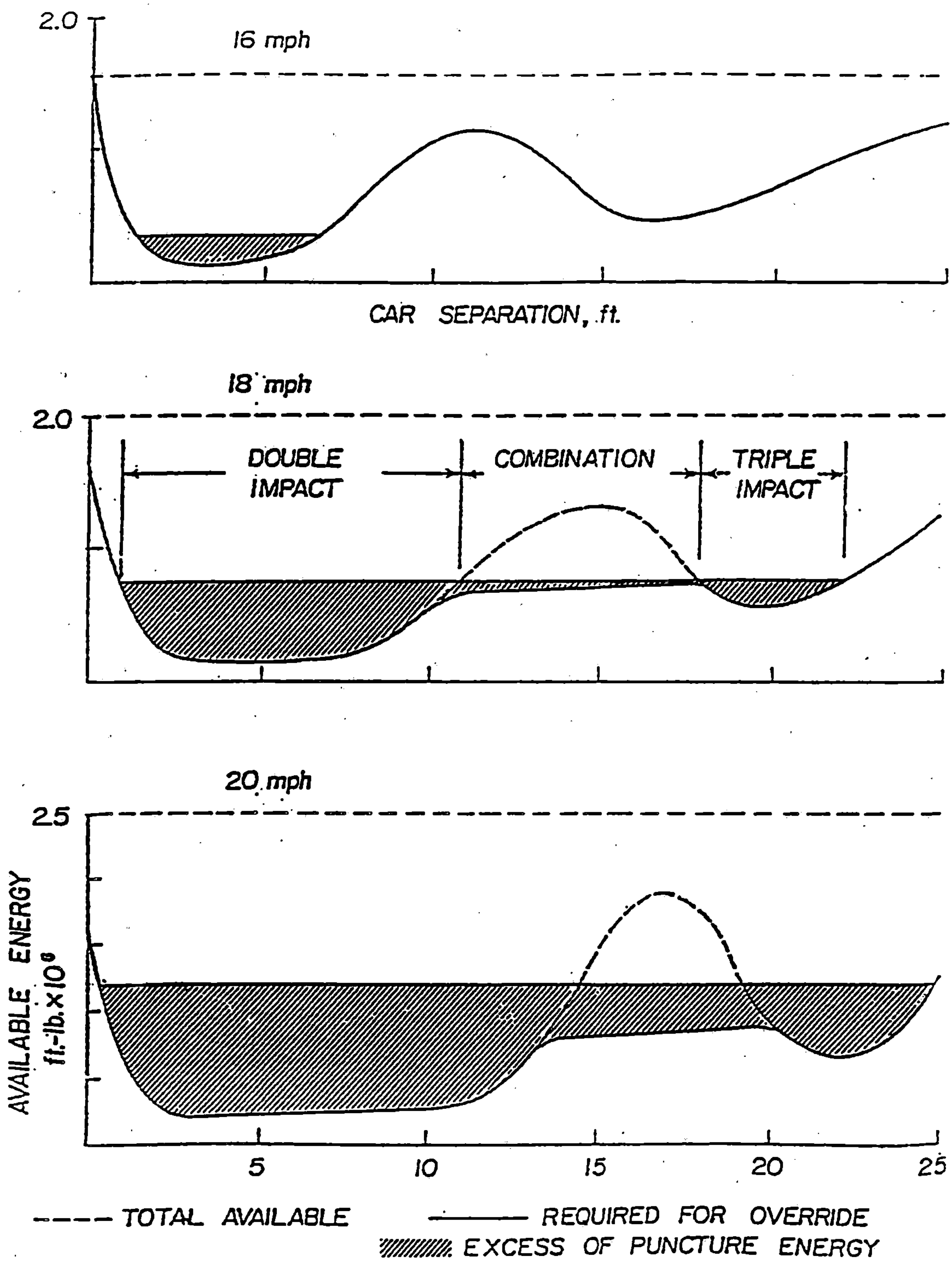


Figure 4-5 Relation between impact velocity, car spacing and absorbed energy

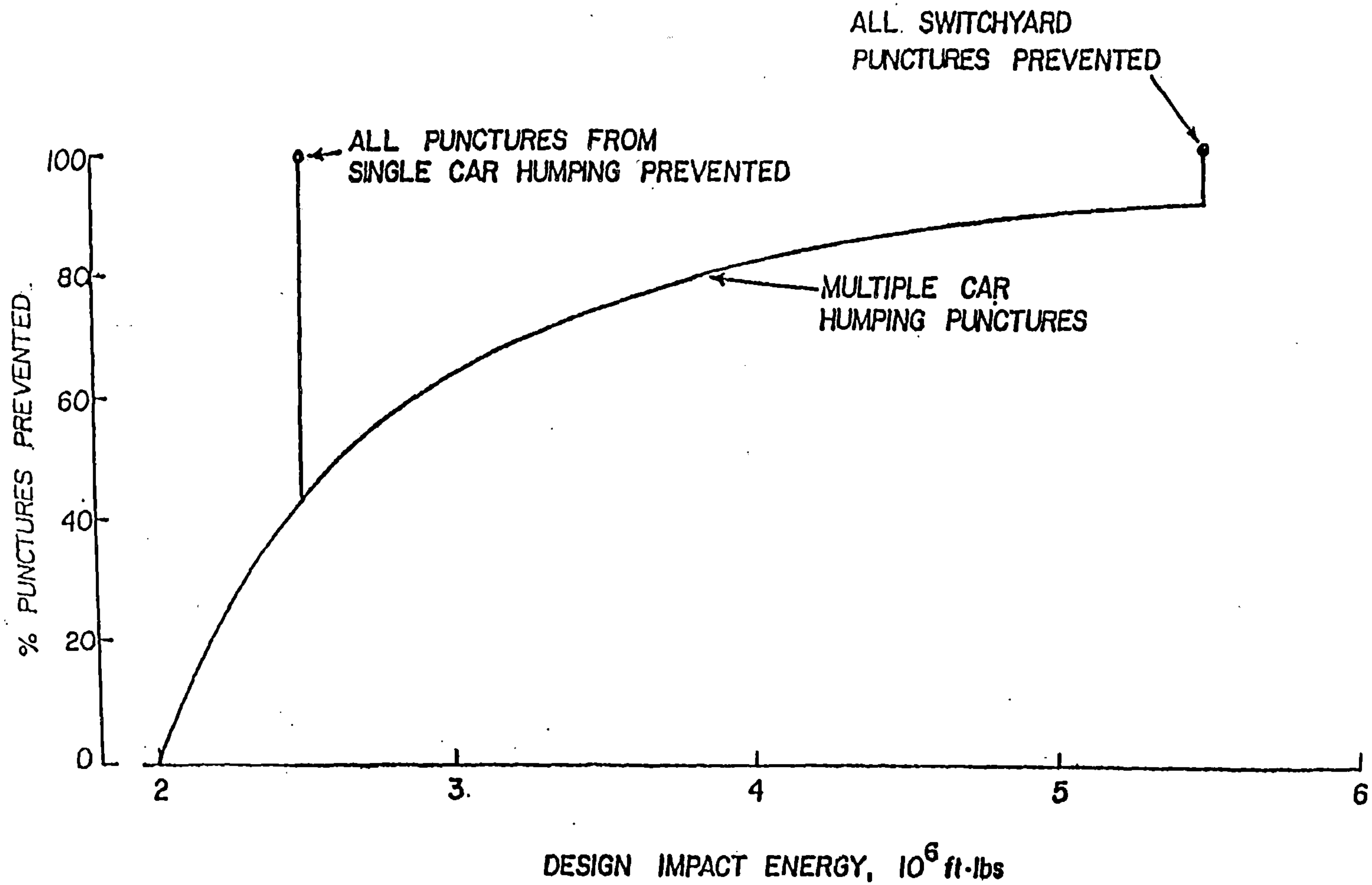


Figure 4-6 Design impact energy

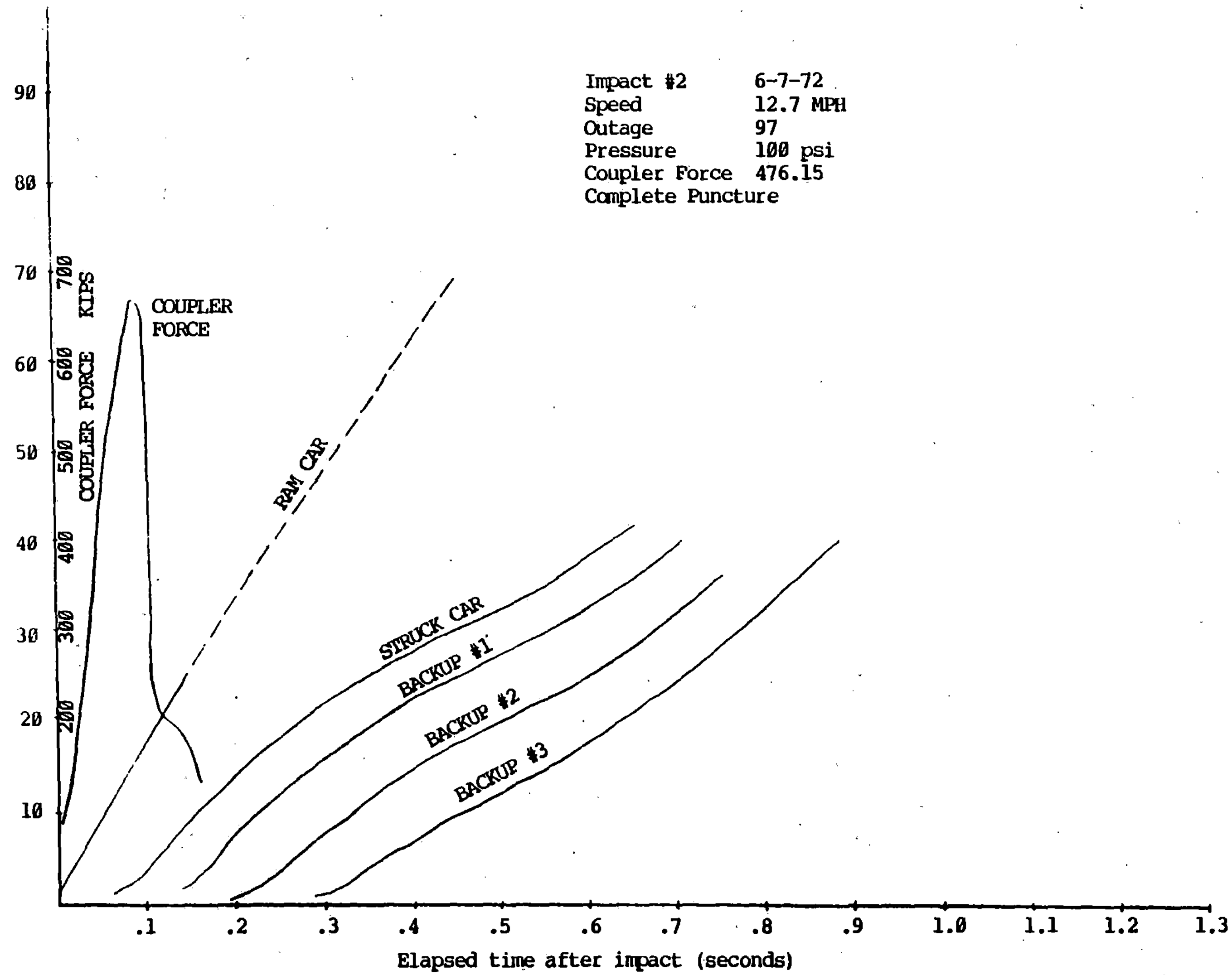


Figure 4-7 Coupler force and car displacements, impact #2 (no head shield, puncture)

Material	Tensile Strength	Puncture Threshold Speed	
TC-128-B*	76,200 psi	10.0 mph	0.188" thick heads
A-285-C	62,200	8.6 mph	tested at 100 psig
TC-128-B*	76,200	8.1 mph	0.152" thick heads
A-515-70	71,700	7.2 mph	tested at 100 psig

In the first case (0.188" thick heads) the threshold puncture speed is raised 1 mph per 10,000 psi increase in material tensile strength. The trend is more in the second case. ?

It can also be observed from Figure 4-8 that the effect of internal pressure is to decrease the threshold velocity for head puncture.

The effect of head thickness can best be seen by comparing the threshold velocity for puncture for TC-128-B material at 100 psi internal pressure at various head thicknesses as shown in Figure 4-9. Based on a very limited database the threshold velocity for puncture goes up very nearly in a linear manner with head thickness.

Shang (Appendix G of Reference 4) performed a rather extensive sensitivity study of tank car parameters based on empirically derived equations.

It has been shown that several parameters influence the vulnerability of unprotected tank heads. Some are extremely important, some are not. To assess their relative importance, the study was conducted with reference to a tank car under the following prescribed conditions:

Weight of impacting car, W_1 = 263,000 lbs

Weight of struck cars $W_2 = 4W_1$; $\alpha = W_2/W_1 = 4$

Tank radius, R = 60 inches

Tank head thickness, t = 11/16 in.

Ultimate shear strength, τ_u = 60.2 ksi

Internal pressure, P_i = 100 psi

Outage = 2%

sign?

The corresponding puncture velocity was calculated to be

V = 12.2.

With reference to the above prescribed conditions, a parametric sensitivity comparison can be made on the basis of the percentages of increase (or decrease) in a particular parameter needed to increase the puncture velocity by a given amount while the others remain unchanged. The parameters examined are: Tank thickness t, tank material τ_u, weight of impacting car W_1, weight ratio α, and internal pressure P_i. The results are shown in Figure 4-10. It is observed that the reduction in the weight of the striking car is the most influential parameter in increasing the puncture velocity. In contrast, the weight ratio of impacting car vs. struck car is the least influential parameter.

4.2.5 Temperature

Ambient temperature can have a devastating effect on the impact resistance of some steel materials and hence the structures using these materials. A test commonly used to evaluate the impact resistance of materials is the charpy impact test which determines the amount of energy that a small specimen can withstand as a function of

* These two materials were nominally TC-128-B, although they did not meet the tensile strength requirements of that specification.

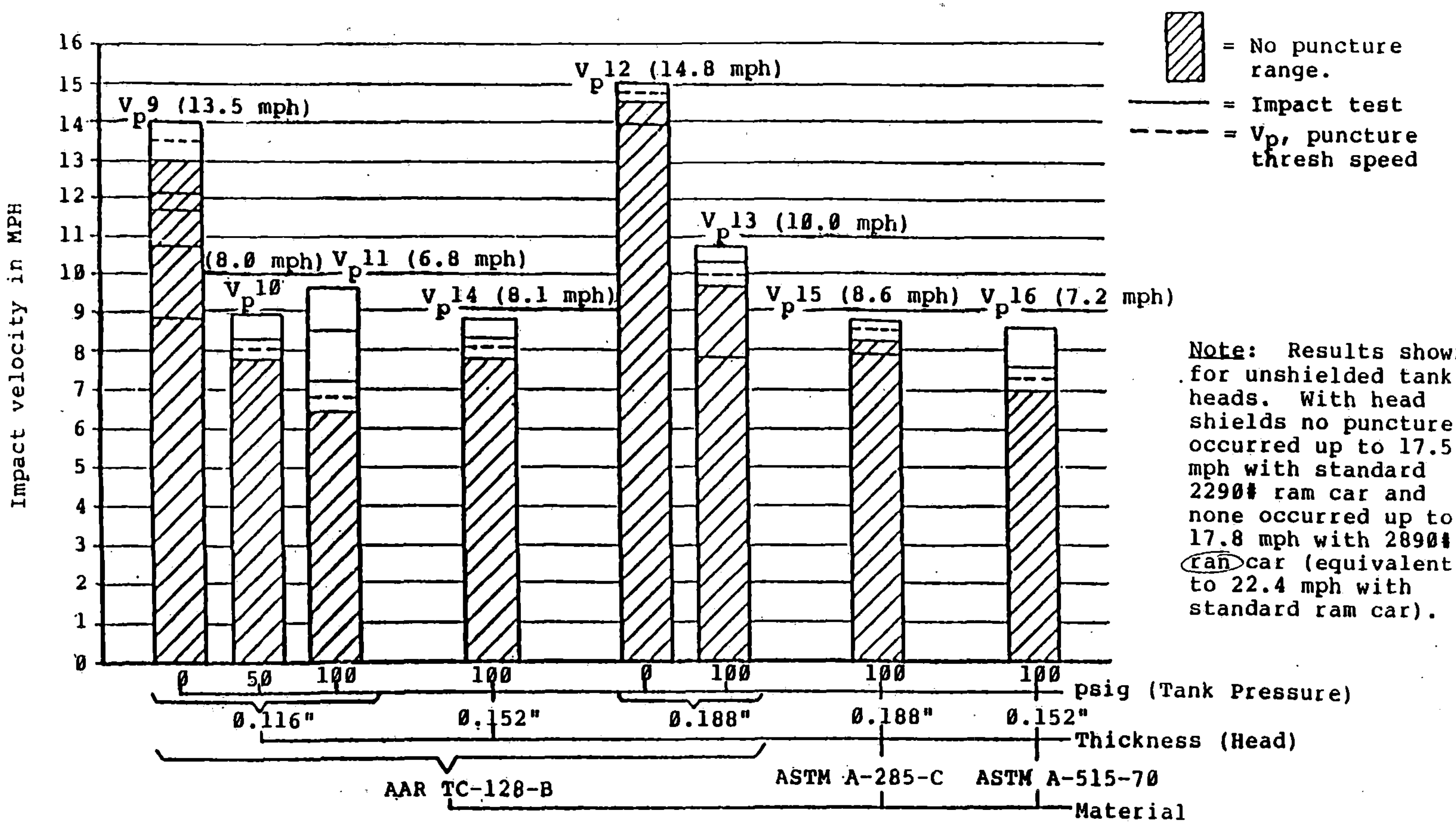


Figure 4-8 Summary of all tests without head shields

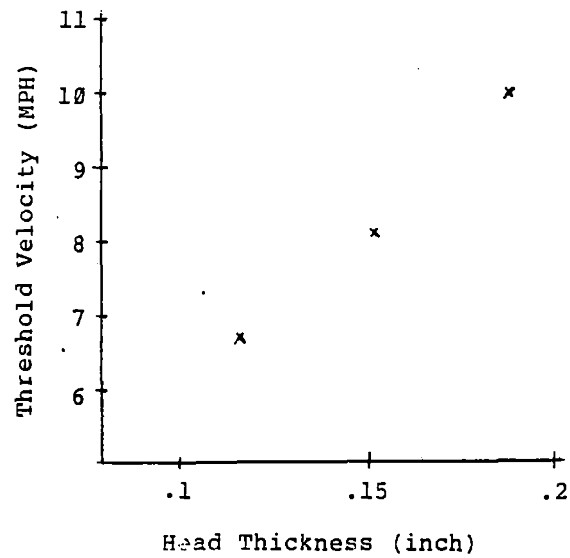
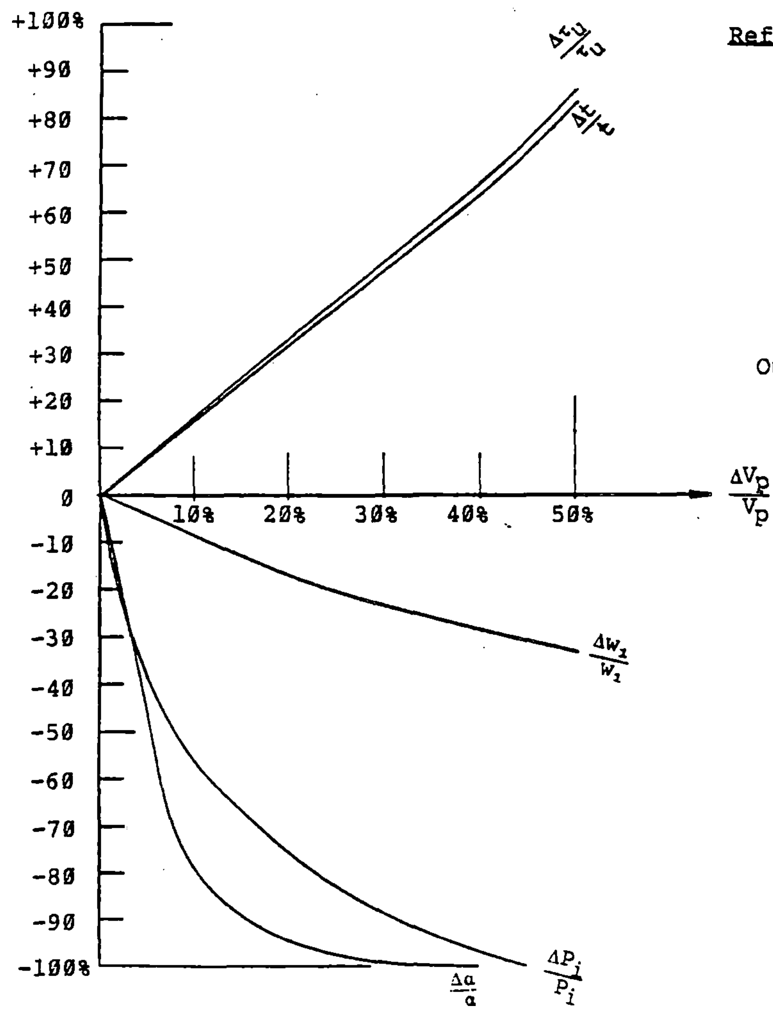


Figure 4-9 Threshold velocity for puncture vs. head thickness



<u>Reference Tank Car Condition</u>

w_1 = 263,000 lbs.

$\alpha = w_1/w_2 = 4$

R = 60 inches

t = 11/16 inch

P_i = 100 psi

τ_u = 60.2 Ksi

Outage = 2%

V_p = 12.2 mph

Figure 4-10 Parametric sensitivity evaluation temperature. Extensive testing was conducted by the National Bureau of Standards (Reference 7) on commonly used tank car materials such as ASTM A212 steel and TC 128 steel. The charpy specimens were standard size of .394 x .394 x 2.165 inches. Tests were conducted at various orientations; L, Longitudinal direction is the principal rolling direction; T, transverse direction and S, short transverse direction. Figure 4-11 shows typical data on the energy absorbed as a function temperature for TC-128-B steel specimens (see Figure 2B in Reference 7). Note that there is a transition from ductile (high energy absorptive) to brittle (low energy absorptive) in the temperature of about 100°F to -100°F (see Reference 7). Hence cold temperature can significantly reduce the energy absorptive capability of a tank car head.

4.3 Effects of Impacts On Structures

The most detrimental effect of head impact is the potential loss of lading material which in many cases could be hazardous. Small scale test facilities offer the potential for establishing threshold velocities for puncture since a large number of tests are frequently required to establish the threshold velocity. These facilities could also prove useful in doing tests to estabish permissible depth of dent above which the tank head must be replaced.

4.4 Analysis Methods

4.4.1 Semi-Theoretical Method of Shang

A review of the literature indicated that Shang (Reference 3) has done the most extensive work in developing analytical models for assessing the vulnerability of tank car heads during impact. His method is based on using the form of a Hertz theory for the force indentation process given by:

$$F = \alpha (d)^{1.5}$$

where F = contact force

d = indentation

α = constant

The constant α depends on the geometrical and physical properties of the tank head and the magnitude of internal pressure, and is determined experimentally. Based on analysis of considerable data, Shang arrived at the following empirical results:

* Estimation of Maximum Coupler Impact Force During Impact

Using the data obtained from full scale and reduced scale tests, the formula for computing the maximum coupler force developed during impact was slightly revised to yield

$$F = 0.00383\ \alpha^{3/32}\ (W_1 V)^{1.5}\ \lambda \qquad (4\text{-}1)$$

where

F = estimated maximum coupler force generated during impact, kips

$\alpha = W_2/W_1$

W_1 = weight of striking car(s), in 1,000 lbs

W_2 = weight of struck car(s), in 1,000 lbs

V = velocity of striking car(s) prior to impact, mph

λ = internal pressure parameter shown in Figure G-1 from Reference 3.

* Prediction of Threshold Puncture Velocity

After equating the coupler force to the puncture resistance of the tank head, the following equation is established for predicting the threshold puncture velocity of unprotected tank car heads subjected to coupler knuckle impacts:

CVN IMPACT TEST RESULTS FOR FOUR ORIENTATIONS OF AAR TC128-B STEEL SPECIMENS—
CALLAB SHELL PLATE.

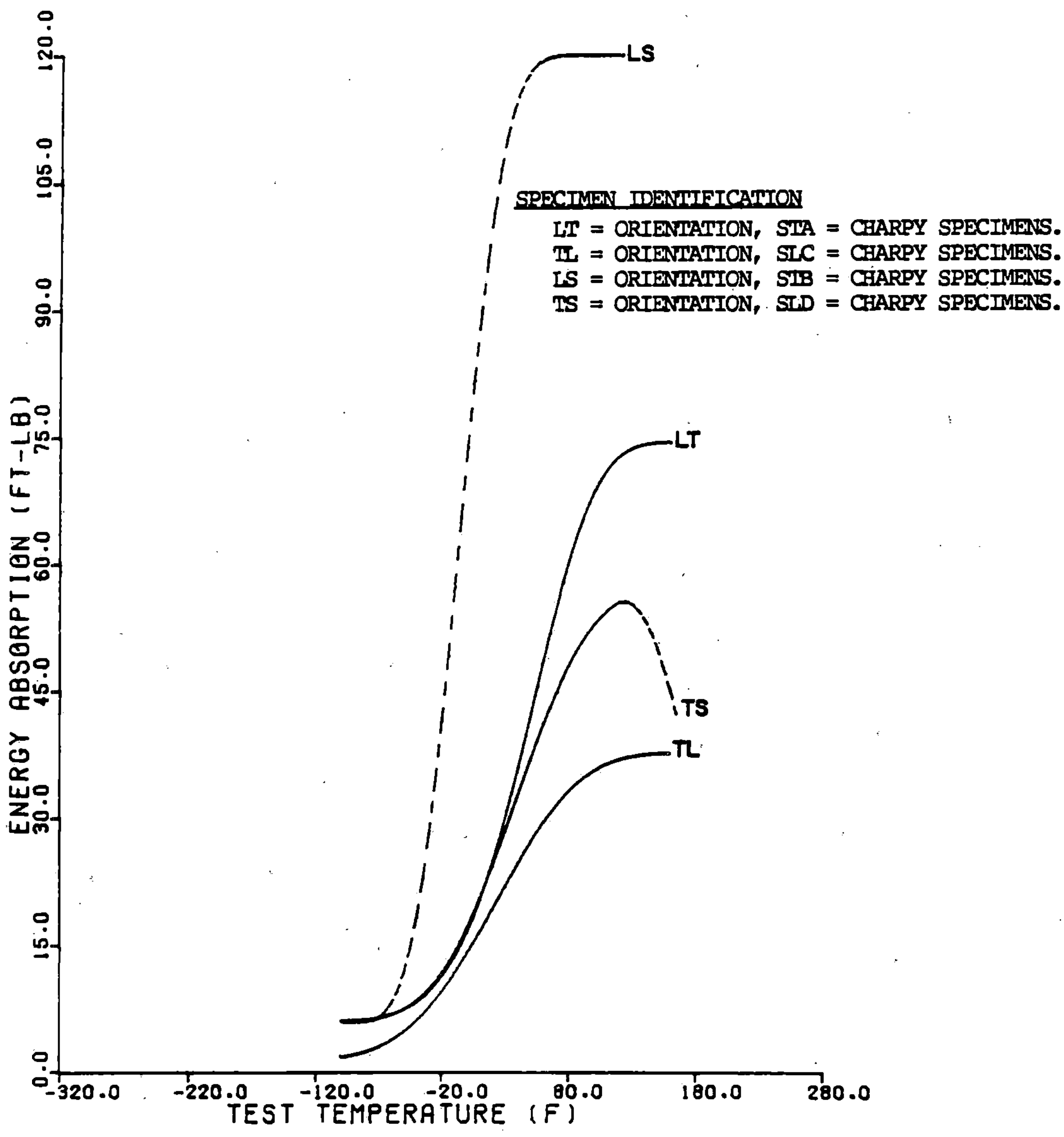


Figure 4-11 Energy absorption vs. test temperature

$$V_p = \frac{27.6}{W\ (\alpha)^{1/16}} \left[\frac{\tau_u Rt}{\lambda}\right]^{1/1.5} \qquad (4\text{-}2)$$

where

V = threshold puncture velocity, mph

τ_u = ultimate shear strength of head material, ksi

R = tank radius, inches

t = tank thickness, inches

Comparisons of predictions with experimental data appears to be reasonably good. Figure 4-12 shows computed impact velocity for puncture as function of internal pressure for model tests of class 112A cars. The threshold puncture velocity zone is defined with reference to the value computed from Equation 4-2 to include +10% variation in the influential parameters of radius, thickness, material property, internal pressure, car weight, or any combination of these parameters. These comparisons indicate that the procedures developed for predicting the threshold impact velocity and the associated coupler force are reasonably accurate except as applied to a fully loaded car. For a fully loaded car, the method predicts a much higher threshold velocity than actually measured.

No attempt was made to develop a theoretical basis for predicting the puncture characteristics of fully loaded cars and to correlate such theory with the test data. This was because only one car was tested under such a condition; and further, such a condition is not common in transportation.

It appears extremely fortuitous that such a simple model is able to give reasonable predictions; particularly when such things as strain rate effects, impact location and lading condition are not present as variables. More exact finite element methods should be capable of eliminating these deficiencies.

4.4.2 Energy Method for Shock Mitigators

Yang (References 8, 9) conducted studies using highly deformable materials (shock mitigators) to protect brittle materials during impact. His results indicated that the combined application of the dynamic field equation of solids together with Hertz's law gave reasonable results for the impact problem of plates and shells. It is believed that the method can be extended to predict the protection that a shock mitigator can give to a tank car head.

This method was developed for the design of shock mitigators for glass submercibles. Glass, being very strong in compression makes an ideal pressure hull, but being very brittle is very susceptible to impact loads. A 2.0 inch shock mitigator of Adiprene was demonstrated to be a very good shock mitigator for an eight foot diameter glass spherical hull. The same basic theoretical method with slight modification can be used to assess the effectiveness of mitigation in preventing tank car head punctures. Hertz theory is selected in the analysis since it deals with the problem of predicting the load distribution during initial point contact between the impacting bodies. Classical Hertz theory has been modified to account for inertia effects and large plastic deformation of the shock mitigating material. In effect, the theory developed is based on an energy balance requiring that the work done by the shock mitigator be equal to the kinetic energy of the system. The loading on the tank car shell head can be obtained by solving the non-linear equation of motion in conjunction with the material constitutive equation (stress versus strain, strain rate, etc.). The equations can be developed in dimensionless form to determine appropriate scaling laws. Once the loading has been determined, the stress in the shell head can be determined using classical shell theory. The method can be modified to account for the energy absorbed by the shell head as well as that absorbed by the protective shield and the shock mitigator.

4.4.3 Finite-Element Methods

Although the literature does not have extensive results of the use of finite-element methods for the analysis of the tank car impact problem, finite element methods do offer strong potential. HONDO, a finite element elastic-plastic transient

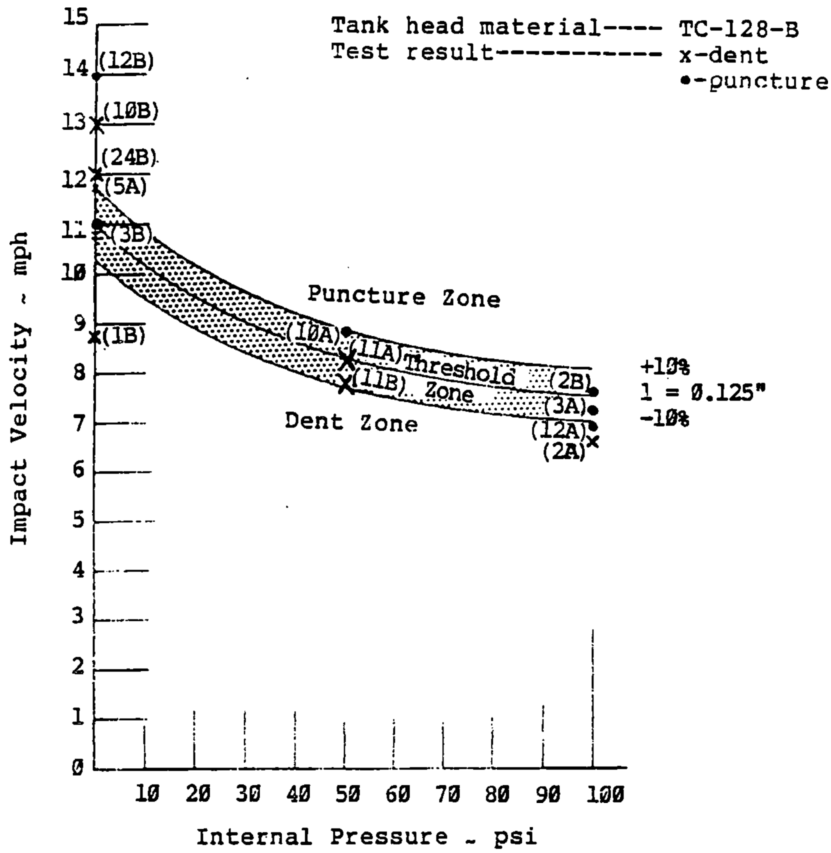


Figure 4-12 Impact velocity vs. internal pressure (for model test N 27Ø54 of class 112A cars)

code has been used for some impact problems but may not be suited for the tank car impact problem because of excessive run times. Finite element codes such as STAGS, DOASIS which are elastic-plastic codes offer potential for computing the static force deflection curves and hence energy absorptive capability of the head. The force-deflection curve can then be used in the equations of motion for the impacting cars to calculate force-time pulse and the indentation-time curve. An iterative process will be needed to include strain rate effects.

4.4.4 Collision Dynamics

Analysis of switchyard impact tests and other rail vehicle collision test results have been conducted with complicated dynamic models that attempt to represent the detailed behavior of each vehicle and coupling at the component level (References 10, 11, 12, 13, 14). Interpretation of test results with such models tends to emphasize adjustments of the many input parameters to match observed behavioral details (e.g. position-time, force-time). Orringer, et al (Reference 2), adopts an approach that restricts analysis to fundamental physical models that allow attention to be focused on model validity and physical interpretation of the test data. This approach leads to an empirical estimate of energy absorption from the test data, rather than a prediction from component properties. The authors analyze the collision-process using one dimensional analysis and bound the problem by considering perfectly elastic collision and completely inelastic collision. A model is also developed to account for sloshing of the fluid. The model was then extended to cover two dimensional behavior during collision. Some general conclusions using these models to analyze switchyard impacts are as follows:

* The presence of a loose light car in a train makeup increases the chance of override, tank head impact, and puncture in switchyard operation.

* Inelastic energy absorption by mechanical components and kinematic dissipation is not able to prevent tank head punctures. About 70% of the initial energy remain available for the final collision; far more apparently required to puncture a tank head. The authors derive equations to compute energy available for head puncture (Reference 2 page 65).

The energy levels presented in Section 4.2.2 are consistent with those obtained from the mathematical models discussed. As will be seen later, it is believed that the impact process can be adequately simulated by neglecting gravity effects such as in a head on collision. Hence a one dimensional theoretical model should give accurate results in most cases.

4.5 Scaling Laws

In order to develop a methodology for small scale testing it is necessary to develop appropriate scaling laws to scale the model results up to full scale behavior. Scaling laws are generally developed using dimensional analysis. Gorman (Reference 5) developed the scaling parameters using dimensional analysis. Similar dimensional analysis was done in Appendix A of Reference 4 by Professor H. L. Langhaar of the University of Illinois.

For a model study of derailment, the car need be similar to the prototype so that centers of gravity are preserved and moments of inertia are scaled properly. If L_f is the length of the prototype and L_m is the length of the model then we define

$$\lambda = \frac{L_m}{L_f}$$

as the scale factor which can be applied to all geometric dimensions. If the collision process is controlled by gravity then it can be shown that the ratio of model velocity to prototype velocity should follow Froude's law viz

$$\frac{V_m^2}{L_m g} = \frac{V_f^2}{L_f g}$$

or $V_m = \sqrt{\lambda} V_f$

where V_m is the model velocity and V_f is the prototype velocity. However, there are many accidents in which damage is not affected by gravity, e.g., head-on collisions.

Table 4-1 lists 11 dimensionless ratios obtained by Gorman through use of the Buckingham π theorem. It is customary that all geometric parameters be scaled by the same parameter. Hence

$$\frac{D_m}{D_f} = \frac{t_m}{t_f} = \frac{d_m}{d_f} = \frac{R_m}{R_f} = \frac{h_m}{h_f} = \lambda$$

where subscript m refers to the model and subscript f refers to the full scale.

If we use the same material for the model and the full scale such that σ_{y_m} and E_{s_m} are equal to σ_{y_f} and E_{s_f}; the scaling laws can be somewhat simplified; namely

$$\frac{m_m}{m_f} = \left(\frac{\rho_{fm}}{\rho f_f}\right)\lambda^3$$

$$\frac{\dot{\varepsilon}_m^{\,2}}{\dot{\varepsilon}_f^{\,2}} = \frac{\rho_{fl_f}}{\rho_{fl_m}}\,\frac{1}{\lambda^2}$$

$$P_{o_m} = P_{o_f}$$

$$\frac{m'_m}{m'_f} = \frac{m_m}{m_f} = \frac{\rho_{fl_m}}{\rho_{fl_f}}\,\lambda^3$$

$$\frac{K_{d_m}}{K_{d_f}} = \lambda$$

$$\frac{\rho_{fl_m}}{\rho_{fl_f}} = \frac{\rho_{hd_m}}{\rho_{hd_f}}$$

$$\frac{V_m}{V_f} = \sqrt{\frac{\rho_{fl_f}}{\rho_{fl_m}}}$$

Gorman (Reference 5) suggests that although the material used in the construction of the model has the same density as that used in the full size tank, it is possible to alter the effective density of the model head by the attachment of small masses to its surface. This variable can be retained in the scaling laws in order to grant the experiment designer a great degree of flexibility. Note that if the fluid density is the same for model and full scale, scaling says that model impact velocity should be the same as the full scale impact velocity ($V_m = V_f$).

<u>TABLE 4-1</u>

$\pi_1 = \dfrac{\sqrt{m/hd}}{D\sqrt{\dfrac{\zeta_{hd}}{E_s}}}$	The ratio of the time required for the impact force to reach a maximum to the time required for plastic waves to reach the edge of the head.
$\pi_2 = \dfrac{m}{m'}$	The mass ratio of impacting body to impacted body.
$\pi_3 = \dfrac{\sigma_y}{E_s}$	Ratio of yield stress to secant modulus.
$\pi_4 = \dfrac{t}{r}$	Ratio of head thickness to radius of curvature.
$\pi_5 = \dfrac{\sigma_y^2}{E_s \zeta_{hd} \dot{\varepsilon}^2 t^2}$	Ratio of strain energy to head kinetic energy of deformation.
$\pi_6 = \dfrac{mV^2 E_s}{\sigma_y^2 d^3}$	Ratio of impact kinetic energy to strain energy.
$\pi_7 = \dfrac{t}{D}$	Ratio of head thickness to diameter.
$\pi_8 = \dfrac{P_e R}{\sigma_y t}$	Ratio of pressure induced stress to material yield stress.
$\pi_9 = \dfrac{\sigma_y^2}{E_s \zeta_{f1} \dot{\varepsilon}^2 t^2}$	Ratio of strain energy to energy absorbed by fluid.
$\pi_{10} = \dfrac{d}{D}$	Ratio of indenter characteristic length to head diameter.
$\pi_{11} = \dfrac{\zeta_{f1} D^3}{m}$	Ratio of characteristic mass of fluid to mass of impacting body.

Perhaps the most complicating consideration in the scaling is strain rate effects. It is well known that strain rate can have a significant effect on the stress-strain characteristics of mild steel. If we assume that the fluid densities are equal for model and full scale then the strain rate for the model is:

$$\dot{\varepsilon}_m = \frac{\dot{\varepsilon}}{\lambda}$$

For $\lambda = 1/5$, a commonly used scale factor, the model strain rate is 5 times that of the prototype. In order for the equal velocity $V_m = V_f$ to hold, this difference in strain rate must have negligible effect on the material stress-strain characteristics for the model and full scale material. This depends on the materials being used and must receive close consideration in the testing methodology adopted. This is discussed in more detail later in the report.

In summary, based on this survey, it appears that in the collision process where head punctures are encountered such as in head-on collisions, gravity effects can be neglected. If we use the same material and fluid density, then the model impact velocity should be the same as the full scale impact velocity. However, special emphasis must be put on strain rate effects to ensure that the stress-strain behavior of the model material and full scale material are very nearly the same. For equal velocity scaling and same material density, the following scaling holds for other important impact parameters:

Force	$F_m = \lambda^2 F_f$
Energy	$E_m = \lambda^3 E_f$
Acceleration	$a_m = \frac{a_f}{\lambda}$
Velocity	$V_m = V_f$
Time	$t_m = \lambda t_f$

4.6 Small Scale Testing Facilities

Small scale test facilities identified from the literature indicated two basic types; namely vertical drop test facilities and horizontal impact facilities. The following is a description of existing facilities and associated capabilities/limitations.

4.6.1 1/12 Scale Head Impact Test Facility (Vertical Drop Testor)

This section summarizes the subject tests which were conducted at General American Transportation Corporation's Test Center in Sharon, Pennsylvania, under the RPI-AAR Tank Car Safety Project. These tests involved dropping a 1/12 scale coupler backed up by various weights from various heights on 1/12 scale 2:1 ellipsoidal model heads.

The objective of the test program was to gain preliminary insight into the phenomenon of railroad tank car head punctures as caused by couplers of adjacent cars.

Specific parameters studied were: height of drop, weight of impacting mass, internal pressure behind impacted head, and outage of liquid behind impacted head.

The tests were performed by placing the model head in a fixture under a guide tube and dropping a weighted model coupler in it as shown in Figures 4-13 and 4-14. All heads were impacted at their centers.

Coupler forces were measured from strain gages applied on the coupler shank shown in Figure 4-13 and recorded on an oscillograph. An attempt was made to obtain deflection data during the impacts; however, due to inertia of the transducer, the results were not satisfactory.

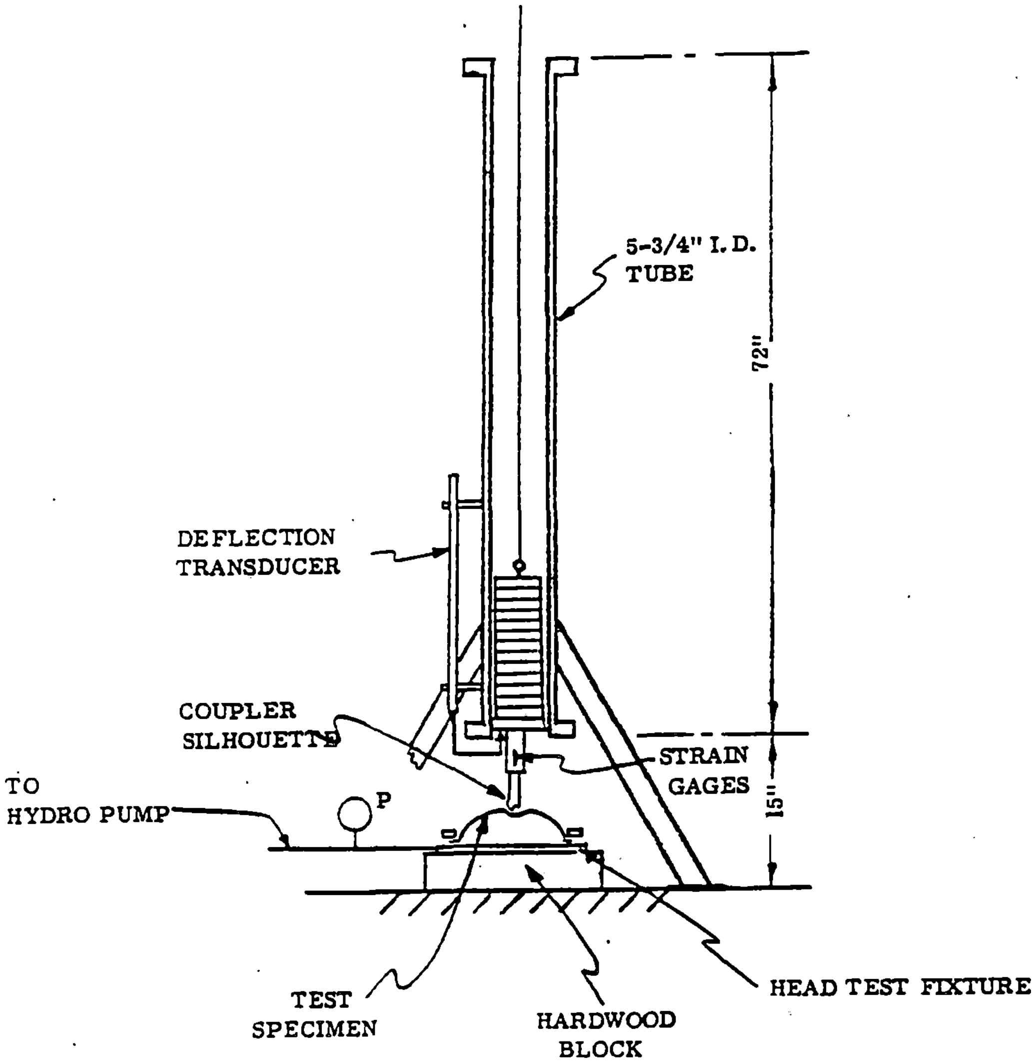


Figure 4-13 Drop test apparatus

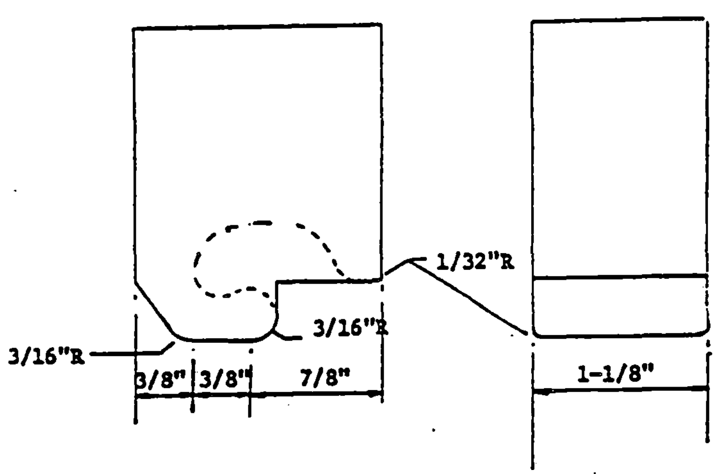


Figure 4-14 Striking object simulated coupler profile

The tests consisted of dropping 50, 75, and 100 pound weights on heads from heights of one (1), three (3) and five (5) feet. Eighteen tests were conducted. The first nine drops were made with the heads empty. The tenth was made with the head pressurized with water with a vapor space, and the eleventh was made with the head shell full (no vapor space) and pressurized. Tests twelve through eighteen were made to determine the threshold at which the model heads would puncture when pressurized and filled with water to both a shell full and partially full (nominal air space) condition. The model heads were fabricated from ASTM 365-killed steel with an inside diameter of 10 inches and shell of thickness of .0598 inch.

One of the deficiencies in a vertical drop test of this type is the inability to offset gravity and hence the inability to accurately scale the results up the full scale. The inertia effect of lading cannot be simulated. Test of this type can be used to study indentation, energy absorption and puncture characteristics and validate theoretical structural models (finite-element for example) by comparing observed results with theoretical predictions.

Recognizing there are inaccuracies associated in scaling up the results, it is interesting to compare threshold conditions of puncture to full scale results.

With a 1/12 scale factor, the model heads simulate a 120" diam. x 0.718" thick head on a full scale car. Since the model head material has a tensile strength around 65,000 psi, it is not unreasonable to assume that the full scale car head is of TC-128-B steel which is equivalently 0.718 x (65,000/81,000) = 0.575" thick. This is thinner than the typical 11/16" thickness used on a 112A340W car, so (provided the scale laws applied) the 1/12 scale tests should predict threshold puncture conditions which are less severe than actual. Tests from Table B-1, Appendix B of Reference 4 indicate that the threshold puncture energy for 85 psig internal pressure is 425 ft-lbs. This would correspond to dropping a 100 pound weight from a height of 4.25 feet. The corresponding impact speed would be $V = \sqrt{2gS} = \sqrt{2(32.2)4.25} = 16.6$ ft/sec or 11.3 mph. The scaled up impact car weight would be 100 lbs. x $(12)^3$ = 172,800 lbs. This is fairly close to a threshold puncture speed of about 12.5 mph observed in full scale tests discussed in Section 4.1. The 1/12 scale test most nearly approximating a 9.3 mph full scale test was one which involved a 9.4 mph speed with 85 psig pressure under the head. The impact force extrapolated to full scale is 5210 x 144 = 750,000 lbs, and the extrapolated dent depth is 13/16 x 12 = 9.7". The corresponding values in the actual full scale test were 504,000 lbs and 6".

Hence even though the vertical drop test does have limitations, it does appear to give reasonable scaled up results and may prove very useful in screening shock mitigation concepts and in validating theoretical structural models.

4.6.2 1/5 Scale Model Tests

This section summarizes tests which were conducted by ACF Industries' AMCAR Division, Engineering and Research Center, at St. Charles, Missouri, under the RPI-AAR Tank Car Safety Project. ACF Report N0923 (Reference 15) which is on file in the RPI-AAR Project Library, contains full details of the test program, and includes all drawings of the test tanks, trucks, draft gears, hammer car, air brake arrangement, coupler assemblies, and a number of photographs of the test set-up and the impacted tank car heads.

The objective of this program was to study head puncture phenomena for two designs of the head-to-sill connection on stub sill non-pressure and pressure tank cars and, more specifically, to determine if a significant difference exists in head puncture vulnerability between these two designs.

All test tanks were designed with 22" I.D. heads. Since the head I.D. on most current non-pressure cars is around 110", the non-pressure car scale factor is 22/110 = 1/5. On the other hand, the head I.D. on most current pressure cars is around 118.5", yielding a scale factor of 22/118.5 = 1/5.386.

* Non-pressure cars

A sketch of the test arrangement (from Reference 15) for the non-pressure cars is shown at the top of Figure 4-15. The dimensions, weights, and volumes have been extrapolated to the full scale conditions. All non-pressure cars were loaded with water to 2% outage and tested under 0 psig pressure.

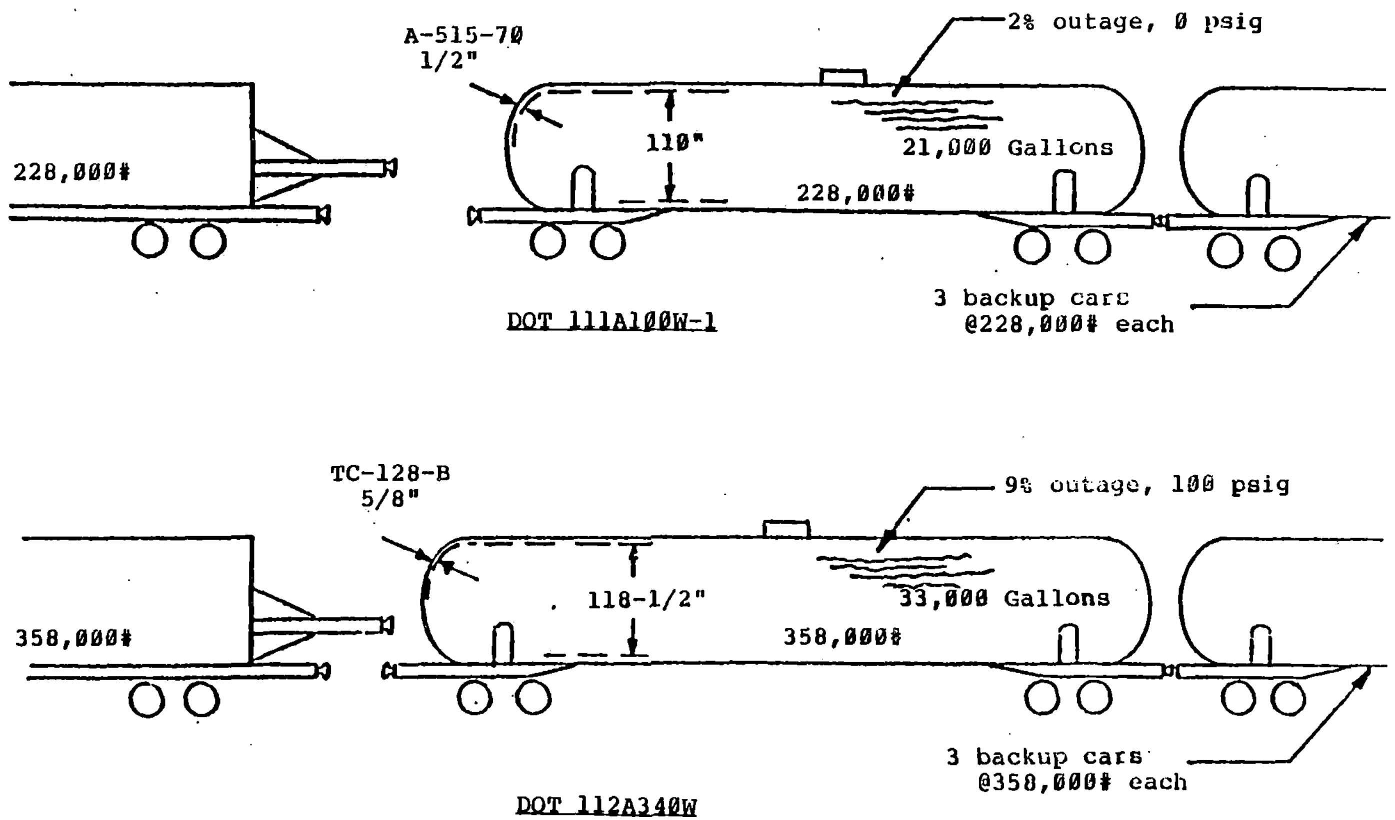


Figure 4-15 1/5 Scale head impact test set-up

(All values scaled up to full scale)

The heads were constructed of A-515-70 steel from 0.109" thick blanks. These blanks were purchased 3/16" thick and were surface ground to the required thickness by removing equal material from each side. The heads were cold pressed and the resulting thickness ranged from 0.107" to 0.110" in the areas of impact. Head material properties were determined in four tensile tests, using two radial and two circumferential specimens. Average properties from these four tests were:

Yield point	- 44,250 psi
Tensile strength	- 71,090 psi
Elongation in 2"	- 26.1%

* Pressure cars

Details of the pressure car tests are shown at the bottom of Figure 4-15. The head material for the pressure cars was TC-128-B, also purchased in 3/16" thick blanks. These were surface ground on both sides to a thickness of 0.125". After cold pressing, the thickness varied from 0.121" to 0.125" in the areas of impact. The average material properties as determined from two radial specimen and two circumferential specimen tensile tests were:

Yield point	- 48,975 psi
Tensile strength	- 75,600 psi
Elongation in 2"	- 28.6%

* Scaled Up Results

It is of interest to compare the 1/5 and full scale tests. The results of full scale tests indicate that the threshold puncture speed for a full scale 112A340W car under 100 psig pressure is between 9.3 mph and 12.7 mph and that the puncture (ram) force is between 504,000 lbs. and 675,000 lbs. These are for conditions of a 348,000 lb. ram car, a 340,000 lb. test car, 9% outage, 11/16" thick head, an impact location 24" above the sill, and three backup cars weighing a total of 382,750 lbs. The corresponding values for the 1/5 scale tests are seen to be about 7.2 mph and 670,000 lbs.

The puncture speed is lower in the 1/5 scale test primarily because the scaled three backup cars weighed over 2-1/2 times as much as the three backup cars in the full scale tests. The correlation of puncture force, however, is good.

In summary it appears that 1/5 scale model tests offer the potential of a cost effective, reliable way of studying head puncture. The 1/5 scale factor permits reasonable thicknesses for shell components, and should allow the effects of lading to be accurately simulated. It is obviously more realistic than the vertical drop test facilities.

4.7 Instrumentation Techniques Used In Impact Testing

* Force Measurements

In most impact tests, contact forces were measured because they were helpful to the understanding of the impact phenomenon. Strain gages were mounted on coupler shanks to form force dynamometers which could be calibrated (References 2, 4) and could then be used to register the time histories of the forces acting on tank heads during impacts.

* Displacement Measurements

Optical means were exploited in the measurements of displacement of both the ram (impacting) car and the impacted test car. For impact tests described in Appendix C of Reference 4, long boards painted with alternate black and white stripes of equal width were placed on both cars. Photo cells wired into a bridge circuit scanned the striped boards during the impacts, causing a variable period wave to be recorded on the oscillograph tape. The wave period would be the time it took for the car to travel a distance equal to the width of the stripes. On the tape there were also recordings of time lines to avoid inaccuracy of time measurement stemmed from the imperfection of the recording mechanism. From the displacement histories of the cars, the minimum distance between the two striped boards can be found. This distance could be related to the depth of the permanent indentation (if no puncture) on the tank head.

* Velocity Measurements

In the RPI-AAR Test #27093 (Reference 4), impact velocity was oscillographically recorded and was transduced by an electrical contact pip board positioned along the test track. A roller mechanism attached to the leading end of the hammer car contacted the transducer and produced one occurence signal on the oscillograms for each 2" increment of hammer car travel immediately before impact. A hammer car velocity versus impact ramp position curve was developed prior to both the pressure car tests and the non-pressure car tests for assistance in controlling the tests and also for backup information in case the event of an oscillograph failure. Two precise timing signals were recorded on each oscillogram; one from a 1000 cps crystal oscillator and also the Bureau of Standard radio signal from Station WWV.

* Acceleration Measurements

Longitudinal accelerations of the impacted car and ram car were measured in the RPI-AAR 1/5 scale Head Impact Tests (RPI-AAR Test #27093, Reference 4) with Statham type A5A-50-350 resistance-wire accelerometers. The dynamic signals were recorded with galvanometers having a flat frequency response of 0-100 cps. The impacted car accelerometer was mounted on the side of the inboard stub sill, and the hammer car accelerometer was mounted on top of the ram. Maximum acceleration data of each test were presented in the report.

* Motion Pictures

Motions of the cars during impacts were recorded in the films with cameras in several impact tests (References 2, 4). Cameras were focused at different points and recordings were made at different frame speeds. Motion pictures of the impact phenomenon is very useful in double-checking the speed of the ram car at the moment of impact.

5. RECOMMENDED SMALL SCALE TESTING METHODOLOGY

5.1 Tank Head Puncture Test Requirements

Current DOT performance standards require that the tank car with protection devices shall be capable of sustaining, without loss of contents, coupler to tank head impacts within the area of the tank head at relative car speeds of 18 miles per hour when

(a) The weight of the impact car is at least 263,000 pounds.

(b) The impacted tank car is coupled to one or more backup cars which have a total weight of at least 480,000 pounds and the hand brakes are applied on the first car.

(c) The impacted test car is loaded with water at six percent outage with an internal pressure of at least 100 psig.

(d) At least two separate tests are conducted with the coupler on the vertical centerline of the ram car. One test shall be conducted with the coupler at a height of 21 inches above the top of the sill, the other test shall be conducted with the coupler height at 31 inches above the sill.

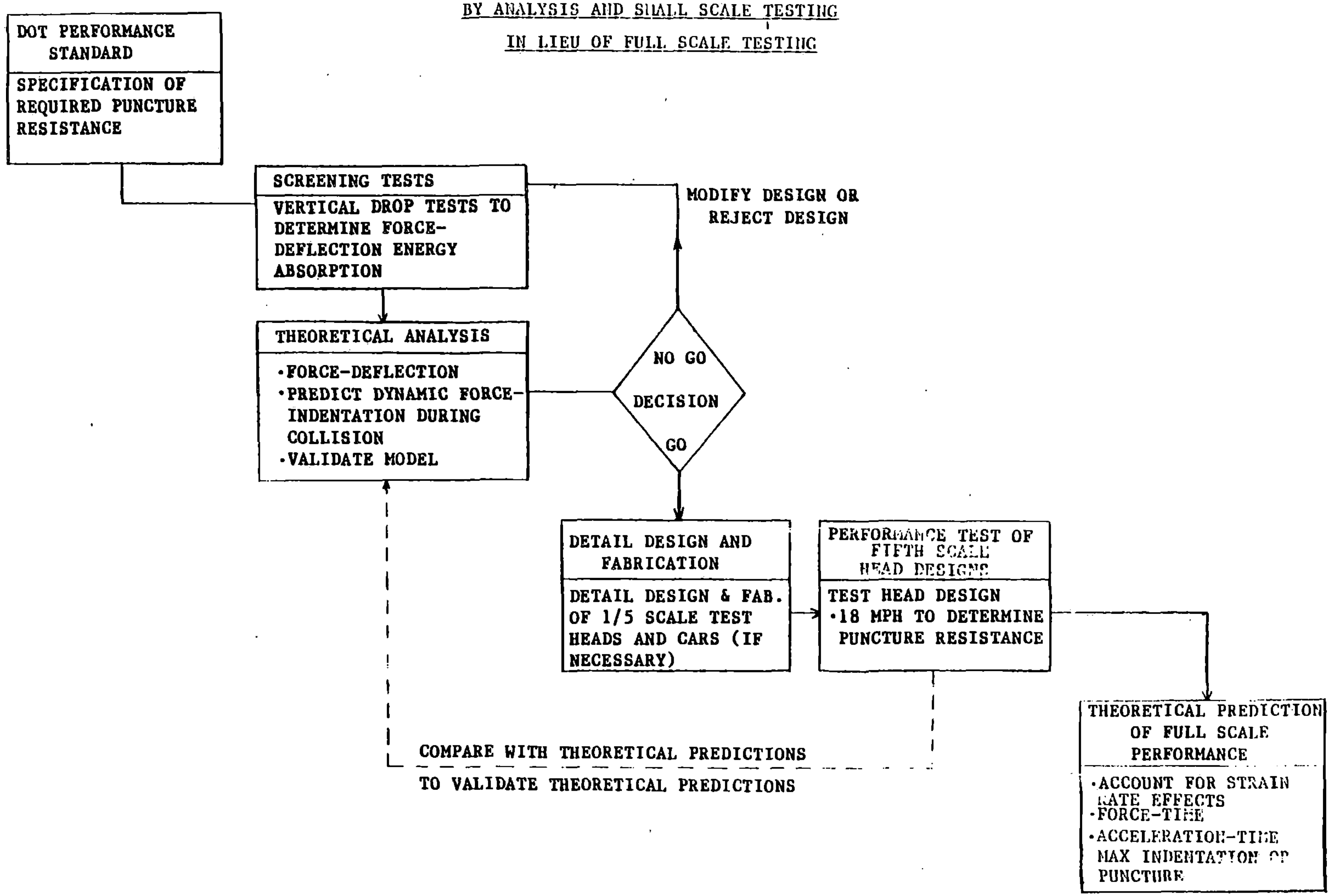
A test is successful if there is no visible leakage from the standing tank car within one hour after impact.

5.2 Evaluation of Tank Head Designs

Current standards require that compliance of new tank head designs with the above requirements be verified by full scale testing. A significant cost advantage could be realized if the puncture resistance of new tank head designs could be verified by small scale model tests. Based on the results of the literature search described, the following small scale methodology is proposed as a means of simulating the DOT performance test requirements described. The methodology is illustrated in Figure 5-1. It is based on building 1/5 scale models of the impacted tank car and backup

FIGURE 5-1

PROPOSED METHODOLOGY TO
EVALUATE PUNCTURE PERFORMANCE OF TANK CAR HEAD DESIGNS
BY ANALYSIS AND SMALL SCALE TESTING
IN LIEU OF FULL SCALE TESTING



cars with the equivalent full scale characteristics given in Section 5.1. Impact tests would be conducted at at least 18 mph to evaluate resistance of the tank head designs to impact with the coupler at equivalent full scale positions of 21 inches and 31 inches above the sill. The steps involved are described in the following.

5.3 Screening Tests-Vertical Drop Tests

Although vertical drop tests of the type described in Section 4.6.1 do not entirely simulate the impact of tank cars, they do provide a rather inexpensive means of evaluating the energy absorption capabilities of new tank car head designs including those with shock mitigating covers/shields. A 1/10 scale vertical drop facility as shown in Figure 5-2 will be used to compare the threshold energy for puncture of the new design with that of a conventional head designs which can be used as a standard. Only those designs which exceed a threshold value (to be determined) would be recommended for further evaluation.

A study should be conducted in a follow-on program to determine the preferred vertical drop test setup based on cost and suitability of satisfying environmental conditions. The setup shown in Figure 5-2 is discussed in the following to illustrate basic testing procedure which could be used. The block with a protruding model coupler attached could be dropped along the guide tube to hit on the model tank head fastened on a slightly elevated planform. The weight of the block should be variable and the guide tube should be long enough such that we can simulate the worst possible impact condition for the larger scale model. Self-temperature-compensating strain gages would be mounted on the coupler shank and the inner and/or outer surfaces of the tank head. Accelerometer would be mounted on the block and a high speed camera would be used to keep track of the denting of the tank head. Multi-channel tape recorder would be employed to record the strain and the acceleration histories.

The setup could also be used to evaluate protecting devices. For example the setup for the model tank head with shield is shown in Figure 5-3. The head shield is to be weakly supported just above the tank head.

The setup for the model tank head with both head shields and mitigator is shown in Figure 5-4. The mitigation could be glued on the shield and the shield would be supported as stated above.

For low temperature tests, thermocouple would be attached on the specimen. Coolant such as dry ice would be applied to the surface of the test specimen as evenly as possible.

Before each test is performed, each of the measurement devices would be calibrated and a sinusoidal signal with known magnitude and frequency recorded in every channel ahead of test data to serve as reference. Steps should also be performed to ensure proper function of the trigger device for the camera and the correct location of the tank head to be impacted.

Some of the most important variables which would be monitored are the following:

1. The impact force.
2. The impact velocity.
3. The tank car head impact stresses.
4. Tank head deformation.

The impact force would be monitored with a load cell. The load cell would be inserted between the impact head and the impact mass to monitor the transmitted force.

The impact velocity and acceleration would be monitored with an accelerometer mounted on the back of the impacted mass. The output will give a direct measure of the acceleration during the impact. Impact velocity can be measured with simple optical devices as mentioned in Section 4.7.

Several strain gages would be mounted on the inner surface of the tank head. During the impact these gages will monitor the strain history of the head deformation.

To record the tank head deformation history a high speed movie camera could be used.

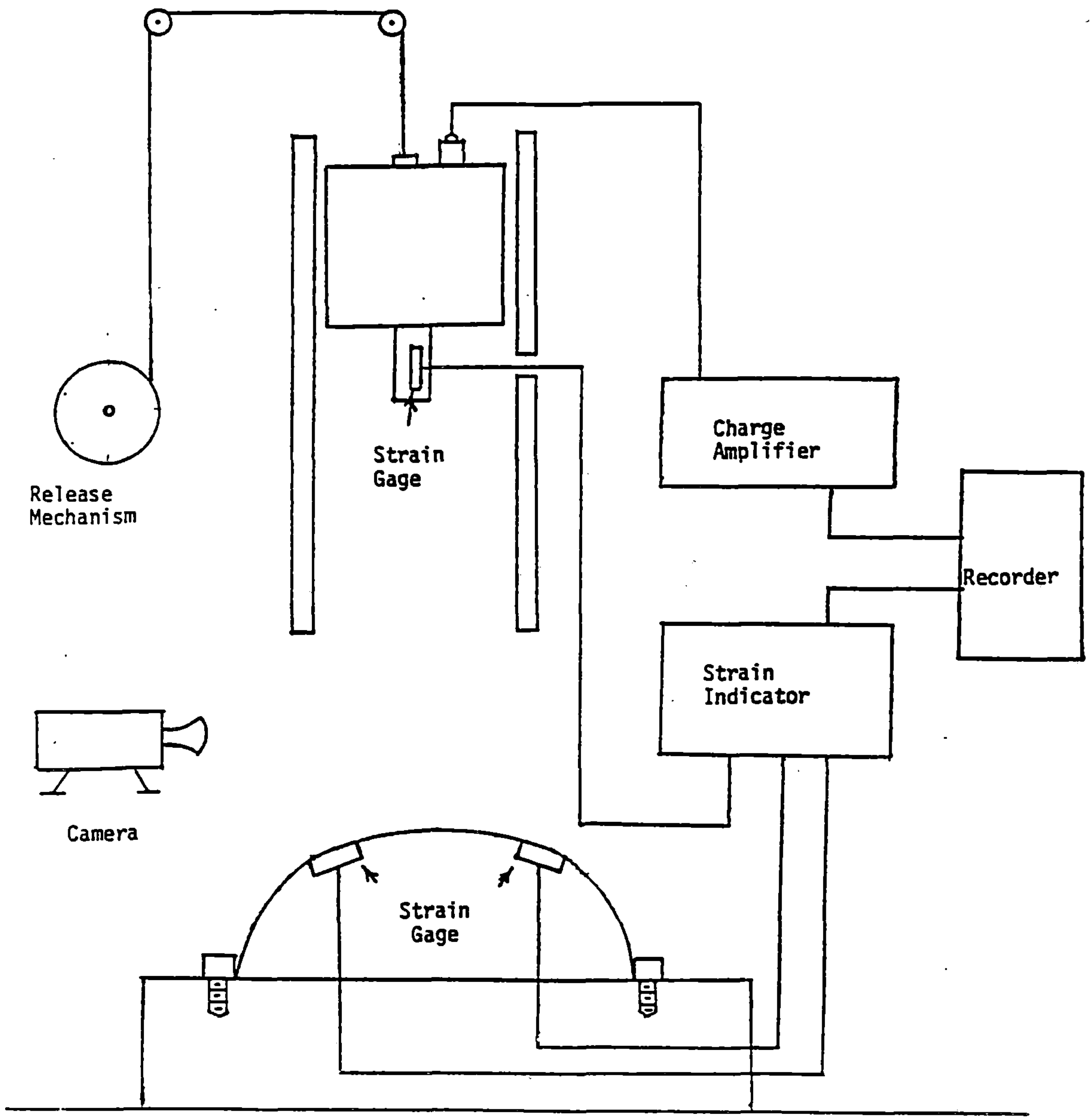


Figure 5-2 Test set-up with tank head only

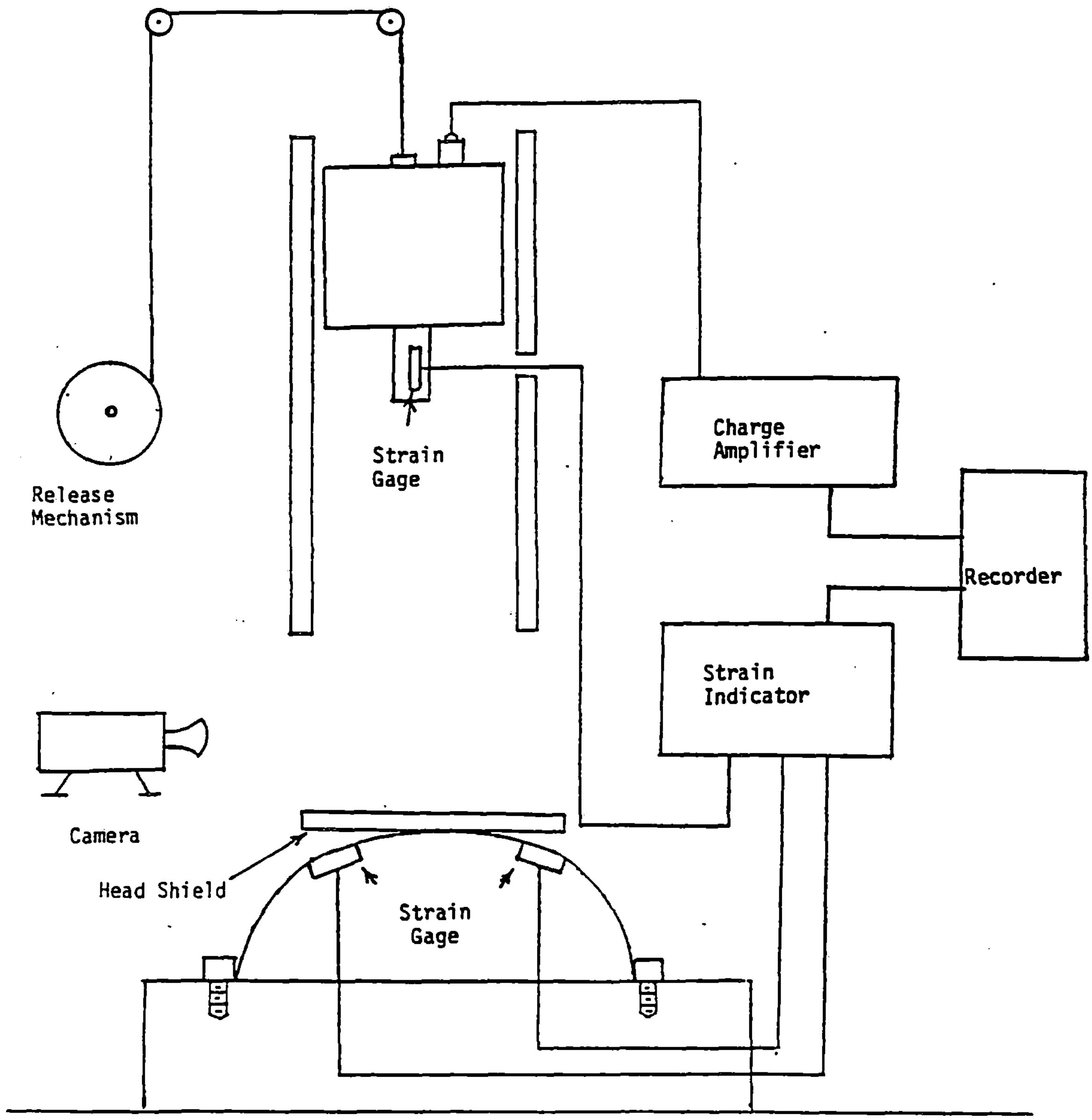


Figure 5-3 Test set-up with tank head and head shield

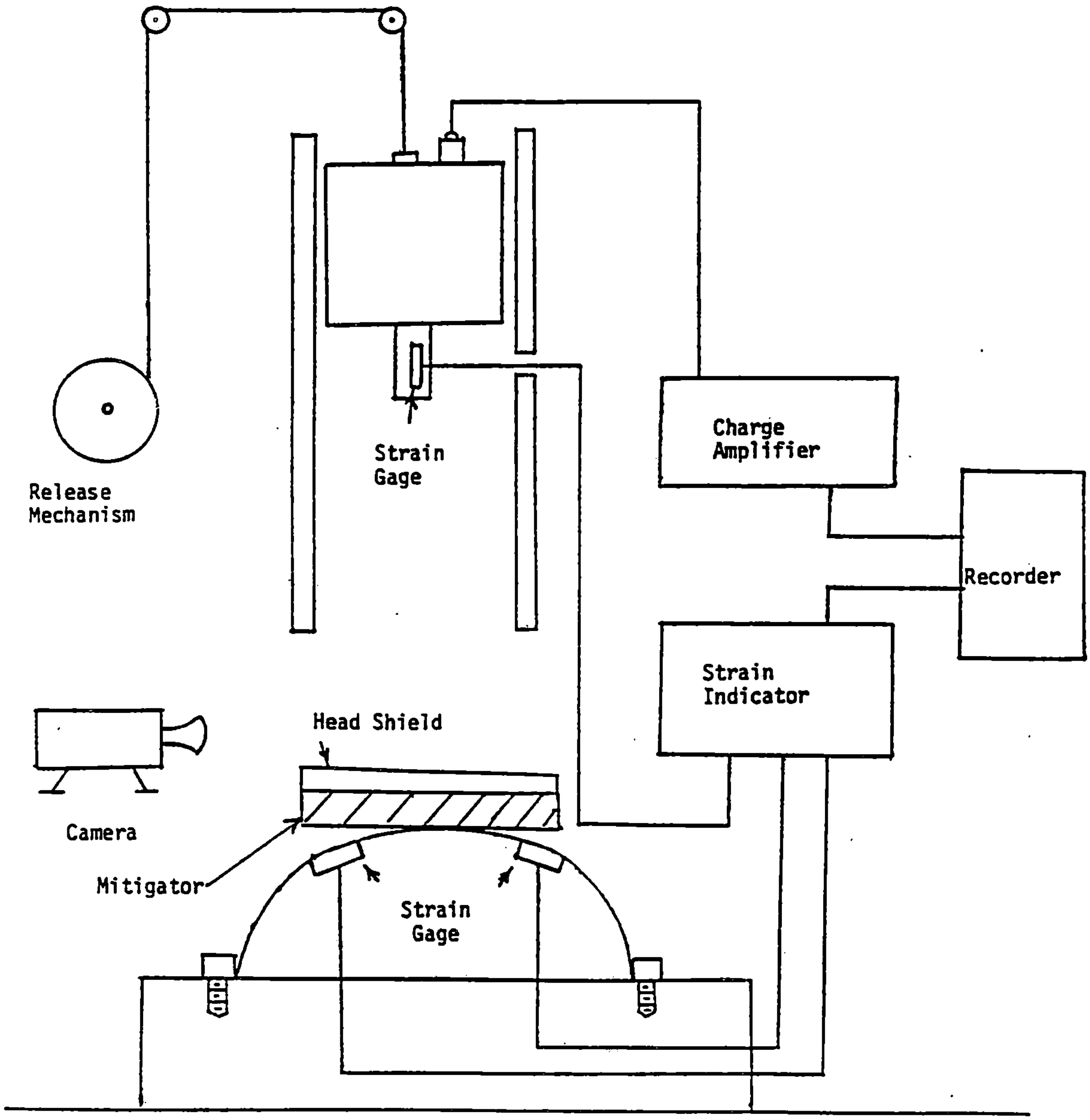


Figure 5-4 Test set-up with tank head, mitigator and head shield

5.4 Theoretical Analysis

A finite element structural model will be used to estimate the force-indentation characteristics of candidate head designs. One of the deficiencies in the scaling laws (Section 4.5) is the inability to duplicate strain rate effects with a scaled model. A method frequently used when there is uncertainty in scaling is to develop theoretical models, validate their accuracy by comparison with small scale tests, then use the validated theoretical models to predict the performance of the full scale. The vertical drop test results will provide the necessary experimental data to validate the theoretical finite element model. Sensitivity of stress-strain behavior to strain-rate will be included in the theoretical model in an approximate manner. Strain rate generally has the effect of increasing the stiffness (increase in elastic limit) and decreasing the strain to failure but with a net increase in the energy absorption.

Once the theoretical force-indentation model has been validated by comparison with the vertical drop tests, the impact-force time pulse, indentation-time, energy absorption can be calculated during impact of the 1/10 scale models conforming to the equivalent full scale characteristics of Section 5.1. A theoretical estimate can be made of the indentation at the required 18 mph and an assessment made of the probability of puncture. If the results of the analysis are positive then the tank car head design can be committed to detail design and fabrication. If the results are negative the design may be rejected or modified and reanalyzed.

Substantial effort has been concentrated on the study of analytical means that can be incorporated as a portion of the small scale testing methodology. As stated above, this is to eliminate the doubts in scaling due to the effect of strain rate on the constitutive properties of the materials of the tank head and the protective devices. Another reason is to have a pre-test estimation of the threshold velocity at which the system under test will be punctured. There exist finite element programs that seem to be suitable for this purpose, and some have been tested.

The first program tried is called HONDO II which is an extension of the computer code called HONDO. HONDO is a finite element method computer program designed to calculate the large deformation elastic and inelastic transient dynamic response of axisymmetric solid. HONDO II is the late version of HONDO that includes capabilities such as sliding interfaces, restarting, etc. More description of HONDO II can be found in Appendix A. A simplified model shown in Figure 5-5, corresponding to the Impact Test No. 1 of first series of RPI-AAR Test 27053 (Reference 4, Appendix C), was studied with HONDO II. Mass B, simulating the ram car with initial speed of 4.3 mph was arranged so that the total mass was equal to 128,900 lb., corresponding to the test data and the diameter of the impact area was similar to that of the coupler. The test car tank head, mass A in Figure 5-5, was simplistically modelled as a two-to-one-ratio ellipsoidal shell with rim constrained. The purpose of this exercise was to see whether experimentally determined dent depth of 2 3/4 inches and dent area diameter of 30 inches could be approximated with HONDO II. After several successful test runs, it was realized that due to the relatively thin wall of the tank car (wall thickness is small compared to the diameter of the tank head), the amount of computation was beyond the capacity of the computer used.

Another finite element program called "STAGS" was later selected to simulate the tank car impact phenomena. STAGS was designed for the dynamic analysis of thin shell under both static and dynamic loadings. Sliding interface which is essential for the simulation of two bodies under impact situation was not provided in STAGS. Therefore, the simulation was done by modelling the two impacting cars as one continuum with the part representing the ram car moving initially at the impact velocity and with the other part representing the impacted car standing still. The initial study showed that the predictions of the deformation and the stress of the tank head with both STAGS and HONDO II were similar within a few microseconds. However, the predictions of the stress level in the coupler shank were significantly different, and it seemed that more reasonable results were obtained with HONDO II. The amount of computation needed to complete an impact simulation with STAGS was much smaller than that with HONDO II, but it was still unaffordably large. A complete simulation was never accomplished with either HONDO II or STAGS.

In order to reduce the computation involved in the numerical simulation process, simplified approach was studied. Since the mass of the tank head impacted was only a small fraction of the total mass of the car, the inertia force due to its deforming

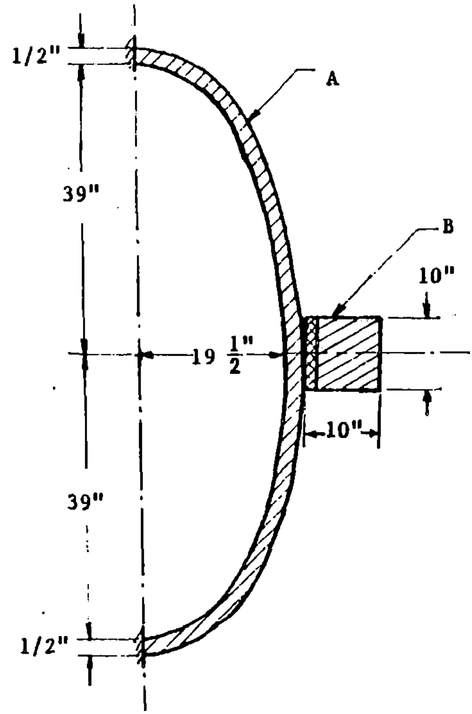


Figure 5-5 Finite element model for HONDO II

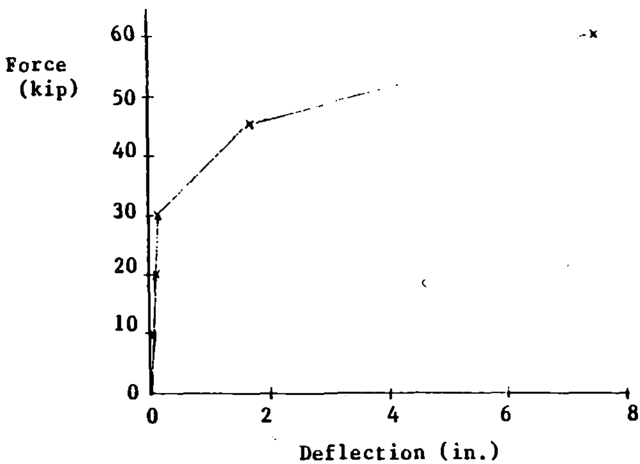


Figure 5-6 Force-deflection curve obtained from static analysis

acceleration was negligible. Therefore, the tank head could be approximated as a nonlinear spring and the tank cars could be simply modelled as rigid bodies. One previous test (Test No. 3 of second series of RPI-AAR Test 27053, Reference 4, Appendix C) was simulated to see if this approach could yield reasonable results. The first step was to generate the force-deflection curve of the rim-restrained tank head, and a program called "DOASIS" was used. DOASIS is a finite element program designed for the static analysis of elastic-plastic axisymmetric deformations of axisymmetric bodies. Different load distribution within the contact area had to be tried out to see which distribution gave results comparable to the experimental ones. A concentrated load with its magnitude the same as the maximum force measured in the test yielded a deflection at apex too large in comparison with the depth of dent recorded for that test. With several trial distributions, the deflection at apex was reduced to reasonable values, but the diameter of the dent still did not correspond well with the experimental results. This lack of agreement could be attributed to the fact that in real tests the tank head inertia effect would tend to impede the deformation of contact area and thus effectively increase the stiffness. Under the same conditions the force-deflection curve was then generated with DOASIS as shown in Figure 5-6. The unloading curve could not be simulated with DOASIS, so it can only be speculated. Since the impact of concern always involved a lot of plastic deformation, it was assumed that the unloading curve was vertical line drawn through the point of maximum force. The dent depth reported in Reference 4 was 6.5 inches, that corresponds to energy absorption of around 300,000 in-lb from Figure 5-6. The inelastic impact of two bodies with masses and initial velocities corresponding to the test, however, requires energy dissipation of 485,000 in-lb. Possible reasons for the discrepancy are the following.

1. Due to wheel bearing friction, the effective impact speed could be lower than the reported speed of 6 mph, which should be used in the calculation of energy dissipation and lower its value.

2. Figure 5-6 shows that the loading force associated with 6.5 inches of apex deflection is 57 kips, which is smaller than the maximum force of 59 kips reported in Reference 4. If the curve is corrected, the energy absorbed should be higher.

3. The actual unloading probably involved reduction in deflection, and this could result in actual maximum deflection in the impact process greater than 6.5 inches, which is the permanent dent depth. This would increase in energy absorption somewhat.

4. Parts of the impacting tank cars other than the tank head had some deformation and thus contributed to the energy absorption. More study has to be performed to identify these parts, but it might be too complicated to combine these parts in one mathematical model.

5. There might be some measurement errors, e.g., the recorded maximum force of 59 kips might be lower than actual force, or the reported impact speed of 6 mph might be higher.

It can be concluded here that although the tank head deformation does not absorb all the energy dissipated in an impact, it contributes to a substantial portion of the energy absorption. Therefore, the force-deflection curves of the restrained tank head with and without protection devices obtained from finite element programs can be first-cut analysis to evaluate the puncture resistance. However, DOASIS is not sufficient in the analysis of a system under large deformation with high strain, and this would render the higher loading results less trustworthy. The applicability of DOASIS would be even worse when energy absorbing materials are used as part of the protection devices since they would be designed to be highly deformed and strained. Shown in Figure 5-7 is a half cross-section of the tank head with conforming mitigator and head shield that was studied with DOASIS and the results obtained were unreasonable when certain loads were reached. Other suitable finite element programs, such as NONSAP, a program for nonlinear stress analysis, have to be tried to see if more reliable results can be obtained.

5.5 Detailed Design of 1/5 Scale Models

Those tank/protection device designs which pass the vertical drop and theoretical analysis phases will be committed to detail design for fabrication of 1/5 scale models of the impacted car and backup cars. It is expected that standard impacted and backup cars can be designed and fabricated with the capability of being modified to

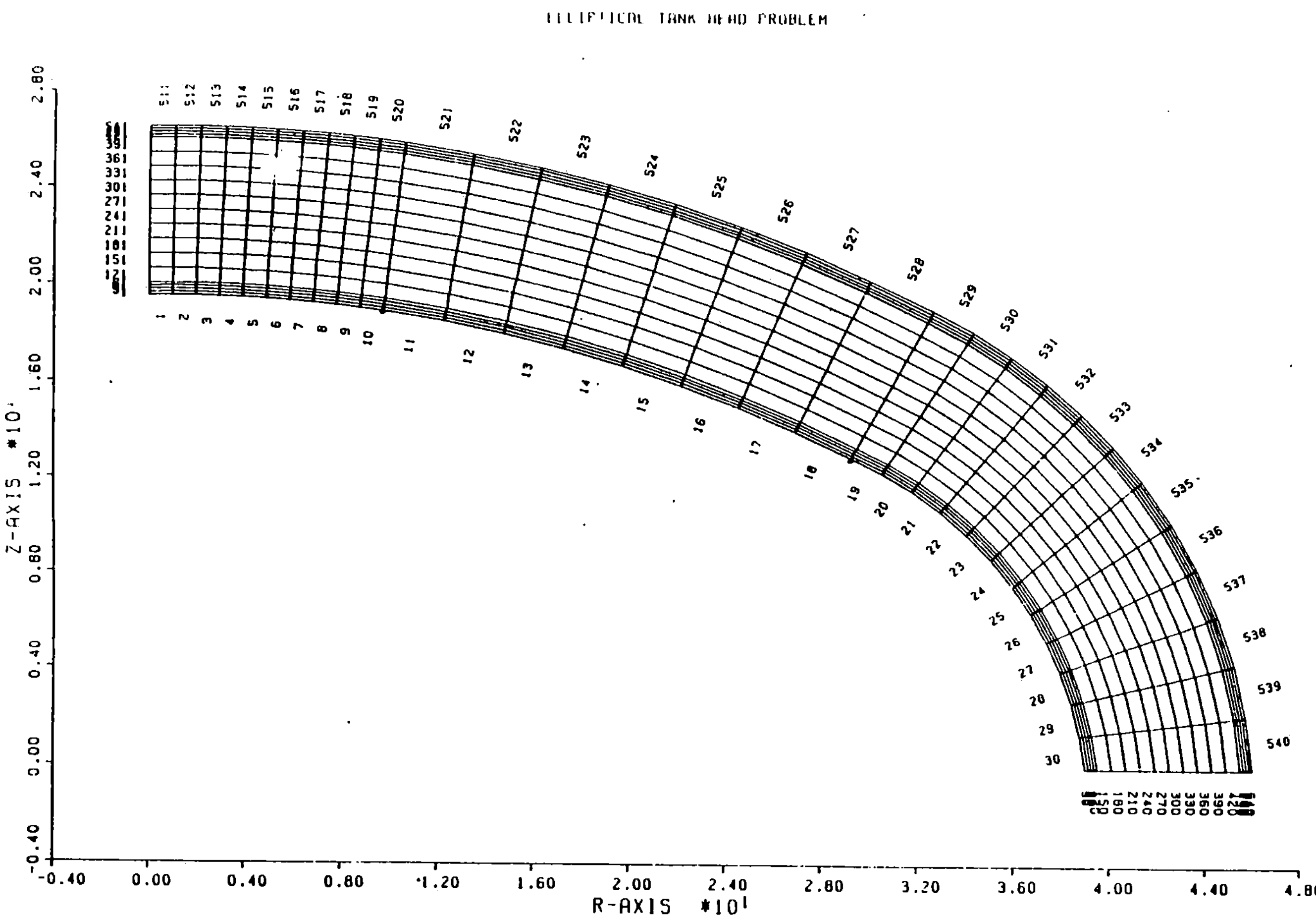


Figure 5-7 Mesh of Tank Head with Mitigator and Edge Shield for [illegible]

accomodate new tank heads and protection devices as new designs evolve for evaluations. Flexibility of removal of old heads and replacement with new head designs will minimize the cost of evaluating new head designs. The tank cars should be fabricated from the same materials used in current designs; for example TC-128-B. The design should be very similar to the 1/5 scale tests discussed in Section 4.6.2. In fact, if those cars are still avaliable they would serve as excellent prototypes to check out the methodology being proposed.

5.6 1/5 Scale Models To Simulate DOT Tank Head Performance Tests

As previously discussed in Section 4.6.2, the most realistic method of simulating full scale collisions of tank cars is by horizontal impact of 1/5 scale model cars with lading simulated and model velocity the same as the full scale. In the horizontal collision with the coupler impacting 21 inches (full scale) and 31 inches above the sill, gravity plays very little part in the collision process and equal velocity scaling applies. Hence the model test setup must be capable of velocity of at least 18 mph, various ways have been studied to develop their velocity on a 1/5 scale model of the impact car. Figure 5-8 shows one possible approach where the impact car runs down a ramp of about 15 feet elevation to obtain the required velocity. The impacted car plus backup cars are stationary on the tracks with the hand brakes applied in the first car in conformance with the requirements of Section 5.1. The velocity can be varied by controlling the position up the incline where the car is released. The test setup would be similar to that shown in Figure 5-9.

Experimental measurements will be made of force-time; acceleration-time, velocity-time and indentation as a function of velocity. These measurements will then be compared with the theoretical predictions (Section 5.4) to validate the theoretical models.

5.7 Prediction of Full Scale Performance

A first cut at the predicted performance of the full scale can be obtained from the 1/5 model tests by using the scaling laws as discussed in Section 4.5. For example, for a 1/5 scale model ($\lambda = .20$); the following laws apply assuming the same structural material is used in the model and full scale prototype.

Force	$F_p = \frac{F_m}{\lambda^2} = 25F_m$
Energy	$E_p = \frac{E_m}{\lambda^3} = 125E_m$
Acceleration	$\partial_p = \frac{\partial_m}{\lambda} = 5\partial_m$
Velocity	$V_p = V_m$
Time	$t_p = \frac{t_m}{\lambda} = 5t_m$
Strain rate	$\dot{\varepsilon}_p = \frac{\dot{\varepsilon}_m}{5}$

If the material exhibits little sensitivity to strain-rate, the scaling laws should give reasonably accurate results for the full scale. If strain rate is found to be important, then the theoretical models can be used to calculate the full scale performance factoring in the change of material strength (constitutive equation) due to strain rate. Vertical drop test and 1/5 scale model tests provide sufficient data to validate the theoretical models, hence, there should be a high confidence level in the full scale predictions.

5.8 Instrumentation Requirements

Velocity of ram car at the instant of impact is the most important measurement to be made accurately. From several tests with different velocities of impact and with all other conditions kept the same, the threshold velocity of the specimen are to be determined.

In order for the scale model results to be predictive of prototype results under specified conditions, various parameters have to be properly controlled to satisfy the scaling laws. Masses of cars, diameter and thickness of the tank head, strengths of the materials, temperature, and pressure and outage of the lading in the impacted car are the important parameters, and their measurements should be properly instrumented.

In order to ensure that the velocity is correctly measured, more than one method of instrumentation should be applied. Motion pictures, if they can be taken, would be very helpful in verifying events that happen during an impact.

In any controlled experiments, there exists the possibility that some results obtained may be in doubt due to unforeseen conditions. Sometimes the results are contrary to what one expects basing on engineering intuition, and this may cause concern. Under these situations, more extensive measurements can help in the understanding of the phenomenon in question and in resolving the difference between the intuition and the results. This is especially true for scale model tests since the similarity in the phenomena between a model and the prototype may be lost in the construction of the model, and thus render the results inapplicable to the prototype. Measurements such as the contact force, the displacement of the impacting coupler during impact, the strains at several strategic locations and the accelerations of a few points of the cars are useful. Energy and momentum conservation as well as Newton's second law of dynamics can be applied on the measured data to identify the critical mechanisms. Also, the measurements facilitate the checking of the validity of the theoretical analysis which in turn helps in the understanding of the phenomenon. Furthermore, a valid theoretical analysis can be economically applied to improve the tank car puncture resistance with either new designs or new protection devices.

In order to obtain the extra measurements, various instrumentation such as described in Section 4.7 has to be used. Frequency response of the instrument chosen should cover a range from zero to several hundred Hz. Recording devices are required to enable data reduction, signal processing and energy or momentum calculation.

5.9 Sensitivity of Results To Experimental Errors

The thickness of the head shell model will have direct bearing on any predicted threshold puncture velocity. With the RPI-AAR report RA-05-1-17 (Reference 4) as a guideline, it can be seen that puncture velocity is proportional to thickness to the 1/1.5 power, or

$$V_p \propto t^{1/1.5} \qquad \text{from Equation 6-2.}$$

When this is applied to a 1/5 scale model of a 11/16" thick prototype, it can be seen that a thickness tolerance of within ±0.005" in the model head will maintain accuracy within 5.72%.

Similarly, the head radius R is proportional to puncture velocity by the power of 1/1.5.

$$V_p \propto R^{1/1.5}$$

A 1/5 scale model of a 100 inch tank car prototype held to a tolerance of ±0.05" in radius will maintain accuracy within 0.67%.

Since these tolerance should be easy to achieve, it can be assumed that the variations in threshold puncture velocity due to dimensional variations should be well within 6.4%.

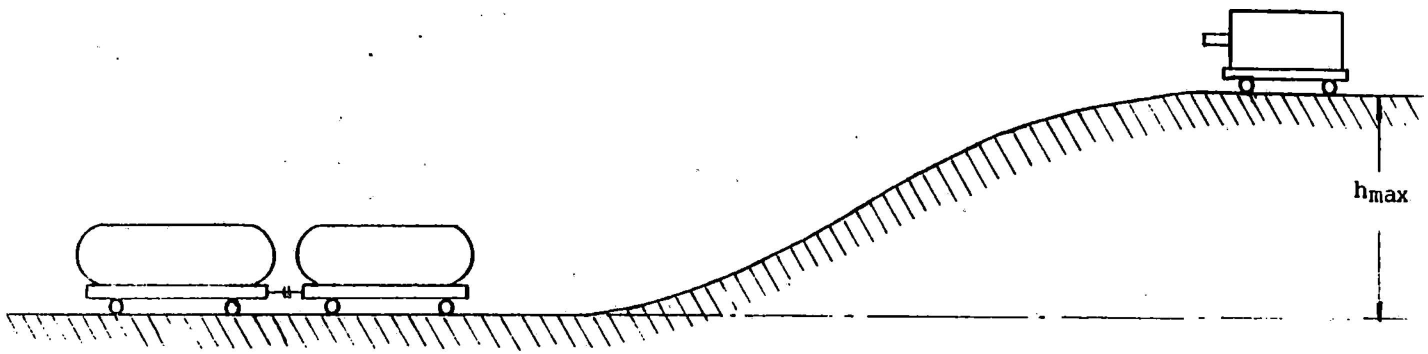


Figure 5-8 Impact test setup

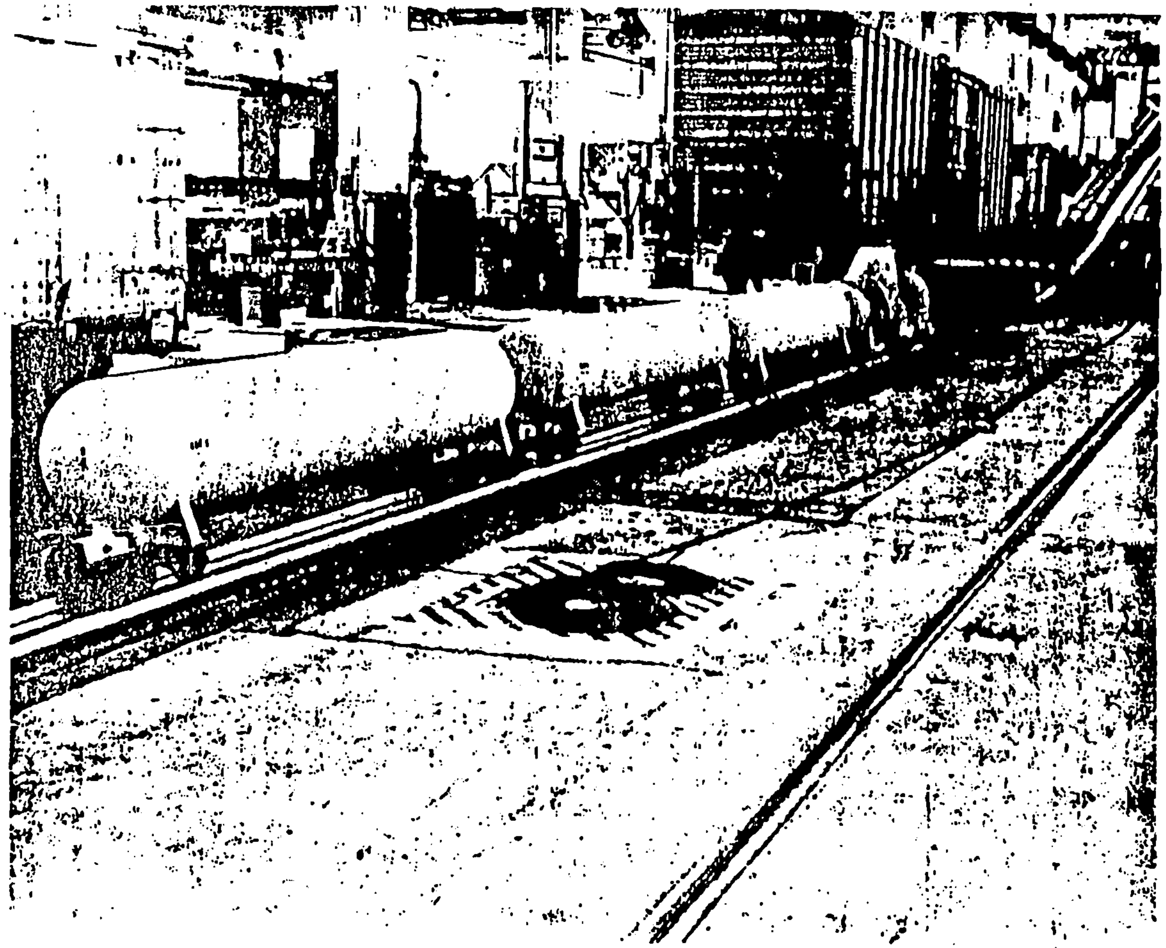


Figure 5-9 Test set-up for pressure cars
Left to right: 3 backup cars identical to test car, test car, and ram car.

6. CONCLUSION

Based on the results of this study it is concluded that:

(a) It is feasible to design a small scale testing facility to adequately simulate full scale tests at impact speeds of up to about 20 mph in conformance with the requirements of the DOT performance tests. A scale factor of about 1/5 appears suitable but a more extensive study should be conducted to optimize the scale factor based on cost and test accuracy.

(b) The methodology proposed allows the experimental model results and theoretical models to be used to accurately predict the performance of the full scale cars.

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APPENDIX A - HONDO II COMPUTER CODE

HONDO II is an extension of the computer code called HONDO. So before describing the additional features which make up HONDO II, the features of HONDO will be described. HONDO is a finite element method computer program designed to calculate the large deformation elastic and inelastic transient dynamic response of two dimensional solids. To accomodate a wide variety of applications, finite strain calculations are incorporated. Spatially, the program employs a four mode isoparametric quadrilateral element. In time, the program performs the integration using a central difference explicit method time integration. Since the time integration scheme is only conditionally stable with respect to step size, the program continuously monitors the step size and adjusts it to keep the calculation stable. The program contains five different material subroutines. In addition, it is very easy to add a new material subroutine. Initial conditions on velocity but not displacement are allowed. Pressure loadings are allowed. Thermal loads are not allowed. The program functions entirely in core, and because no stiffness matrices are calculated and stored, rather large problems are easily accommodated.

HONDO II resulted because new program capabilities and improvements were incorporated into HONDO. The capabilities which have been added are sliding interfaces, a global energy balance caculation, an interface to the mesh-generation program OMESH (another Sandia Code), and a restart feature. Improvements have been made in the numerical procedures for the elastic-plastic constitutive model, and the user may now choose between a constant stress element and a complete bilinear element with four stress states per element. This latter option, while requiring measurably more storage and calculating time, does not suffer from the "keystoning" (element shapes which result in stress free state due to constant stress assumption) which is a byproduct of the constant stress assumption. The more efficient constant stress element may now be used a long as keystoning is not a problem and then, if it becomes a problem, the complete bilinear option can be used. HONDO II can be used to analyze projectile impact problems because of the addition of the sliding interfaces. This will be discussed in more detail later.

Development of Small Scale Impact Simulation Techniques, 1982
Biotech Research Laboratories, Inc, Advanced Technology & Research, Inc.