



AGRICULTURAL TRACTOR OPERATOR PROTECTION

124

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16. Abstract (Limit: 200 words) A mathematical model that would simulate tractor dynamics at the time of impact of the roll/over protection system (ROPS) frame with the soil was developed. The model of tractor motion which was developed was able to predict the velocity, positions and energies of the tractor system throughout a complete overturn of the tractor. The model was usable for the condition of a tractor traveling over an embankment or on the ASAE standard S306.3 overturn test course. Data were gathered from accidental overturns and the position of the tractor, and energy in the system at the time of the ROPS impact were established. Interactive computer graphics displays of the motion and ROPS impact were developed. The computer simulation was verified in model studies. Other objectives of the research were to develop a similitude analysis of tractor and soil interaction during the roll/over of a tractor to the rear and to the side, to conduct scale model studies and measurements to verify the similitude analysis of the behavior of the tractor/ROPS system as it impacts the soil, to conduct a similitude analysis of the performance of a ROPS on an agricultural tractor when subjected to a pendulum test, and to model the pendulum impact test and verify the similitude analysis by experimental measurements on at least two different size models, and to specify a test procedure for ROPS which incorporates modelling and test results to predict the performance of ROPS.					
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Summary Progress Report

A detailed report of the accomplishments of this project is attached to this summary. The reprints and manuscripts submitted also contain complete documentation of the results of this investigation. The following gives a brief summary of the progress of the project and results as they pertain to each of the original objectives.

Objective 1. To develop a mathematical model that simulates tractor dynamics at the time of impact of the protective frame with the soil.

A mathematical model of tractor motion was developed that can predict the velocity, positions and energies of the tractor system throughout a complete overturn of the tractor. This mathematical model is general enough to permit the simulation of tractor motion over an embankment and in an ASAE standard overturn test. Various steering motions are permitted and the model has the potential to simulate the influence of external forces and moments on the motion of a tractor during an overturn. The model has been verified by experimental model studies and further checked by using the model to simulate documented accidental overturns. Although all features of the mathematical model have been reduced to a computer program solution, not all features have been tested. Therefore, the effects of braking and engine performance on the tractor motion have not been investigated.

A mathematical model capable of simulating the motion of an articulated four wheel drive agricultural tractor has also been developed. This model in many respects mimics the previous model but eventually could permit the study of 4-wheel drive agricultural tractor overturns. Limited applications of this model have been developed.

A computer graphics display of agricultural tractor overturns was programmed and has been used to study the position of the tractor at the time of impact of the ROPS with the surrounding surface. This display technique is essential to the interpretation of the results of the mathematical model of tractor motion. Because

this display is interactively controlled by the computer operator we are able to to view the tractor in true three dimensional perspective from any viewing position around the tractor. Therefore, we can precisely determine when and where the ROPS contacts the surface and then establish the tractor velocity and energy at the contact time.

Results - Objective 1

1. We have developed and programmed a mathematical model of agricultural tractor motion where the tractor travels over an embankment or an ASAE standard S306.3 overturn test course.
2. Accidental overturns have been studied and the position of the tractor and energy in the system at the time of ROPS impact have been established.
3. Interactive computer graphics displays of the tractor motion and ROPS impact with the surrounding surface have been developed and the precise point of impact has been defined for several typical overturn situations.
4. The computer simulation of tractor motion has only been verified in model studies. Full-sized verification of the computer and mathematical model of tractor motion have not been completed.

Objective 2. To develop a similitude analysis of tractor and soil interaction during the rollover of a tractor to the rear and to the side.

A similitude analysis was completely defined to model the side overturn test according to the ASAE Standard S306.3. Although the analysis was based on the standard overturn it was general enough to accommodate other types of side overturns.

The analysis was divided into two components dealing with the tractor overturning behavior and with the deformation of ROPS upon impact with the soil surface. Development of similitude relationships for the tractor motion did not pose any technical difficulty. The parameters included were geometry, weight, and location of center of gravity for the tractor body, the front end and the rear wheels. Also included in the analysis were tire damping, stiffness and frictional characteristics. The

analysis pertained to free wheeling of tractor over a specific terrain and no provision was made to include braking (either directly or through the engine), steering or engine characteristics.

The second component of the overall similitude analysis dealt with the deformation behavior of the protective frame upon impact with the soil surface. Keeping in view the various soil conditions that are encountered during rollover accidents, it was decided to model a non-energy absorbing surface. The scale factor for force was determined by the relationship for the tractor weight which was the 3rd power of the geometric scale factor. This predefined constraint on the design conditions for model ROPS resulted in the distortion of design conditions related to the ROPS material strength properties if the prototype ROPS material was used to build the models. This problem was circumvented by using compensated distortion analysis. This analysis was developed by considering theory of elastic beams. The analysis was extended to include plastic behavior of ROPS by including the material yield stress. The analysis successfully created relationships that permitted the use of solid cross section models to represent hollow thin walled section ROPS.

The complete analysis resulted in a set of design equations to build models of the tractor, overturn ramp, ROPS, and set speed and orientation of tractor during the overturn test. It also resulted in the prediction equations to predict from models results, prototype tractor motion and ROPS deformation due to overturn impacts. Although rearward overturn was not considered separately in the analysis, the scaling relationships apply equally well to this situation.

Results - Objective 2

1. Similitude relationships for modelling agricultural tractor overturns in a side overturn were developed. The design equations provided the relationships necessary to design model tractor and the prediction equations could be used to predict motion of the full scale tractor during an overturn.

2. A similitude analysis was developed that permitted prediction of permanent ROPS deflection for prototype frames where compensated distortion model theory provided the basis for the predicted ROPS behavior.

Objective 3. To conduct scale model studies and measurements to verify the similitude analysis of the behavior of the tractor - ROPS system as it impacts the soil.

Two scale model tractors and two scale model overturn test courses were constructed and the similar motion of the tractors over these test courses was verified by recording it on two simultaneous high speed film cameras. These 1/8 and 1/4 scale model tractors were dynamically as well as geometrically similar. Five model ROPS frames were constructed of both welded and bolted designs. These model frames were tested by model tractor overturns on the model test courses. The 1/8 scale ROPS frame deflections were used to predict the permanent deflections of the prototype frame. The predicted prototype deflections for the bolted frame designs were better than those for the welded designs.

Predicted deflections ranged from 0.2% to 61% less than the measured values of prototype deflection. In the case where the predicted deflection was 61% less than the measured, close inspection of the model frame revealed a substantial modification in frame strength because of the effects of welding. Careful attention to construction in the other frames maintained the error in the range from 0.2% to 13.5% underprediction for permanent deflection of the prototype frame.

Results - Objective 3

1. Model tests of tractor-ROPS overturns on a standard test course predicted prototype permanent ROPS deflections within a 13.5% error when effects of welding in the model were carefully controlled.
2. Control of the effects of welding joints in 1/8 scale models was difficult and resulted in up to a 61% underprediction of permanent deformation in the prototype.

Objective 4. To conduct a similitude analysis of the performance of a ROPS on an agricultural tractor when subjected to a pendulum test.

The similitude analysis that was developed for a pendulum impact on structure included the influence of structural geometry, pendulum geometry, operating parameters for the pendulum and the properties of the material of the structure. The basic design conditions for the modelling of a pendulum impact were established and the analysis showed the requirements of a model pendulum impact test for the ROPS. A semi-theoretical analysis of the pendulum impact on the ROPS by Ashburner was modified to correspond to the similitude analysis and the insight provided by the theoretical analysis helped to define the effect of strain rate on deflection of the structure. The theoretical analysis provided a basis for defining the component relationships between the dependent (deflection related) pi term of the similitude analysis and the independent pi terms defining the influence of the various variables on the structural deflection.

Results - Objective 4

1. A similitude analysis was completed that included not only definition of the geometric and operational behavior of pendulum impact on a structure, but also included the influence of strain rate effects in the structural material.

Objective 5. To model the pendulum impact test and verify the similitude analysis by experimental measurements on at least two different size models.

Simple cantilever beams models were constructed to verify the similitude analysis of pendulum impact as discussed in Objective 4. Twenty different impact tests were completed according to an experimental schedule designed to provide data for component equations describing beam deflections as functions of the individual pi terms. An instrumented pendulum and test fixture were constructed to accomplish the above tests and the experimental procedures for determining the influence of distortion of the design conditions was developed.

Results - Objective 5

1. In the similitude modelling of large elastic and plastic deformation of structures made of strain rate sensitive materials, it was not possible to build a true model. The design conditions related to the strain rate property of the material was distorted.
2. The effects of rate of strain upon the predicted permanent and maximum deflection was small. A 400 percent increase in the rate of strain caused only a 14.9 percent increase in permanent deflection and 19 percent change in maximum deflection.
3. A prediction factor was developed, based on distorted model theory, to calculate the effect of the strain rate sensitive design condition upon permanent and maximum deflection.
4. Analytically determined deflections were in good agreement with experimentally determined values except for the elastic component of total deflections.

Objective 6. To specify a test procedure for ROPS which incorporates modelling and test results to predict the performance of ROPS.

Similitude analysis and modelling theory were developed to predict the deformation of ROPS in the ASAE S306.3 side overturn test and the standard quasi-static test of ROPS performance. One eighth and 1/4 scale model tractors and overturn test courses were built and the deflections of several model (1/8) and prototype (1/4) ROPS were measured and the predicted deflections were compared with the measured ROPS deflections. The procedure that was developed gives the designer of ROPS the technique and instructions to build the model test tractors and the design techniques and procedures for testing the model ROPS frames. We also give the specific steps for predicting frame deformations from the experimental data by accounting for strain rate effects and the nature of the plastic deformation of the ROPS.

The standard quasi-static test of ROPS was also modelled and performance of 1/8 and 1/4 scale frames was measured and experimentally determined. We verified that compensated distortion models and dissimilar model cross sections could be used to predict prototype behavior. In fact we were able to show an excellent prediction of full size ROPS deflections using this modelling approach. Step by step design procedures for designing a model ROPS were developed and applied to the design of one manufacturer's roll over protective frame. We constructed the model with solid cross sections according to the theory we had developed and our test of the 1/4 scale model gave an excellent prediction of the force-deformation curve for the full scale frame. The full scale frame performance data was independently supplied by the manufacturer.

Results - Objective 6

1. A test procedure was developed that permits modelling the ASAE Standard S306.3 side overturn and the ROPS deflections. Step by step instructions for the model design and ROPS design were created and the procedure for predicting prototype ROPS deflection was established. ROPS deflections for the prototype could be predicted within 25% of the measured value for welded frames and more precisely for bolted frames.
2. A test procedure was developed that permits modelling the ASAE Standard S306.3 quasi-static ROPS frame test. Model ROPS were designed that showed that compensated distortion theory and dissimilar model cross section could be used to accurately predict the force-deformation characteristics of a full scale ROPS.

List of Publications and Manuscripts

1. Davis, D. C. and G. E. Rehkugler. 1975. SIMTRAC - A computer program for simulating tractor motions. Mimeographed manual, Department of Agricultural Engineering, Cornell University, Ithaca, NY 14853.
2. Kelly, J. E. and G. E. Rehkugler. 1976. Computer graphics display of simulated tractor motion. ASAE Paper No. 76-1523 presented at the 1976 Winter Meeting of the American Society of Agricultural Engineers. Palmer House, Chicago, Illinois, Dec. 14-17, 1976. ASAE, St. Joseph, Michigan. 13 pp. (Manuscript to be submitted for publication in Trans. of the ASAE).
3. Rehkugler, G. E., V. Kumar and D. C. Davis. 1976. Simulation of tractor accidents and overturns. Trans. of the ASAE 19(4): 602-609, 613.
4. Srivastava, A. K. and G. E. Rehkugler. 1976. Strain rate effects in similitude modelling of plastic deformations of structures subject to impact loading. Trans. of the ASAE 19(4): 617-621.
5. Srivastava, A. K., G. E. Rehkugler and B. J. Masemore. 1976. Similitude modelling of ROPS overturn tests. ASAE paper No. 76-1524 presented at the 1976 Winter Meeting of the American Society of Agricultural Engineers. Palmer House, Chicago, Illinois, Dec. 14-17, 1976. ASAE, St. Joseph, Michigan. 45 pp.
6. Srivastava, A. K., G. E. Rehkugler and B. J. Masemore. 1977. Similitude modelling applied to ROPS testing. (Manuscript in review for publication in the Trans. of the ASAE). 49 pp.
7. Masemore, B. J. 1977. The relationship between ROPS loading in a side overturn test and tractor weight. M.S. thesis in preparation. (To be completed by 1 August 1977).

SUMMARY REPORT

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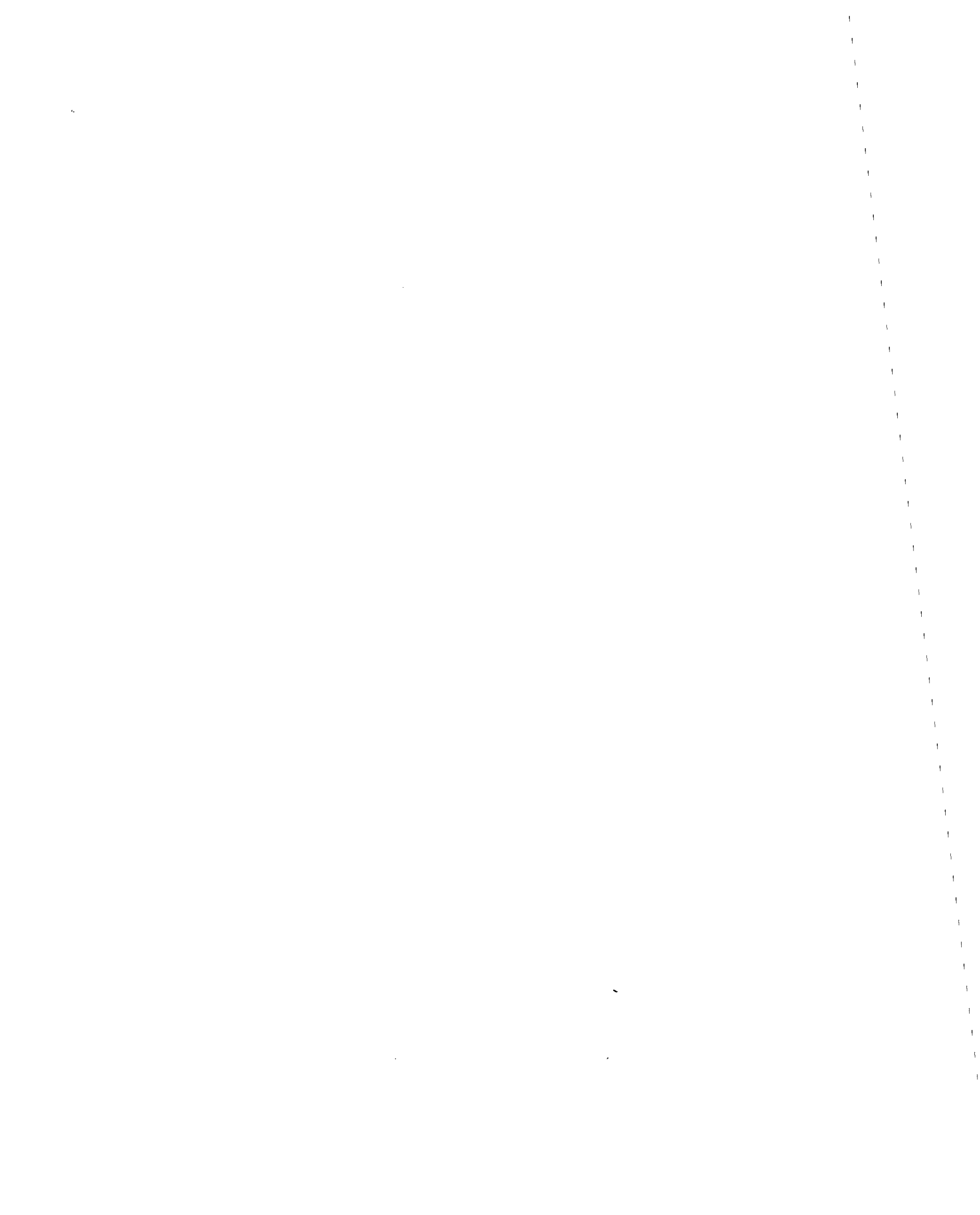


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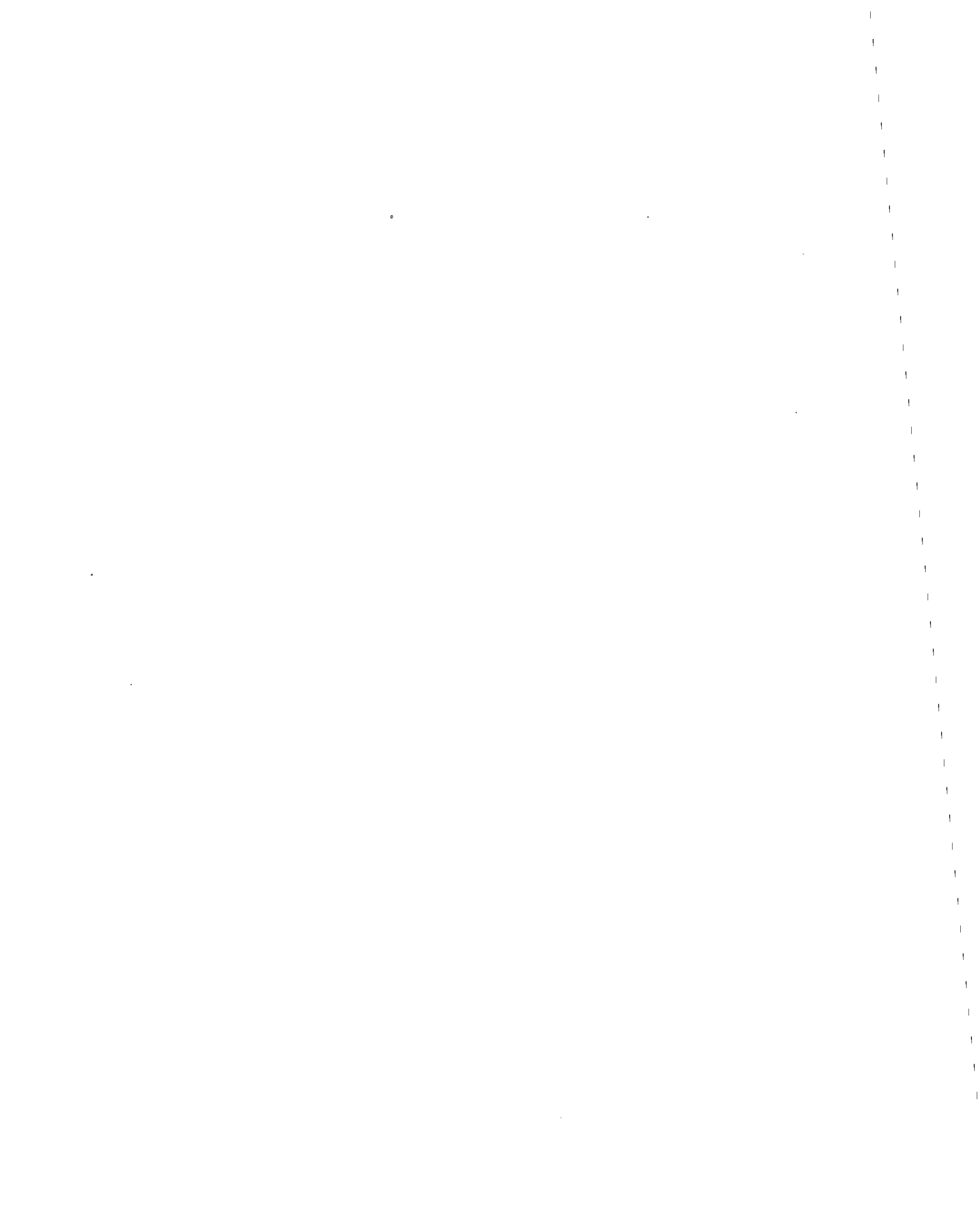
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AGRICULTURAL TRACTOR OPERATOR PROTECTION

INTRODUCTION

Agricultural tractor accidents claim 800 to 1,000 lives per year in the United States. Approximately two-thirds of these deaths result from tractor overturn. Both sideways and rearward overturns contribute to the death toll with one rearward overturn to every three sideways overturn accidents (Volpe, 1971). Variations in the conditions leading to an accident are so large that measures for predicting an overturn and responding in some way to avoid it is probably impossible. This has lead to the development of the protective enclosures and frames which can prevent the tractor operator from being crushed to death or receiving serious injuries in the event of a tractor overturn.

In order for these protective enclosures, referred to as the rollover protection structures (ROPS), to potentially be safe, they must comply to the test standards adopted by the ASAE and SAE (ASAE Yearbook, 1976). Present test procedures are destructive and require full scale testing of the ROPS. Testing with full size frames can be expensive and time consuming.

The standards adopted by the ASAE/SAE are based on the European test standards and some actual full scale field overturns of tractors conducted by some manufacturers. European test standards were developed on the basis of overturn tests in the 1950's with tractors weighing less than 10,00 lbs. whereas the tractors presently used in this country are several times heavier than the average tractor used in the original tests. Actual overturn tests with full size tractors do not generally permit a good control of the test conditions. These standards have been criticized by some as being inadequate (Splinter, 1970 and Steinbruege, 1970). The main area of criticism has been the relationship of the required amount of energy to be absorbed during the standard static or pendulum test to the tractor weight. The relationship was developed in Europe by actual field overturning of tractors of various sizes fitted with experimental frames. This relationship simply has been extrapolated as a

straight line to encompass heavier tractors found in the United States. This relationship needs to be further verified by field testing with heavier tractors. An extensive testing program with high weight full sized tractors would be prohibitively expensive.

In a report prepared by the Secretary of Transportation in 1971 (Volpe, 1971) it was recommended that voluntary use of ROPS should be encouraged for the tractor industry. It was also suggested in the report that the protection structures conform to the standards recommended by the ASAE and the SAE. This project was submitted to develop simplified and less expensive testing procedure for ROPS which could reduce the cost of manufacturer of these systems for all tractors. Reduced testing and development cost could reduce the cost of ROPS to the consumer and hasten the adoption of this safety device. Although, since then, OSHA has made it mandatory that the new tractors manufactured after October 24, 1976 be fitted with ROPS, the advantages of less expensive testing procedures still remain.

Using principles of similitude modelling to evaluate protective enclosures affords a unique technique for evaluating safety equipment and may be extended to other safety equipment. Scale model testing of ROPS is an excellent means of evaluating new designs and saving both cost and time. Model testing can be utilized to study current standards. It can also be employed to conduct parametric studies to develop a greater understanding of protective structure behavior during a rollover. This study could provide further credibility for current test standards.

Very little work has been reported on the attitude and kinetic energy of the farm tractor at the time of impact of the tractor and protective enclosure with its surroundings. Watson (1967) illustrates a theoretical analysis of kinetic energy at impact based on the geometry of the tractor, sideways rollover on a uniform slope and sideways overturning down a bank. He concludes that overturning down an embankment is a more severe test of the protective enclosure than rollover on a uniform slope. He also recognizes that there is a wide range of kinetic energies

and tractor configurations that make it difficult to generalize test procedures developed for one size of tractor to other tractors and safety enclosures.

Simulation of tractor motion can be a powerful tool in determining tractor dynamics for a variety of situations. Davis (1973) has demonstrated the capabilities of a simulation model for defining the general 3-dimensional motion of an agricultural wheel tractor and experimentally verified his model using a 1/12 scale tractor model. This encourages the use of the simulation model for full size tractor overturns because the mathematical simulation model and computer program have proven performance not only mathematically but also experimentally.

A parametric study of tractor motion can be valuable in determining the influence of various input data parameters on the tractor motion during an overturn and provides a learning experience without creating a physical model of the vehicle. Among the things that can be learned are the influence of the vehicle to surface relationships (traction parameters) upon tractor motion and the amount of energy in the vehicle at impact of the rollover protective structure (ROPS) with the terrain.

Simulation of tractor motion can give the data necessary for determining the loading on ROPS in an accidental overturn. Study of influence of steering changes on tractor motion can be used to learn appropriate steering action to avoid an imminent accident. This may further supply interactive displays of tractor motion such that one can give training to tractor operators and/or study the influence of design and operation changes on tractor stability. The simulation studies provide data for evaluation of load direction and magnitude to be applied to model ROPS structures and relates to a dynamic structural analysis.

OBJECTIVES

The objectives of the research was to provide a test procedure for evaluating the performance of agricultural tractor operator rollover protection system that circumvents full scale testing for all systems and tractor combinations. Specific objectives were:

1. To develop a mathematical model that simulates tractor dynamics at the time of impact of the protective frame with the soil.
2. To develop a similitude analysis of tractor-ROPS and soil interaction during rollover of a tractor to the rear and to the side.
3. To conduct scale model studies and measurements to verify the similitude analysis of the behavior of the tractor-ROPS system as it impacts the soil.
4. To conduct a similitude analysis of the performance of a ROPS on an agricultural tractor when subjected to pendulum test.
5. To model the pendulum impact test and verify the similitude analysis by experimental measurements on at least two different size models.
6. To specify a test procedure for ROPS which incorporates modelling and test results to predict the performance of ROPS.

COMPUTER SIMULATION OF TRACTOR MOTION

Simulation of agricultural tractor accidents and overturns contributes to giving the necessary data to determine the expected loading of the ROPS as a result of an overturn. In addition, the simulation work gives a better understanding of tractor motion dynamics and the relationships of terrain, operator steering response, tractor design and tire-surface characteristics to tractor stability and the type of overturning accidents that can occur. The objectives of the computer simulation studies of agricultural tractor overturns were to:

1. Demonstrate the ability of SIMTRAC (Davis and Rehkgler, 1975) to simulate ASAE Standard S306.3 side overturns and accidental side overturns of full sized tractors.
2. Evaluate the influence of surface-tire parameters on tractor overturns.
3. Relate simulated overturns to documented accidental overturns.
4. Determine the attitude, velocity and energy in the tractor at the time of impact of the ROPS with the surrounding surface.
5. Provide visualization of the overturn so that the interaction of the ROPS with the surrounding terrain could be observed.
6. Consider the influence of steering parameters on the motion of the tractor during overturn.
7. Evaluate the adequacy of a linear relationship between tractor weight and energy to be absorbed by the ROPS as prescribed by present standards for agricultural tractor ROPS.

Simulation and Results

The motion of a 4600 kg two wheel drive tractor was simulated in overturns on concrete and soil embankments with several types of steering actions, bank heights and with and without overturning ramps (Rehkgler, et al., 1975 and Kelly and Rehkgler, 1976). Some of the results of these simulations may be summarized under categories of steering effects and tire-surface characteristics effects. (Detailed results are given in the papers referenced above).

Steering Effects

We were able to demonstrate that under some circumstances that steering downhill over the embankment could prevent overturning of the vehicle. However, in some cases the steering action had relatively little influence on tractor behavior. This occurred primarily when the surface was concrete. A concrete surface was unable to provide large lateral tire forces to effect a reorientation of the tractor.

Turning and skidding on any terrain results in a large loss of kinetic energy in the tractor system. This is beneficial in the sense that it significantly reduces the energy that the ROPS must absorb when and if the overturn occurs.

Tire-Surface and Terrain Characteristic Effects

The concrete surface did not provide as large lateral tire forces as the soil surface. Therefore, the tractor tended to be more stable on the concrete surface used in some cases where identical terrain geometry features and steering conditions were used. The tractor operating on concrete did not overturn. Because the concrete surface did not provide a large amount of rolling resistance, the kinetic energy of the tractor decreased at a slower rate as compared to operation on a soil surface.

The ASAE Standard S306.3 side overturn test simulation resulted in much higher energies in the tractor at the time of impact of the ROPS with the surface as compared to the accidental overturns simulated. The ASAE Standard side overturn results in very little skidding and thus very little energy is dissipated before the ROPS strikes the surface. The roll velocity, however, tends to be lower which may otherwise result in a smaller loading of the ROPS.

ROPS Loading from Simulation Results

The simulations plus the graphics displays that were developed allowed a study of the interaction of the ROPS with the surrounding surface. We were able to define the point of contact of the ROPS with the surrounding surface and the velocity and orientation of that point on the ROPS relative to the surrounding surface. The

orientation of the vehicle as well as the kinetic energy, and velocities of the vehicle also permit an estimate of the direction of application of forces to the ROPS. The ASAE Standard side overturn gave a different ROPS impact point and loading as compared to the accidental overturn that was modelled. For example, in the accidental overturn the left rear ROPS was the first point to contact the surface but in the ASAE Standard overturn the left front ROPS was the first point to contact the surface.

In many instances it was difficult to visualize the tractor orientation just from the simulation data and simple line plots of the accidents and overturns. However, we were able to develop 3D computerized graphics displays that gave extensive capabilities in viewing the overturns and tractor motion (Kelly and Rehkugler, 1976). The interactive graphics display allowed us to precisely determine the point of impact and to tabulate the force, velocity, and energy data for the vehicle at that point.

Tractor Size and ROPS Loading

Current standards (ASAE S306.3) indicate that the ROPS loading energy should be a linear function of the tractor weight. We have completed a preliminary investigation of this relationship by examining a simple two dimensional simulation of a tractor sideways overturn. We have experimentally obtained the mass moments of inertia and geometry of tractors ranging in sizes from 1800 kg to over 8000 kg. The hypothesis that the energy to be absorbed by the ROPS should be a linear function of tractor weight has been studied with the simulation model plus the experimental data. Preliminary results indicate that the energy to be absorbed by the ROPS should increase in a non-linear fashion and it is evident that the nature of the ROPS failure should influence the adequacy of the specified standard energy absorption by the ROPS. Figure 1 illustrates this behavior for three types of failure for the ROPS. Note the energy to be absorbed by the ROPS versus relative to the standard energy, increases with tractor size. The preliminary results also show that

tractor and ROPS geometry will also have a significant influence on the energy that must be absorbed by the ROPS in a side overturn.

SIMILITUDE MODELLING

Theory of models as presented by Murphy (1970) is briefly reviewed here.

Consider the following prototype system:

$$\pi_1 = \phi(\pi_2, \pi_3, \pi_4, \dots \pi_n) \quad (1)$$

This equation shows a general relationship of dependent dimensionless variables, π_1 , to the independent dimensionless system variables, $\pi_2, \pi_3, \dots \pi_n$. Since this equation is entirely general, it applies to any other system which is a function of the same variables. Hence, it applies to a specific system called the 'model'.

$$\pi_{1m} = \phi(\pi_{2m}, \pi_{3m}, \pi_{4m}, \dots \pi_{nm}) \quad (2)$$

An equation of predicting π_1 from π_{1m} may be found directly by dividing Eq. (1) by Eq. (2),

$$\frac{\pi_1}{\pi_{1m}} = \frac{\phi(\pi_2, \pi_3, \pi_4, \dots \pi_n)}{\phi(\pi_{2m}, \pi_{3m}, \pi_{4m}, \dots \pi_{nm})} \quad (3)$$

Now, if the model is designed and operated such that:

$$\begin{aligned} \pi_{2m} &= \pi_2 \\ \pi_{3m} &= \pi_3 \\ \pi_{4m} &= \pi_4 \\ &\vdots \\ \pi_{nm} &= \pi_n \end{aligned} \quad (4)$$

it follows that:

$$\phi(\pi_2, \pi_3, \pi_4, \dots \pi_n) = \phi(\pi_{2m}, \pi_{3m}, \pi_{4m}, \dots \pi_{nm}) \quad (5)$$

The nature of the function is identical for model and prototype because Eq. (1) is general. It, therefore, follows that

$$\pi_1 = \pi_{1m} \quad (6)$$

Eq. (6) constitutes the prediction equation which must be valid if the design and operating conditions, Eq. (4), are satisfied. In general, the following steps are taken in completing a model study.

- (a) Identification of pertinent dimensionless variables and development of design conditions.
- (b) Design and fabrication of the model system and determination of appropriate experimental techniques.
- (c) Collection of appropriate experimental data.
- (d) Analysis and interpretation of results.

Frequently it is impossible to satisfy all the design conditions in a model system. The proper material may not be available, details may impose fabrication problems and available equipment may impose limitations precluding the possibility of meeting all of the design conditions. A model which does not meet one or more of the design conditions is by definition called a distorted model.

Type of distortion depends upon the nature of the pi-term or terms that are not satisfied by the model system. Regardless whether the distortion is due to geometrical loading or material scaling, there are generally two ways to account for this problem.

- (a) Determination of prediction factor, δ , so that

$$\pi_1 = \delta \pi_{1m} \quad (7)$$

- (b) Introducing controlled distortion of additional pi-terms so that,

$$\pi_1 = \pi_{1m}$$

The second method is called the compensated distortion analysis. In compensated distortion analysis if distortion is present in two design conditions they are controlled, if possible, to compensate each other. In the similitude modelling of ROPS standard testing both of the above methods were employed to account for distortion in the various pi-terms as will be shown later.

PREVIOUS WORK

Application of principles of similitude modelling to the testing of ROPS requires modelling considerations of large elastic-plastic deformation of structures due to static or dynamic loadings. Substantial amount of work has been reported in predicting elastic deformations due to the static or dynamic loadings on structures. Such literature will not be reviewed here. Research reported in the area of modelling of large elastic-plastic deformation of structure will be mentioned here.

Brown et al. (1974) reported results of structural modelling of ROPS under dynamic loading. They modelled the dynamic portion of the SAE J3342, Protective Frame Test Procedure and Performance Requirements. They conducted two analyses and derived two sets of design conditions. In the first analysis, rate of strain was higher in the model than the prototype by a factor of square root of the length scale factor for the same material. In the second analysis the strain rate in the model was kept equal to that in the prototype and the mass scale was equal to the geometric scale factor. The second analysis resulted in better predictions from model results. This approach was not applicable to model tractor overturn accident because of the limitation in mass scaling.

Ashburner (1972) employed similitude modelling to determine frame stiffness and collapse load which was used to predict frame deformation due to pendulum impact. He did not, however, have the same constraints on the force scaling as that of the tractor overturn situation. His approach can be applied to the static testing of ROPS but could not be applied for the overturn field testing.

Holmes and Colton (1973) demonstrated the use of scale model experiments for safety car development. They were interested in determining the response of a safety car bumper when impacted against a rigid pole in head-on accidents. The modelling included dynamic plastic deformations of the bumper. The strain rate effects were initially included in the analysis, however, the related design conditions were not

satisfied by the model. It was indicated that the contribution due to the elastic modulus was much greater as compared to the strain rate coefficient. Therefore, the design condition related to the strain rate was ignored. It was also indicated that the high speed alloys used in the study did not exhibit strong strain rate dependence. The design condition related to gravitational acceleration was satisfied but was considered unimportant because the major forces on the model were much larger than the vehicle weight and were in the plane perpendicular to the direction of the gravitational forces. Their approach could not be directly applied to the problem of tractor ROPS overturn test because the gravity effects could not be overlooked.

Static Test

This section is concerned with developing similitude modelling of the static test part of the standard ROPS testing procedure and performance requirements. The static test is optional to the dynamic test procedure. In the static test procedure the protective frame is mounted on the laboratory mounting base which may be the tractor chassis (or the equivalent) for which the protective frame is designed. This mounting procedure is to assure the integrity of the entire system. The protective frame is instrumented with the necessary instrumentation to obtain the required load deflection data at a location and in the direction specified by Figures 2 and 3. If the protective frame is a one or two upright design, mounting connections are instrumented with the necessary equipment to record the required force to be used to calculate the input energy, E_{ir} . Instrumentation is placed on mounting connections before the installation load is applied.

A rear load is applied first and L and d are recorded simultaneously. The rear load is applied according to the details included in Appendix A. The test is stopped when,

- (1) the strain energy absorbed by the frame is equal to or greater than the required input energy, E_{ir} or;
- (2) deflection of the frame exceeds the allowable deflection or;

- (3) frame load limit occurs before the allowable deflection is reached in the rear load.

Using the data obtained from the L-D diagram as shown by example in Fig. 3 the following quantities are calculated. (ASAE S306.3)

$$E_{ir} = 0.47 W \quad (E'_{ir} = 0.14 W') \quad (8)$$

$$FER_{ir} = \frac{E_u}{E_{ir}} \quad (9)$$

$$FSB = \frac{P_u}{P_b} \quad (10)$$

A side load is applied on the same frame and L and D are recorded simultaneously. Side load application is at the upper extremity of the frame at a 90° angle to the longitudinal side farthest from the point of rear load application. Loading is stopped when,

- (1) strain energy absorbed by the frame is equal to or greater than the required input energy E_{is} or;
- (2) deflection of frame exceeds the allowable deflection or;
- (3) the frame load limit occurs before the allowable deflection is reached in side loading.

Using the data obtained from the L-d diagram as shown by example in Fig. 3 the following quantities are computed.

$$E_{is} = 723 + 0.4 W \quad (E'_{is} = 100 + 0.12 W') \quad (11)$$

$$FER_{is} = \frac{E_u}{E_{is}} \quad (12)$$

$$FSB = \frac{P_u}{P_b} \quad (13)$$

In order to perform the previously described test using scale models, one must identify all of the pertinent system variables, establish design conditions and develop scaling laws. Table 1 lists the pertinent system variables. Since the static testing involves large elastic-plastic deformations, the yield stress has

been included in the analysis. The system variables were used to formulate the following non-dimensional pi-terms:

$$\phi \left\{ \frac{d}{\ell}, \frac{\lambda}{\ell}, \frac{I_i}{\ell^4}, \frac{Z_i}{\ell^3}, \frac{L}{E\ell^2}, \frac{\sigma_o}{E}, \frac{E}{G}, \frac{\tau_o}{G} \right\} = 1 \quad (14)$$

Either $\frac{D}{\ell}$ or $\frac{L}{E\ell^2}$ can be chosen as the dependent pi-term and expressed as a function of the others. These pi-terms were used to develop model design conditions.

If,

$$\left(\frac{\lambda}{\ell} \right)_m = \left(\frac{\lambda}{\ell} \right) ; \quad \left(\frac{I_i}{\ell^4} \right)_m = \left(\frac{I_i}{\ell^4} \right)$$

Table 1. List of pertinent variables for static ROPS test.

Variable	Symbol	Dimension*
Deflection	d	L
Load	L	F
Characteristic length	ℓ	L
Frame geometry	λ	L
Area moment of inertia	I_i	L^4
Section modulus	Z_i	L^3
Modulus of elasticity	E_i	FL-2
Modulus of rigidity	G	FL-2
Tensile yield stress	σ_o	FL-2
Shear yield stress	τ_o	FL-2

*F - Force
L - Length

$$\left(\frac{Z_i}{\ell^3} \right)_m = \left(\frac{Z_i}{\ell^3} \right) ; \quad \left(\frac{L}{E\ell^2} \right)_m = \left(\frac{L}{E\ell^2} \right) ;$$

$$\left(\frac{\sigma_o}{E} \right)_m = \left(\frac{\sigma_o}{E} \right) ; \quad \left(\frac{E}{G} \right)_m = \left(\frac{E}{G} \right) ;$$

and

$$\left(\frac{\tau_o}{G} \right)_m = \left(\frac{\tau_o}{G} \right)$$

then

$$\left(\frac{d}{\ell} \right) = \left(\frac{d}{\ell} \right)_m$$

If the length scale factor, $n = \frac{\ell}{\ell_m}$, then with the above set of design condition various scaling relationships were developed and are given as follows.

$$\begin{aligned}\lambda_m &= \lambda/n \\ I_{im} &= I_i/n^4 \\ Z_{im} &= Z_i/n^3 \\ L_m &= L/n^2 \\ \sigma_{om} &= \sigma_o \\ E_m &= E \\ \tau_{om} &= \tau_o \\ \sigma_m &= \sigma\end{aligned}$$

The last four scaling relationships are obtained by assuming that the same material (c.g., cold rolled mild steel) was used to build the model ROPS as the full size one. This is commonly done in structural modelling to satisfy the strict requirements of the material constitutive relations. It can be seen that no difficulty is encountered in satisfying the scaling requirements. The prototype deflection can be predicted by the following equation:

$$d = nd_m \tag{15a}$$

The force can be predicted by the following equation:

$$P = n^2 P_m \tag{15b}$$

Eq. (15a) and (15b) can be used to construct the load-deformation diagram for the full scale ROPS from the results obtained by testing its model. Therefore, the equations constitute the prediction equations.

The above analysis extends to the frame mounting details also. The load deformation characteristics of the chassis are included in the model design because it is considered to be rigid.

Pendulum Test

In the pendulum test of the standard ROPS testing procedure the loading is produced by use of a 4410 lb (2000 kg) weight acting as a pendulum. The square impact face of the weight is 27 $\pm$ 1 inch by 27 $\pm$ 1 inch and the weight is constructed such that its center of gravity is within 1 inch of its geometric center. This weight is

suspended from a pivot point 18 to 22 ft above the point of impact on the frame and is conveniently and safely adjustable for height. A diagram of the pendulum is shown in Figure 4. Details of the tractor adjustments and tie-downs are given in ASAE S306.3 Figures 5 and 6 show a tractor and ROPS in the rear and side impact mode, respectively. The pendulum is raised a predetermined height before impacting the protective frame according to the following equation,

$$H(\text{in}) = 4.92 + 0.0019W(\text{lb}) \text{ or } (H'(\text{cm}) = 125 + 0.107W'(\text{kg})) \quad (16)$$

Impact load is applied at the rear followed by an impact at the side. Both maximum instantaneous and permanent plastic deformation are recorded during the test. These deformations must meet the performance criterion specified in the standards.

It must be noted that compared to the static test, loading is dynamic which results in dynamic plastic deformations. Most ROPS are made of mild steel which is strain rate sensitive, i.e., the yield stress is a function of the rate of loading. Figure 7 shows a simplified diagram to illustrate the effects of the rate of strain. Significance of this phenomena has been realized by many researchers working in the area of dynamic plastic deformations of structures. Elevation of dynamic yield stress as a function of rate of strain has been studied experimentally and a model has been developed. This model reported by Perrone (1971) is presented below.

$$\sigma_y = \sigma_o \left\{ 1 + \left(\frac{\dot{\epsilon}}{c} \right)^{\frac{1}{p}} \right\} \quad (17)$$

Values of c and p have been found to be 40.4 per sec and 5, respectively for mild steel. This model requires knowledge of the rate of strain which in many situations is not known explicitly. Further more, the rate of strain is not uniform across a cross-section of a cantilever beam. The following model as reported by Ashburner (1972) does not include strain rate explicitly but instead it requires knowledge of time before yielding.

$$\frac{M_{po}}{M_{pso}} = \left[\frac{t_o}{D} \right]^A \quad (18)$$

Values of D and A are 0.17 sec. and -0.1, respectively for mild steel. He also reported that,

$$\frac{M_{po}}{M_{pso}} = \frac{\sigma_y}{\sigma_o} \quad \text{if} \quad \frac{M_{po}}{M_{pso}} < 1.5 \quad (19)$$

Obviously, Equations (18) and (19) have limitations, but they do provide a good estimate for most of the conditions encountered in pendulum tests of ROPS. The model used in this study consisted of Equations (18) and (19) because an analytical verification of the strain-rate effects was sought from the mathematical model developed by Ashburner for predicting ROPS deformation.

Dimensional Analysis

In the case of dynamic loading, the static yield stress was replaced by dynamic yield stress. The dynamic yield stress, as indicated earlier is a function of static yield stress and strain rate parameters D and A. This resulted in the following pi-terms related to the rate of strain, A and $\frac{\ell}{D^2 g}$. These terms imposed additional design parameters.

$$A_m = A \quad (20)$$

$$\left[\frac{\ell}{D^2 g} \right]_m = \left[\frac{\ell}{D^2 g} \right] \quad (21)$$

The first design condition was satisfied by choosing the same material for the model as well as the full scale frame. However, the second design condition could not be satisfied and was distorted.

It was decided to study effects of such distortion before developing a similitude analysis for dynamic loading. The effects of distorting the pertinent design condition were studied by dynamically bending cantilever beams as struck by a pendulum. Table 2 lists all variable pertinent to the system described previously. The variables in Table 2 can be grouped into dimensionless groups and are presented in the following generalized prediction equation.

$$\frac{d}{\ell} = \phi \left\{ \frac{b}{\ell}, \frac{h}{\ell}, \frac{\ell_i}{\ell}, \frac{s}{\ell}, \frac{E}{\sigma_o}, \frac{\ell}{D^2 g}, A, \frac{E \ell^2}{M g}, \frac{v^2}{g \ell}, \phi, \frac{g t}{V} \right\} \quad (22)$$

In this study only the maximum and permanent deflections, independent of time, were measured. Time, t , (and gt/v), therefore, were excluded from the analysis. The pi-terms held constant were E/σ_0 , A , s/ℓ , ℓ_1/ℓ and ϕ . The following equation expresses the pi-terms that were varied during the experimental investigations:

$$\frac{d}{\ell} = \phi \left\{ \frac{b}{\ell}, \frac{h}{\ell}, \frac{\ell}{D^2 g}, \frac{E\ell^2}{Mg}, \frac{v^2}{g\ell} \right\} \quad (23)$$

The term $\frac{\ell}{D^2 g}$ contains a strain rate variable and represents a property of the material. Change in the value of this term can be brought about by changing the value of D . But a change in this value corresponds to a change in the right hand side of equation (18) which in turn results in a change in yield stress according to the same equation. A similar change can be caused by changing time to yielding, which corresponds to changing the rate of strain. Therefore, by studying the effects of changing $\ell/D^2 g$ on the dependent π -term, sufficient information can be obtained to determine strain rate effects in similitude modelling of rapidly loaded ROPS.

Experimental Analysis

A compound (rigid arm) pendulum was designed to apply impact force to test beams. The suspension length and the weight of the pendulum could be changed to achieve desired values of the system non-dimensional variables. The height of the suspension point of the pendulum was adjustable. Cantilever beams were held vertically in a vise which was rigidly bolted to a common heavy steel base plate (Fig. 8). A semi-cylindrical rod, mounted horizontally across the face of the pendulum block at the point of center of percussion, was used to strike the beams. A solenoid actuated mechanism was used to hold the pendulum at a predetermined angle until it was released (Fig. 9). An angular displacement transducer was mounted at the pivot axis and continuously recorded the position of the pendulum during a test. Figure 10 shows the method of adding weights to the pendulum. Test beams were made of cold rolled mild steel and were machined to the desired sizes. Figure 11 shows the various cylindrical impact pieces and the beams used during the investigation. The



beams were annealed by heating them to 900°C and then leaving them in the furnace to cool.

The test beams were built and tested according to the experimental schedule shown in Table 3. One pi-term was varied while the others were kept constant. Table 4 presents the values of the dimensional system variables corresponding to the experimental schedule. Permanent plastic and maximum instantaneous deflections of the cantilever beams were measured. Figure 12 shows a typically bent beam after being struck by the pendulum.

Results and Discussion

Experimental results are shown in Fig. 13 through 17. These figures present both the dependent π -terms representing the maximum and permanent deflections. Effects of (b/ℓ) are shown in Fig. 13 and Fig. 14 shows the effects of changing (h/b) . These π -terms are related to section modulus of the beams and an increase in these π -terms is related to increase in the value of section modulus which causes a rapid decline in the deflection values. Regression equations were fitted to the data and are shown in the respective figures. Figure 15 shows the effect of changing (ℓ/D^2g) . It can be seen that increasing the value of this variable causes a slight increase in both maximum and permanent deflections. Increase in the maximum deflection is greater than that of the permanent deflection. It was also seen that data for maximum deflection show a greater scatter. This might have been due to the indirect method of measuring the maximum deflection. A 400 percent increase in the value of ℓ/D^2g causes a 1.48 percent in dimensionless permanent deflection and a 19.1 percent increase in the value of the maximum dimensionless deflection.

The theory of similitude modelling was applied to the system represented by Equation (23) and the following design conditions were established.

$$b_m = \left[\frac{\ell_m}{\ell} \right] b \quad (24)$$

$$h_m = \left[\frac{b_m}{b} \right] h \quad (25)$$

$$D_m = \left[\frac{\ell_m}{\ell} \right] \left[\frac{g}{g_m} \right]^{\frac{1}{2}} D \quad (26)$$

$$M_m = \left[\frac{E_m}{E} \right] \left[\frac{\ell_m}{\ell} \right]^2 \left[\frac{g}{g_m} \right] M \quad (27)$$

$$V_m = \left[\frac{g_m}{g} \right] \left[\frac{\ell_m}{\ell} \right]^{\frac{1}{2}} V \quad (28)$$

For $E_m = E$ and $D_m = D$ (obtained by choosing same material for model as the prototype) and $g_m = g$ (for the same gravitational field) all of the above design conditions were satisfied except for Equation (26). Distortion factor, α , was obtained as follows:

$$\alpha = \left[\frac{\ell_m / D_m^2 g_m}{\ell / D^2 g} \right]$$

or,

$$\alpha = \frac{\ell_m}{\ell} \quad (29)$$

Regarding the beam of the largest dimension as the prototype and other as models of different scale factors, prediction factor, δ , was developed.

$$\delta = \frac{(d/\ell)}{(d/\ell)_m} \quad (30)$$

Figure 18 illustrates a plot of the prediction factor, δ and the geometric scale factor, n , for both dimensionless maximum and permanent deflections. It can be interpreted from this plot that if the strain rate effects are disregarded in structural modelling of dynamic plastic deflections then a maximum prediction error of 13 for permanent and 16 percent for maximum deflection could be encountered in predicting prototype behavior from a 1/4 scale model. However, correction in predicting results could be made by using the prediction factors illustrated in Figure 18. It must however, be noted that Figure 18 presents the effects of distorting a certain pi-term which can be caused by any of the variables involved in that pi-term.

Error Analysis

Before accurate conclusions could be drawn, it was necessary to estimate the experimental error. The error in determining maximum and permanent dimensionless deflections is contributed by the error in each of the system variables. Equation (23) is written in a generalized form as follows.

$$\pi_1 = \phi (\pi_2, \pi_3, \pi_4, \pi_5, \pi_6) \quad (31)$$

$$\Delta \pi_1 = \frac{\partial \phi}{\partial \pi_2} \Delta \pi_2 + \frac{\partial \phi}{\partial \pi_3} \Delta \pi_3 + \frac{\partial \phi}{\partial \pi_4} \Delta \pi_4 + \frac{\partial \phi}{\partial \pi_5} \Delta \pi_5 + \frac{\partial \phi}{\partial \pi_6} \Delta \pi_6 \quad (32)$$

$$\pi_2 = \frac{b}{\ell}$$

$$\Delta \pi_2 = \frac{\partial \pi_2}{\partial b} \Delta b + \frac{\partial \pi_2}{\partial \ell} \Delta \ell \quad (33)$$

$$\pi_3 = \frac{h}{b}$$

$$\Delta \pi_3 = \frac{\partial \pi_3}{\partial h} \Delta h + \frac{\partial \pi_3}{\partial b} \Delta b \quad (34)$$

$$\pi_4 = \frac{\ell}{D^2 g}$$

$$\Delta \pi_4 = \frac{\partial \pi_4}{\partial \ell} \Delta \ell + \frac{\partial \pi_4}{\partial D} \Delta D + \frac{\partial \pi_4}{\partial g} \Delta g \quad (35)$$

$$\pi_5 = \frac{E \ell^2}{mg}$$

$$\Delta \pi_5 = \frac{\partial \pi_5}{\partial E} \Delta E + \frac{\partial \pi_5}{\partial \ell} \Delta \ell + \frac{\partial \pi_5}{\partial (mg)} \Delta (mg) \quad (36)$$

$$\pi_6 = \frac{v^2}{g \ell}$$

$$\Delta \pi_6 = \frac{\partial \pi_6}{\partial v} \Delta v + \frac{\partial \pi_6}{\partial \ell} \Delta \ell + \frac{\partial \pi_6}{\partial g} \Delta g \quad (37)$$

The component equations are given as follows:

$$\pi_1 = 8.925 \times 10^{-5} \times \pi_2^{-3.58} \quad (38)$$

$$\pi_1 = 0.01 \times \pi_3^{-3.146} \quad (39)$$

$$\pi_1 = 0.174 \times \pi_4^{-1.106} \quad (40)$$

$$\pi_1 = 2.17 \times \pi_5^{-0.89} \quad (41)$$

$$\pi_1 = 0.193 \times \pi_6^{0.94} \quad (42)$$

During the experimental schedule the values of pi-terms were held constant at the following levels:

$$\begin{aligned}\bar{\pi}_2 &= 0.125 & \bar{\pi}_5 &= 16.34 \\ \bar{\pi}_3 &= 0.400 & \bar{\pi}_6 &= 0.96 \\ \bar{\pi}_4 &= 0.448\end{aligned}$$

The system variables were held constant at,

$$\begin{aligned}\ell &= 5 \text{ in} & mg &= 45.87 \text{ lb} \\ b &= 5/8 \text{ in} & v &= 43 (\theta = 21.8^\circ) \text{ in/sec} \\ h &= 1/4 \text{ in} & v &= (\ell_i/\ell_c) \sqrt{2 g \ell_c (1 - \cos \theta)}\end{aligned}$$

$$\Delta v = \frac{\partial v}{\partial \ell_i} \Delta \ell_i + \frac{\partial v}{\partial \ell_c} \Delta \ell_c + \frac{\partial v}{\partial \theta} \Delta \theta + \frac{\partial v}{\partial g} \Delta g \quad (43)$$

It was estimated that the error in measuring the system variables was,

$$\begin{aligned}\Delta \ell &= 0.2 \text{ in} & \Delta mg &= 2 \text{ lb} \\ \Delta b &= 0.01 \text{ in} & \Delta \theta &= 1 \text{ degree} \\ \Delta h &= 0.01 \text{ in} & \Delta \ell_c &= 0.5 \text{ in} \\ \Delta E &= 0 & \Delta D &= 0 \\ \Delta g &= 0\end{aligned}$$

The error in the dependent pi-terms was calculated as,

$$\begin{aligned}\Delta \pi_2 &= -0.0030 \\ \Delta \pi_3 &= 0.0096 \\ \Delta \pi_4 &= 0.0179 \\ \Delta \pi_5 &= 0.587 \\ \Delta \pi_6 &= 0.094\end{aligned}$$

The partial derivatives of Eq. (32) were computed to be,

$$\begin{aligned}\frac{\partial \phi}{\partial \pi_2} &= -4.366 & \frac{\partial \phi}{\partial \pi_5} &= -0.0098 \\ \frac{\partial \phi}{\partial \pi_3} &= -1.387 & \frac{\partial \phi}{\partial \pi_6} &= 0.182 \\ \frac{\partial \phi}{\partial \pi_4} &= -0.045\end{aligned}$$

Substituting these values in Eq. (32) the overall system error was computed.

$$\Delta\pi_1 = 0.0102$$

The maximum system error was found by taking the sum of the absolute values of each one of the terms on the right hand side of Eq. (32),

$$|\Delta\pi_1|_{\max} = 0.018$$

The maximum error was calculated to be 9 percent for permanent deflection. Therefore the results presented here are subject to a maximum error of 9 percent.

Theoretical Analysis of Strain Rate Effects

Ashburner (1972) developed an analytical method to predict deformation of ROPS subject to impact by a pendulum. He predicted the maximum and permanent deflections of the frame and the peak deceleration of an impacting pendulum. The model included elastic-plastic behavior of frames as well as elevation of yield stress due to the dynamic loading. The following set of equations constitute the model:

$$\text{Maximum acceleration} = \frac{M_{po}}{W\ell} \quad (44)$$

$$\text{Maximum deflection} = \frac{E_w \ell}{M_{po}} + \frac{M_{po}}{2k\ell} \quad (45)$$

$$\text{Final deflection} = \frac{E_w \ell}{M_{po}} - \frac{M_{po}}{2k\ell} \quad (46)$$

$$\frac{M_{po}}{M_{pso}} = \left[\sigma/D \right]^A \quad \text{if} \quad \frac{M_{po}}{M_{pso}} < 1.5 \quad (47)$$

$$M_{pso} = bh^2 \sigma_o/6 \quad (48)$$

$$K = Ebh^3/4\ell^3 \quad (49)$$

$$E_w = MV^2/2 \quad (50)$$

The above model was used to derive a generalized prediction equation in a dimensionless form with non-dimensional parameters being the same as expressed in Equation (23). Let,

$$d_s = \frac{E_w \ell}{M_{po}} \quad (51)$$

$$d_c = \frac{M_{po}}{k\ell} \quad (52)$$

Equations (45) and (46) can be rewritten as,

$$\text{Max deflection} = d_s + \frac{d_c}{2} \quad (53)$$

$$\text{Final deflection} = d_s - \frac{d_c}{2} \quad (54)$$

Time to reach yielding, t_o , in Eq. (47) was estimated from the following equation.

$$t_o = d_c / V \quad (55)$$

From the theory of elastic beams (Higdon et al., 1962) we know that ,

$$\begin{aligned} \sigma &= M_{pso} \frac{c}{I} \\ c &= \frac{d}{2} \\ I &= \frac{bd^3}{12} \text{ which implies that,} \\ M_{pso} &= \frac{bd^2}{6} \sigma_o \text{ and } M_{po} = \frac{bd^2}{6} \sigma_o \left(\frac{t_o}{D}\right)^A \end{aligned} \quad (56)$$

Substituting Eq. (56) in (52) the following was obtained.

$$\begin{aligned} d_c &= \frac{4bd^2 \sigma_o}{6 kh} \left[\frac{t_o}{D} \right]^A \\ \text{or} \\ t &= \frac{4bd^2 \sigma_o}{6 kh v} \left[\frac{t_o}{D} \right]^A \end{aligned} \quad (57)$$

Using the theory of elasticity as applied to the cantilever beams, its stiffness was expressed as follows,

$$k = \frac{p}{d} = \frac{E bd^3}{4\ell^3} \quad (58)$$

Substituting Eq. (58) into (57) and rearranging,

$$\frac{t_o}{D} = \left[\frac{8}{3} \left[\frac{b}{\ell} \right]^{-1} \left[\frac{b}{d} \right] \left[\frac{\sigma_o}{E} \right] \left[\frac{\ell}{D^2 g} \right]^{\frac{1}{2}} \left[\frac{v^2}{g\lambda} \right]^{-\frac{1}{2}} \right]^{\frac{1}{1-A}} \quad (59)$$

Equations (47), (48), and (50) were substituted into Equation (51) to obtain the following.

$$d_s = \frac{6}{8} \frac{MV^2}{bd^2} \sigma_o \left[\frac{t_o}{D} \right]^A$$

Rearranging,

$$\frac{d_s}{\ell} = 2 \left[\frac{3}{8} \right] \left[\frac{E}{\sigma_o} \right] \left[\frac{M_g}{E\ell^2} \right] \left[\frac{V^2}{g\ell} \right] \left[\frac{b}{\ell} \right]^{-3} \left[\frac{b}{h} \right]^2 \left[\frac{t_o}{D} \right]^{-A}$$

Substituting for $\left(\frac{t_o}{D}\right)$ from the Equation (59) and noting that $A' = \frac{A}{1-A}$,

$$\frac{d_s}{\ell} = 2 \left[\frac{3}{8} \right]^{1+A'} \pi_2^{-3+A'} \pi_3^{2-A'} \pi_5^{-(1+A')} \pi_6^{-\frac{B}{2}} \pi_8^{-1} \pi_9^{-\frac{B}{2}} \quad (60)$$

Equations (47), (48), and (49) were substituted into (52) and rearranged to obtain,

$$\frac{d_e}{\ell} = \frac{1}{2} \left[\frac{8}{3} \right] \left[\frac{\sigma_o}{E} \right] \left[\frac{h}{\ell} \right]^{-1} \left[\frac{t_o}{D} \right]^A$$

Substituting for $\left(\frac{t_o}{D}\right)$ the following was obtained.

$$\frac{d_e}{\ell} = \frac{1}{2} \left[\frac{8}{3} \right]^{1+A'} \left[\frac{b}{\ell} \right]^{-(A'+1)} \left[\frac{b}{h} \right]^{A'+1} \left[\frac{\ell}{D^2 g} \right]^{\frac{A'}{2}} \left[\frac{\sigma_o}{E} \right]^{A'+1} \left[\frac{V^2}{g\ell} \right]^{-\frac{A'}{2}} \quad (61)$$

Letting $A = -0.1$ (Ashburner, 1972) $K_1 = 2\left(\frac{3}{8}\right)$ and $K_2 = \frac{1}{2}\left(\frac{8}{3}\right)^B$ the following set of equations was obtained.

$$\frac{d_s}{\ell} = K_1 \left[\frac{b}{\ell} \right]^{-3.09} \left[\frac{h}{b} \right]^{-2.09} \left[\frac{\ell}{D^2 g} \right]^{.045} \left[\frac{E\ell^2}{Mg} \right]^{-1} \left[\frac{V^2}{g\ell} \right]^{.954} \quad (62)$$

$$\frac{d_c}{\ell} = K_2 \left[\frac{b}{\ell} \right]^{-.909} \left[\frac{h}{b} \right]^{-.909} \left[\frac{\ell}{D^2 g} \right]^{-.045} \left[\frac{E\ell^2}{Mg} \right]^0 \left[\frac{V^2}{g\ell} \right] \quad (63)$$

Equation (62) and (63) constitute generalized equations derived from the analytical model. In order to compare the experimental results with theoretically derived ones, a multiplicative relationship was assumed for the component equations to develop an experimentally determined generalized expression. Deflections, d_s and d_c were written in dimensionless form in the following manner:

$$\frac{d_s}{\ell} = \frac{1}{2} \left\{ \frac{d_{\max}}{\ell} + \frac{d_{\text{per}}}{\ell} \right\} \quad (64)$$

$$\frac{d_c}{\ell} = \frac{d_{\max}}{\ell} - \frac{d_{\text{per}}}{\ell} \quad (65)$$

(d_s/ℓ) and (d_c/ℓ) were calculated from the data of $(d_{\max}/\ell)$ and (d_{per}/ℓ) . Component equations were developed for (d_s/ℓ) and (d_c/ℓ) for each independent dimensionless variable by fitting a power curve, $y = ax^b$, using a least square analysis.

$$\frac{d_s}{\ell} = K'_1 \left[\frac{b}{\ell} \right]^{-3.24} \left[\frac{h}{b} \right]^{-3.0} \left[\frac{\ell}{D^2 g} \right]^{0.14} \left[\frac{E \ell^2}{Mg} \right]^{-0.87} \left[\frac{v^2}{g \ell} \right]^{0.96} \quad (66)$$

$$\frac{d_c}{\ell} = K'_2 \left[\frac{b}{\ell} \right]^{-1.53} \left[\frac{h}{b} \right]^{-2.48} \left[\frac{\ell}{D^2 g} \right]^{0.77} \left[\frac{E \ell^2}{Mg} \right]^{-0.05} \left[\frac{v^2}{g \ell} \right]^{1.15} \quad (67)$$

A comparison of Equations (66) and (67) with Equations (62) and (63) respectively, was made to establish agreement between the experimental data and the theoretically derived results. It can be seen that there is a good agreement in the exponents of the expression for $\left(\frac{d_s}{\ell}\right)$. However, the experimental results for $\left(\frac{d_c}{\ell}\right)$ do not compare well with that of the derived expression. Also, correlation coefficients for regression analysis of component equations for (d_c/ℓ) are much lower than that of (d_s/ℓ) . The deflection, d_s , is a combination of plastic and elastic deformations whereas d_c is the elastic component of the total deformation. Low correlation coefficients for d_c and poor comparison of the exponents in its expression could be attributed to the inaccuracy in measuring maximum deflection or an improper form for the component equation.

Conclusions

Based on the foregoing discussion on the effects of the rate of strain in the similitude modelling ROPS dynamic testing, the following conclusions were drawn.

- (1) In the similitude modelling of large elastic and plastic deformation of structures made of strain rate sensitive materials, it is not possible to build a true model of like material because of the distortion of the design conditions related to the strain rate property of the material.
- (2) Effects of rate of strain dependent pi-terms are small. A 400 percent increase in its value causes a 14.9 and a 19 percent change in normalized permanent and maximum deflections, respectively.
- (3) A prediction factor was developed to calculate the effects of the distortion of the strain rate sensitive design condition on the predictions of deflections. This factor was based on distorted model theory.

- (4) Analytical derivations for d_s (representing an average permanent and maximum deflection) were found to be in good agreement with the experimental results. However, a similar derivation for the elastic component of the total deflection, d_c , did not compare well with the experimental results.
- (5) The maximum estimated maximum experimental error was no more than 9 percent in the measurement of deflections.

This analysis presented various model design conditions for pendulum testing and discussed the implications of distortion of the strain rate design condition. ROPS models can be built and tested by pendulum tests by following the scaling laws for geometry, velocity, mass and by choosing the same material for model as the prototype. Due to the distortion of strain rate design condition the following prediction equation should be used to predict the behavior of the full scale frame.

$$\left(\frac{d}{L}\right) = \delta \left(\frac{d}{L}\right)_m \quad (68)$$

This similitude analysis will be summarized with respect to the other types of testing.

Overturn Test

The overturn test, also referred to as the field test, of the standard testing procedure, requires that the test ROPS be fitted on the tractor for which it was recommended. The tractor weight will be adjusted so that it will be at least 110 lbs per maximum horsepower or the maximum gross vehicle weight specified by the manufacturer. The front end weight shall be at least 25 percent of the total tractor weight. The test should be conducted on a firm soil bank as shown in Figures 19 and 20. The soil in the test area will have a cone index of at least 150.

Rear upset shall be induced by engine power with tractor operating in a gear to obtain 3 to 5 mph at maximum governed engine rpm, preferably by driving forward directly up a minimum slope of 60 ± 5 degrees, as shown in Figure 20. The engine clutch may be used to aid in the inducing of the upset.

To induce side upset, the tractor shall be driven under its own power along the specified path of travel at a minimum speed of 10 mph (or at maximum vehicle speed if under 10 mph) and over the ramp. Complete details of the field test may be found in ASAE Standard S306.3 or S383.

Similitude Analysis

The first step in the similitude analysis was to identify all variables pertinent to the modelling of ROPS behavior in the standard overturn test as described earlier. Only variables that affected the tractor motion and deformation of frame due to impact against ground surface were included in the analysis. A list of pertinent system variables is given in Table 5. In Table 5, the geometry variables apply to the tractor, the terrain, and the ROPS. Mass moment of inertia, I_{α} , applies to the tractor mass moment of inertia about its three axes and the product of inertia. The variable, σ_y is the dynamic yield stress and depends upon the static yield stress of the ROPS material and the rate of strain. The dynamic yield stress, σ_y , is related to the static yield stress by Equation (19).

Dynamic deformation of protective frames does not involve high rates of change of frame velocity of deformation which suggests that inertial resistance by the frame may be neglected. Material density, ρ_r , was, therefore, deleted from the analysis. During the analysis, worst possible conditions were considered for overturn accidents by assuming that no energy could be absorbed by the impacted surface. This was achieved by omitting the surface characteristics variable, s , from the analysis. It was also assumed that the ROPS deformation was mainly due to bending and that shear deflections were negligible.

Using the Buckingham's π -theorem, dimensionless pi-terms were formed and are presented in the following generalized equation (Murphy, 1970).

$$\left(\frac{d}{\ell}\right) = \phi \left(\frac{\lambda}{\ell}, \frac{I_{\alpha}}{M\ell^2}, \frac{\rho_t \ell^3}{Mg}, \frac{K\ell}{Mg}, \frac{cv}{Mg}, \mu, \frac{E\ell^2}{Mg}, \frac{E}{\sigma_0}, \frac{v^2}{g\ell}, \frac{vt}{\ell}, A, \frac{\ell}{D^2g} \right) \quad (69)$$

If,

$$\left(\frac{\lambda}{\ell}\right)_m = \left(\frac{\lambda}{\ell}\right)$$

$$\left(\frac{I_{\alpha}}{M\ell^2}\right)_m = \left(\frac{I_{\alpha}}{M\ell^2}\right)$$

and

$$\left(\frac{\ell}{D^2g}\right)_m = \left(\frac{\ell}{D^2g}\right)$$

then,

$$\frac{d}{\ell} = \left(\frac{d}{\ell}\right)_m$$

The subscript m denotes a variable in the model system. As indicated earlier, in order to satisfy the constitutive relationship for the ROPS material, the same model material was chosen as that of the prototype frame. This implied that: $E_m = E$, $\sigma_{om} = \sigma_o$, $D_m = D$, and $A_m = A$. The model tests were performed in the same gravitational field, i.e., $g_m = g$. The design conditions and the previously discussed constraints were used to derive the scaling relationships for the model system design. Table 6 lists the various scale factor obtained from the model design conditions. The scale factors are expressed as functions of the geometric scale factor, $n = \ell/\ell_m$.

From Table 6 it could be seen that, in order to satisfy the design conditions, the density of model tractor had to be greater than that of the prototype by a factor equal to the length scale factor. For example, if the length scale was chosen to be equal to 4, the density of the model tractor and its component would have to be four times that of the prototype. Initial calculations revealed that it was not feasible to design tractor components to meet these requirements. Therefore, the model tractor density was set equal to the prototype tractor density which resulted in the following.

$$M_m = n^{-3} M \tag{70}$$

Equation (70) implied distortion of the force scale. Also, the following design condition arises due to the strain rate effects in dynamic loading.

$$\left(\frac{\ell}{D^2g}\right)_m \neq \left(\frac{\ell}{D^2g}\right) \quad (71)$$

The effects of distorting the above design condition have been discussed by Srivastava and Rehkugler (1976). Distortion in the force scale will be discussed in the next section.

Compensated Distortion Analysis:

Principles of compensated distortion were applied to the problem of distortion in the force scale. In compensated distortion approach to model design, according to Murphy (1970), distortion in one design condition is used to compensate the distortion effects of another such that the model would behave as a true model. If more than one design condition is distorted, then this distortion is controlled, if plausible, in such a way as to produce compensated effects. However, if only one of the design conditions is distorted, then another appropriate design condition is deliberately distorted to compensate the distortion effects. Knowledge of the phenomena affecting the physical system behavior is essential for compensated distortion analysis.

The analysis which now follows presents compensated distortion method applied to the problem of distortion of force scale in the deformation of ROPS in overturn testing. As discussed earlier, it was assumed that the frame deformation was mainly due to bending. Shear, buckling, and localized deformations were neglected in the following analysis. Consider the following generalized equation of an elastic curve due to bending (Higdon, et al., 1962).

$$\frac{1}{\rho} = \frac{B}{EI} \quad (72)$$

Rewriting Equation (72) in a non-dimensional form we obtained.

$$\left(\frac{\ell}{\rho}\right) = \left(\frac{B}{E\ell^3}\right) \frac{\ell^4}{I} \quad (73)$$

From this the following design conditions were established.

$$\left(\frac{B}{E\ell^3}\right)_m = \left(\frac{B}{E\ell^3}\right) \quad (74)$$

$$\left(\frac{\ell^4}{I}\right)_m = \left(\frac{\ell^4}{I}\right) \quad (75)$$

If a model system was designed to satisfy the above conditions, then the following prediction equation would hold.

$$\left(\frac{\ell}{\rho}\right) = \left(\frac{\ell}{\rho}\right)_m$$

For the same material, $E_m = E$, and for force scaling being, n^3 , where $n = \ell/\ell_m$, it can be seen that Equation (74) cannot be satisfied. Introducing a distortion factor, α , in Equation (74), the following was obtained.

$$\left(\frac{B}{E\ell^3}\right)_m = \alpha \left(\frac{B}{E\ell^3}\right) \quad (76)$$

Design condition Equation (75) could be satisfied by maintaining geometric similarity. To compensate distortion α of Equation (76), an additional distortion factor, β , was introduced in Equation (75) as follows.

$$\left(\frac{\ell^4}{I}\right)_m = \beta \left(\frac{\ell^4}{I}\right) \quad (77)$$

Due to the distortion in the model, the prediction equation was written as,

$$\left(\frac{\ell}{\rho}\right) = \delta' \left(\frac{\ell}{\rho}\right)_m \quad (78)$$

or,

$$\delta' = \frac{(\ell/\rho)}{(\ell/\rho)_m} \quad (79)$$

Substituting Equation (73) for both model and the prototype system into Equation (79), we obtain,

$$\delta' = \frac{(B/E\ell^3)(\ell^4/I)}{(B/E\ell^3)_m(\ell^4/I)_m} \quad (80)$$

Substitution of Equation (76) and (77) into Equation (80) yielded,

$$\delta' = \frac{1}{\alpha\beta} \quad (81)$$

Equating δ' to 1 in Equation (81) for a "true" model, we get,

$$\beta = \alpha^{-1} \quad (82)$$

From Equation (76),

$$\alpha = n^{-1} \quad (83)$$



Substituting Equation (83) in (82), we get

$$\beta = n \quad (84)$$

Equation (77) can be satisfied if,

$$I_m = n^{-5} I \quad (85)$$

According to the foregoing analysis, it is seen that distortion in bending moment design condition can be compensated by systematically distorting design conditions related to the area moment of inertia. This results in a different scale factor ($n^{-5/4}$) for the cross-sectional measurements of the ROPS. This analysis is valid only for the ROPS behavior in the elastic range and is limited by the assumptions made to obtain the elastic curve equation. To extend this analyses to include the yield behavior, yield stress of ROPS material, σ_o , had to be included in the analysis. This indicates that the dependent pi-term (ℓ/ρ) would be a function of an additional pi-term ($B/\sigma_o \ell^3$), which results in the following additional design condition.

$$(B/\sigma_o \ell^3)_m = (B/\sigma_o \ell^3) \quad (86)$$

The equation above is not satisfied for $\sigma_{om} = \sigma_o$ so there is additional distortion in the model system related to the yield stress design condition. A prediction factor was developed to account for the distortion in Equation (86). From beam theory the following is known (Higdon, et al., 1962):

$$B_y = \frac{\sigma_o I}{c}$$

whereas B_y is the yield bending moment and c is the maximum perpendicular distance from the neutral axis. The ratio of the model and prototype yield moments can be written as,

$$(B_y/B_{ym}) = (\sigma_o/\sigma_{om}) (I/I_m) (c_m/c) \quad (87)$$

or

$$(B_y/B_{ym}) = n^{\frac{15}{4}}$$

To satisfy Equation (86), the following distortion factor is introduced.

$$(B_y/\sigma_o \ell^3)_m = \gamma (B_y/\sigma_o \ell^3) \quad (88)$$

or

$$\gamma = n^{-\frac{3}{4}}$$

Substituting the expression for B_y to replace B in Equation (73), we obtained,

$$(\ell/\rho)_{\text{yield}} = (B_y/E\ell^3)(\ell^4/I)$$

or

$$(\ell/\rho)_{\text{yield}} = (\sigma_o/E)(\ell/c) \quad (89)$$

By substituting Equation (89) into (79) for both the model and prototype systems the following expression for distortion factor for dimensionless (yield) curvature was obtained.

$$\delta' = n^{-\frac{1}{4}} \quad (90)$$

The above expression can be used to correct predictions made about the curvature at yield by using the generalized prediction equation. However, this information must be related to the deflections. It is known from calculus that for small slopes, which generally is the case for elastic deformation, that,

$$\frac{1}{\rho} = \frac{d^2 y}{dx^2} \quad (91)$$

or

$$y = \int \int \left(\frac{1}{\rho}\right) dx dx$$

or

$$\left(\frac{y}{\ell}\right) = \int \int \left(\frac{\ell}{\rho}\right) d\left(\frac{x}{\ell}\right) d\left(\frac{x}{\ell}\right) \quad (92)$$

The prediction factor, δ , for yield deflection was defined as,

$$\delta = \frac{(y/\ell)}{(y/\ell)_m} \quad (93)$$

Because Eq. (92) holds for the model and the prototype systems, substitution in Eq. (93) yielded,

$$\delta = \frac{\int \int \left(\frac{\ell}{\rho}\right) d\left(\frac{x}{\ell}\right) d\left(\frac{x}{\ell}\right)}{\int \int \left(\frac{\ell}{\rho}\right)_m d\left(\frac{x}{\ell}\right)_m d\left(\frac{x}{\ell}\right)_m} \quad (94)$$

or

$$\delta = \frac{\int \int \left(\frac{\ell}{\rho}\right) d\left(\frac{x}{\ell}\right) d\left(\frac{x}{\ell}\right)}{\int \int n^{\frac{1}{4}} \left(\frac{\ell}{\rho}\right) d\left(\frac{x}{\ell}\right) d\left(\frac{x}{\ell}\right)} ; \text{ for } d\left(\frac{x}{\ell}\right)_m = d\left(\frac{x}{\ell}\right) \quad (95)$$

or

$$\delta = n \frac{1}{4}$$

Eq. (95) presents a prediction factor, used in the prediction equation for deflection as follows,

$$\left(\frac{Y}{L}\right) = n \frac{1}{4} \left(\frac{Y}{L}\right)_m \quad (96)$$

Because the standard static or dynamic testing of ROPS requires the results in the form of force-deformation plots, it was appropriate to present the prediction equations in terms of force rather than moments. The prediction equations are summarized in the following form:

For elastic deformations,

$$P = n^3 P_m \quad (97a)$$

$$d = n d_m \quad (97b)$$

and for yield deflections,

$$P = n \frac{1}{4} \left(\frac{\sigma_o}{\sigma_{om}}\right) P_m \quad (98a)$$

$$d = n \frac{3}{4} \left(\frac{\sigma_o}{\sigma_{om}}\right) \left(\frac{E}{E_m}\right) d_m \quad (98b)$$

Hypothesis: In order to extend the analysis to elastic-plastic region it was hypothesized that by assuming negligible strain hardening effects, the set of equations describing the behavior of the frame at the yield point, could be used to describe its behavior in the plastic range also.

Equations (97a), (97b), (98a) and (98b) are prediction equations for prototype load and deformation for ROPS elastic-plastic behavior. These equations contain material properties for the model and the prototype materials. This would account for any variation in the material properties for model material which often is the case (Easter, 1975). If a load deformation diagram is known for model frame then using these equations, one can construct a similar plot for the prototype frame.

Dynamic Loading:

The prediction equations in the previous section were developed for static loading. In overturn and pendulum tests the forces are dynamic in nature and the previous equations had to be modified to take into account such effects. Ashburner (1972) has discussed, in the following equation, the elevation of yield stress due to dynamic loading.

$$\sigma_y = \sigma_o \left[\frac{t_o}{D} \right]^A$$

For dynamic loading the static yield stress is replaced by dynamic loading yield stress in Eq. (98a) and (98b). Substituting the above equation in Eq. (98a) and (98b) in place of σ_y , the following was obtained.

$$P = n^{\frac{11}{4}} (\sigma_o/\sigma_{om}) (t_o^A/t_{om}^{A_m}) (D_m^{A_m}/D^A) P_m$$

$$d = n^{\frac{3}{4}} (\sigma_o/\sigma_{om}) (E_m/E) (t_o^A/t_{om}^{A_m}) (D_m^{A_m}/D^A) d_m$$

Letting $D_m = D$ and $A_m = A$ for same model and prototype material and from the time scale noting that $t_m = t\sqrt{n}$, the following set of prediction equations were obtained.

$$P = n^{\left(\frac{11}{4} + \frac{A}{2}\right)} (\sigma_o/\sigma_{om}) P_m \quad (99a)$$

$$d = n^{\left(\frac{3}{4} + \frac{A}{2}\right)} (\sigma_o/\sigma_{om}) (E_m/E) d_m \quad (99b)$$

Equation (99a) and (99b) are the prediction equations for the elasto-plastic load-deformation behavior of ROPS in dynamic loading. For mild steel, $A = 0.1$ (Ashburner, 1972). It may be noted that if the strain rate effects are ignored in the similitude modelling, predictions of load deformation would be in error equal to $n^{-(1/20)}$. Assuming a geometric scale factor, $n = 4$, this error would be about 6.7 percent of the prototype behavior.

Overturn Testing:

In the foregoing analysis, applied force and resulting deformation had to be known explicitly in order to predict the prototype behavior. In the overturn test

forces are induced due to the impact of the frame with the terrain surface. It was desired to develop a prediction equation to predict deflections of the protective frame in overturn tests without having to know the forces explicitly.

Let's consider the following equation -

$$\frac{dN}{dy} = P \quad (100)$$

Equation (100) applies to model as well as prototype systems. From this the following equations were written.

$$\frac{y}{y_m} = \frac{\int \frac{1}{P} dN}{\int \frac{1}{P_m} dN_m}$$

Substituting the expression for the force P in the above equations, from (99a) and noting that $N_m = n^{-4} N$ the following was obtained.

$$\frac{y}{y_m} = \frac{n^{-(\frac{11}{4} + \frac{A}{2})} \left(\frac{\sigma_{om}}{\sigma_o} \right) \int \frac{1}{P_m} d(n^4 N_m)}{\int \frac{1}{P_m} dN_m}$$

$$\frac{y}{y_m} = n^{(\frac{5}{4} - \frac{A}{2})} \frac{\sigma_{om}}{\sigma_o}$$

Replacing y with elasto-plastic deflection, d, the following equation for predicting the frame deflection in an overturn test is,

$$d = n^{(\frac{5}{4} - \frac{A}{2})} \left(\frac{\sigma_{om}}{\sigma_o} \right) d_m \quad (101)$$

Dissimilar Cross-Sections:

Previous sections dealt with modelling of elastic-plastic deformations of ROPS in the standard static and overturn tests using frame models of similar cross-sectional shapes designed to satisfy the model design conditions. Generally, commercial ROPS are made from thin walled hollow sections of steel. Modelling of these ROPS would require non-standard sections of extremely thin walls which pose fabrication problems. To overcome this problem it was decided to develop scaling laws to design model

frames of solid cross-sections to represent rollover section members.

An additional variable, f (shape factor), defined by Brown and McNabb (1974) was introduced.

$$f = \frac{B_p}{B_y} \quad (102)$$

Shape factors for the most commonly used steel for ROPS have been calculated by the following equations (Brown and McNabb, 1974). For hollow circular cross-sections,

$$F = \frac{16}{3\pi} \frac{\beta^3 + \beta^2 + \beta}{\beta^3 + \beta^2 + \beta + 1} \quad (103)$$

where $\beta = D_o/D_i$, (the ratio of the outside diameter to the inside diameter) and for hollow rectangular cross-sections (Fig. 21),

$$f = \frac{3}{2} \frac{[1 - (\frac{b}{a})(\frac{d}{c})^2]}{[1 - (\frac{b}{a})(\frac{d}{c})^3]} \quad (104)$$

It should be noted that the shape factor for solid section would be the limiting case of the hollow one.

Introducing shape factor, f , as an additional variable imposes the following additional model design condition.

$$f_m = f \quad (105)$$

This design condition is inherently satisfied if the geometric shape similarity is maintained between the model and the prototype. It is clear that when a model of dissimilar cross-sectional shape is considered, the above condition cannot be satisfied. In the previous analysis yield bending moments followed a scaling relationship for bending moments. In extending the analysis to plastic region it was implied that the elastic-plastic moments would follow the same scaling law. If this were true then the following equation should hold:

$$\frac{B_p}{B_{pm}} = n^{\frac{15}{4}} \quad (106)$$

Substituting Eq. (102) for both the prototype and model system into Eq. (106) the following was obtained -

$$\frac{B_y}{B_{y_m}} = \left(\frac{f}{f_m}\right) n^{-\frac{1}{4}} \quad (107)$$

Using the above equation, the prediction equation for load-deformation behavior was derived as follows.

$$P = n^{\left(\frac{11}{4} + \frac{A}{2}\right)} \left(\frac{f}{f_m}\right) \left(\frac{\sigma}{\sigma_m}\right) P_m \quad (108a)$$

$$d = n^{\left(\frac{3}{4} + \frac{A}{2}\right)} \left(\frac{f}{f_m}\right) \left(\frac{\sigma_o}{\sigma_{om}}\right) \left(\frac{E_m}{E}\right) d_m \quad (108b)$$

Equation (101) was modified for this case as follows.

$$d = n^{\left(\frac{5}{4} + \frac{A}{2}\right)} \left(\frac{f_m}{f}\right) \left(\frac{\sigma_{om}}{\sigma_o}\right) \left(\frac{E_m}{E}\right) d_m \quad (109)$$

Equation (87) is satisfied if $Z_m = n^{-15/4} Z$ for the dissimilar model cross-sections. The model should be constructed with sections of similar outer shapes, i.e., square, rectangular or circular sections.

In summary Eq. (97a) and (97b) can be used to predict the prototype behavior in static loading in elastic range. To predict yield point and elasto-plastic behavior Eq. (98a) and (98b) should be used. Load-deformation behavior in dynamic loading is predicted by Eq. (99a) and (99b) for elasto-plastic deformations. Eq. (101) is to be used for predicting elasto-plastic deformation in overturn test and Eq. (109) will predict similar behavior for dissimilar section frames. Eq. (108a) and (108b) will predict load-deformation behavior for dissimilar model under static loading conditions.

Experimental Verification

The objective of the experimental verification was to verify the scaling relationships for model design and the prediction equations as developed in the previous section. In order to complete the verification it was desired to select a proto-

type tractor and ROPS system and design its model using the scaling relationships. The results obtained by testing the model could be used to predict the prototype behavior. The predicted behavior can then be compared with the actual behavior by testing the prototype system. Testing with a full scale tractor and ROPS in either the static or dynamic loading was beyond the scope of the project and therefore an alternative technique was sought. It was decided to build two tractor-ROPS model systems of different scale factors. The smaller system was regarded as the model for the larger model system. Thus, predictions made from the test results of smaller model systems could be verified by testing the larger model system.

Model Tractor Design

In order to design two tractor models it was necessary to identify a prototype tractor. Initially it was considered desirable to define a generalized tractor and then design models of such a tractor. A study was completed to correlate the tractor horsepower to its geometry and tire specifications and center of gravity. The purpose of trying to design a generalized tractor model was to be able to apply the results of this model to predict the behavior of any size "full scale" tractor by calculating a scale factor. Table 7 presents the linear equations developed to describe the relationships of the various geometrical variables and tractor horsepower. It can be seen from this table that there apparently is no correlation between the tire sizes for both front and rear wheels. It is also evident from this table that except for tractor weight and wheel base the geometric parameters do not show any strong correlation with tractor horsepower. Tire sizes have a great effect on the mass moment of inertia of a tractor and mass moment of inertia has a significant influence on tractor overturning stability and the severity of an overturn. Poor correlation of tire size indicates that mass moment of inertia also would not correlate well with horsepower. Published values of mass moments of inertia were not available on tractor designs of interest to us. Because mass moment of inertia data was not available and because of an expected large variation

in this value for various tractor designs in relation to their weight, it was decided to design models of a specific commercial tractor.

Several tractor manufacturers in the U.S. were contacted for specifications of their commercially manufactured tractors. Deere and Co.* responded with complete information on their tractor model JD4430. Model tractor design equations were used to compute the design specifications of the two model tractor of 1/4 and 1/8-scale sizes as presented in Table 8. In order to actually construct the models it was necessary to obtain various geometric measurements. A full size tractor was not readily available because their particular tractor was a new model among John Deere tractors. A 1/16-size toy replica was bought from the local John Deere dealer. This purpose of the toy model was to measure various geometric dimensions to design the two model tractors. Various views of the 1/16-scale model were photographed with a grid in the background. Figs. 22 and 23 show two of the various views that were photographed. These photographs were used to trace an outline on paper from which the various desired dimensions were obtained. These dimensions and the geometric scale factor were used to design and construct the two model tractors.

In designing model tractors, masses of the rear wheels, front wheels and complete front-end were also considered. Masses for these components were determined from the shipping weights in the parts catalog. Mass moments of inertia of the model tractors were determined by the method described by Davis (1974) which utilizes a trifilar pendulum (Fig. 24). Center of gravity of each model tractor was determined by suspending the model tractor in its longitudinal plane. Each tractor was suspended twice at different points. The intersection of vertical lines drawn from both points of suspension located the center of mass. Masses were added to the model tractors to satisfy mass scaling requirements. An iterative process was

*The use of trade names in this report does not imply endorsement by Cornell University of the products named, nor criticism of similar products not mentioned.

followed to determine the location of the masses so that the scaling requirements for the location of center of gravity and the mass moments were satisfied.

Pneumatic rubber tires for the rear wheels were used in both tractor models. Front wheels of the 1/8-scale model were made of solid rubber. The wheel weights were adjusted to conform the proper mass scaling for the wheels. The front-end was made to pivot and adjustments were provided to limit the rotation of front-end if desired. The model tractors had adjustable "toe-in". The 1/4-scale model tractor is shown in Fig. 25. Two models of the standard overturn ramps as specified by ASAE Standard S306.3 (ASAE Yearbook, 1976) were built. The ramps were made of plywood and were of 1/4-scale and 1/8-scale sizes. Two scale model starting ramps were built and attached to the overturning ramps. The tractor speed as it traveled over the overturning ramp was determined by the release elevation of the model tractors on their respective starting ramps. Fig. 26 shows the 1/8-scale model tractor, and the starting and overturn ramps. The tractors were unpowered and were allowed to roll freely when released from the sloping starting ramp.

Motion Analysis

The first step in the experimental verification was to verify if motion of the 1/4-scale model tractor could be predicted using the motion of the 1/8-scale model. Primary interest was only in the overturning motion of the tractor. The tractor overturning motion was recorded on two 16-mm movie cameras at 64 frame/sec film speed. The cameras were located at right angle with respect to each other and filmed two orthogonal views of the tractors. Grid boards were placed such that they were in the background of the longitudinal and the transverse planes as the two cameras captured the tractor motion. This arrangement was repeated for each of the model tractors fitted with a two-post type of rollbar. The motion of the upper left hand corner of the rollbar was monitored. Figures 27 and 28 show a schematic diagram of the overturn and the starting ramps and the locations of two cameras.

The motion of the appropriate point on the model tractors was computed by frame by

frame analysis. The following equations were derived to compute the parallax errors in viewing Figs. 27 and 28.

$$y = \frac{(LF \cdot LD \cdot XR2 - LF \cdot LD \cdot DF - XR1 \cdot XR2 \cdot LF)}{(LF \cdot LS + XR1 \cdot LC - XR1 \cdot XR2 - LC \cdot LD + LD \cdot XR2)} \quad (110)$$

$$x = \frac{(-LD \cdot LF - LS \cdot y)}{(XR1 - LD)} \quad (111)$$

$$z = \frac{y(\frac{h_f}{l_f} - \frac{X_{R3}}{l_f}) + \frac{X_{R3} l_f}{l_f}}{l_f} \quad (112)$$

The following set of equations were used to predict the motion of the 1/4-scale tractor from its 1/8-scale model.

$$\begin{aligned} x &= n X_m \\ y &= n y_m \end{aligned} \quad (113)$$

$$z = n z_m$$

and

$$t = \sqrt{n} t_m \quad (114)$$

The film was projected on a specially designed translucent screen. The screen was fitted with a pointer which could be moved anywhere on the opposite side of the projection screen. Movement of this pointer resulted in voltage changes in mutually orthogonal potentiometers. Thus, the position of the pointer could be located by knowing the voltage levels of the potentiometers. The high speed film was projected one frame at a time. Each time a frame was projected on the screen the pointer was moved over the upper left hand corner of the ROPS and the corresponding voltages were recorded. This process was repeated for both films containing the front and the side views for both tractor models. The voltages were calibrated for movement on the screen with respect to the grids. A computer program was developed which calculated the true motion in three dimensional space using the voltage levels and the Eq. (110), (111), and (112). Figs. 29, 30 and 31 show both the predicted and measured displacement vs. time data obtained from film analysis for x, y and z displacements, respectively. It can be seen from these figures that 1/4-scale model

tractor motion was accurately predicted by the motion of 1/8-scale tractor model. Fig. 32 shows a sequence of photographs of 1/4-scale and 1/8-scale tractor motion prepared from the high speed movie film. The photographs are selected from frames at equal time intervals. The intervals for the 1/8-scale model corresponds to the scaling requirements for time. Therefore, each set of photographs at different intervals corresponds to the similar events in the prototype and the model systems. Similarity in the motion of these tractors is quite clear from these photographs which implies that the loading conditions created by the overturning would also be similar.

Static Testing

The purpose of the static testing of frame models was to verify Eq. (97a) (97b), (98a), (98b), (108a) and (108b). It was decided to build some simple types of ROPS models for the verification purposes. Fig. 33 shows a drawing of the two-post model frame used in static testing. The frames were built for both scale sizes according to the scaling relationships obtained from the compensated distortion analysis. The frames were made of cold rolled mild steel. The larger frame (referred to as the prototype) was 18 in. tall and 10 in. wide. The frame was made of 5/8 x 3/16 in. stock material. The smaller frame (referred to as the model) was regarded as the model the prototype frame with a geometric factor of 2.08. This scale factor was chosen so that standard stock sizes could be used to build the frames.

An Instron testing machine was used to apply quasi-static loading on the frames. A knife edge supported on roller was used to apply the load in the transverse direction at the upper frame corner as shown in Fig. 34. Load-deformation plots were obtained for both frames. Results of the model frame were used to predict the prototype behavior using the prediction equations (97a), (97b), (98a), and (98b).

The predicted and measured load deflection results are presented in Fig. 35 which shows that the predicted deflection at about 12% less than the measured results at

maximum deflection. It was assumed that the material properties for both frames were numerically equal. The predicted load deflection plot can be used to construct energy vs. deflection diagram required for standard static tests. Another set of frames with welded joints was fabricated from solid square sections of cold rolled steel. The geometric scale factor between the two frames as $n = 2$. Fig. 36 shows the predicted and prototype load-deflection data for the welded frames. The predicted deformation was less than the measured deformation by about 35%. The predicted results do not compare as well with the measured ones for the welded frames as the bolted type of frame. The reason for this was the effects of welding not being 'similar' in model as in the prototype frame. The smaller frame was made of 1/8 in. x 1 1/8 in. steel stock and was difficult to weld. This may not be a problem when building only a 1/4-scale frame as a model of full-scale frame. However, this is a problem area which requires expert welding.

The design equations were applied to construct four-post type ROPS. One quarter-scale and a 1/8-scale size frames were built and tested under static loading conditions. Fig. 37 shows the two frames after static-testing. Good similarity exists between the model and the prototype frames. Fig. 38 shows the static test results for the four-post frames. It can be seen that the modelling theory can be applied successfully to the four-post type of frames.

Testing with Dissimilar Models

The next step in the overall experimental verification was to verify Eq. (108a) and (108b). This was done by building a prototype frame of hollow cross-section steel and a model frame of solid sections. In designing the model frame, previously developed scaling relationships were applied. To determine the cross-sectional dimensions of the solid section the following equation was used:

$$Z_m = n^{-\frac{15}{4}} Z$$

The above equation ensures that the modelling requirements of yield bending moments as specified by Eq. (106) would be satisfied. General shape similarity

between the hollow prototype and solid model was maintained, i.e., the square hollow section was modelled by a square solid section. Both frames were tested in static loading. Load-deformation curves obtained from the prototype frame testing and by predicting from the model data are presented in Fig. 39. Fig. 40 shows similarity in the deformed shapes of the prototype and the model frames. Modelling of hollow cross-sections with solid cross-section models simplifies fabrication of models of geometrically complex prototype ROPS.

It was decided to further verify the modelling equations by comparing predicted results against actual full-size frame test data. Since testing with a full-scale frame was beyond the scope of the project, various ROPS manufacturers were contacted for test results on their commercially built ROPS. International Harvester Company supplied complete detailed drawings of a two-post and a four-post commercially manufactured approved ROPS. Static test results in the form of load-deflection diagrams were also provided by the manufacturer. The two-post type frame was selected for model design. The prototype frame was primarily made of thin walled hollow rectangular vertical and top-cross members. The frame was mounted on a hexagonal rear axle housing by means of four bolts on each side. A model frame of one-quarter scale in size was designed and fabricated. The model frame was made of solid members. The model system included the model frame, the frame mounting parts and the model axle housing. The frame was tested by applying quasi-static loading from the side at a rate of 0.25 cm/sec by the Instron testing machine. The force was applied until enough plastic hinges were developed so that the frame behaved as a plastic 'mechanism'. When the force in the load-deflection diagram did not increase during testing it indicated that the frame had become a plastic mechanism. At this point the testing was stopped and measurements were made to locate various plastic hinges. Points of permanent plastic deformation are referred to as plastic hinges. Values of the shape factors of the corresponding location for the prototype frame were calculated from the detailed drawings provided by the ROPS manufacturers. The

smallest value of the shape factors for the prototype was selected for the prediction equations (108a) and (108b). This was selected because it would result in largest discrepancy between the predicted and the actual data because the shape factor for the solid model was 1.5. Figure 41 shows the two-post full-scale ROPS after static testing. The 1/4-scale model of this frame is shown in Fig. 42 before testing. Figure 43 shows the actual load/deformation plot obtained during static testing of the full scale ROPS and the curve as it was predicted using a quarter-scale model. It can be seen from this figure that the load-deflection behavior of the full size frame was accurately predicted by the model test results. The model frame is shown in Fig. 44 after static test.

Overturn Testing

The purpose of the overturn testing was to verify Eq. (101), prediction equation for frame deflection in an overturn test. Although the prediction equation applies to both sideways and rearward overturn test, only side overturn tests were performed with the model tractors. The models were overturned on concrete floor surface. This was done to simulate the worst possible case of overturn test. The forward speed was 223.5 cm/sec for the 1/4-scale model and 158 cm/sec for the 1/8-scale model tractor. These values correspond to the ASAE Standard S306.3 and follow the velocity scaling requirements. The tire pressure in rear tires was adjusted in both model tractors so that the radial tire spring constant conformed to the appropriate scaling relationships. Rear tire stiffness for the 1/8-scale tractor was 56 N/cm and the same parameter for 1/4-scale model was 224.15 N/cm. According to the model design relationships a geometric scale factor of 2 corresponds to a scale factor for the tire stiffness equal to 4. Since the rear tires for both tractor models were rubber tires and the ramp was made of plywood, it was assumed that the coefficient of tire friction was equal in both cases.

In order to verify the prediction equation it was necessary to measure the resulting frame deflection. Maximum instantaneous deflection required instrumen-

tation to measure it during the rollover. The final or permanent deflection could be measured after completing the rollover. It was decided to develop instrumentation which could be used to measure the maximum and permanent deflection during a rollover. It was desired to measure the displacement of the point on the model frames which impacted against the ground surface during the test. During some preliminary overturn tests with model frames it was noted that the upper left hand corner always impacted the surface and displaced in such a way that it required measurement in all the three dimensions. Thus, the requirements on a displacement transducer were that it should permit dynamic measurements during rollover in three dimensional space but should not significantly affect the strength of the test frames.

Refer to Fig. 45 which shows a three dimensional coordinate system and an arbitrary point, p. Let p_o , p_x and p_t be fixed points with respect to the coordinate system and located as shown in the diagram. Let p be the point whose three dimensional displacement is to be measured. Let l_o , l_x , and l_y be the line segments directed towards point p from the points p_o , p_x , and p_y respectively. Let the distance between the points p_o and p_x be s_x , and that between p_o and p_y be s_y . The following set of equations define the position of point p as a function of the lengths of the line segments:

$$x = (s_x^2 + l_o^2 - l_x^2)/2 s_x \quad (115)$$

$$y = (s_y^2 + l_o^2 - l_y^2)/2 s_y \quad (116)$$

$$z = (l_o^2 - x^2 - y^2)^{\frac{1}{2}} \quad (117)$$

In the above equations the variables are l_o , l_x and l_y . If these variables are measured during the deflection of the point p then its displacement components could be computed using the above set of equations. To measure the lengths, commercially available negator motors and potentiometers were used as shown in Fig. 46. A negator motor is a constant torque spring cable retractor pulley. It consists of a small diameter cable, and take-up reel, a constant force negator

spring band and a torque reel. One end of the cable is attached to the take-up reel and is wound around the take-up reel. The negator spring offers a constant force against pulling of the cable and retracts it with a constant force. The cable was wound on a small pulley mounted on a potentiometer shaft and was connected to the point whose motion is to be recorded. The cable was under constant tension equal to the constant force rating of the torque motor. Any movement of the point to which the cable end was connected caused the cable to be either pulled away or retracted by the torque reel. This resulted in rotation of the potentiometer. The cable was fastened at a point on the pulley surface by a set screw to avoid any slippage of the cable over the potentiometer pulley. A ten turn, 10K ohm wire wound trimmer potentiometer was used. A 20 VDC voltage was applied across the resistor and the output voltage proportional to the linear movement of the cable was recorded. The diameter of the potentiometer pulley was 1.9 cm. The torque motor was rated at 4.448 N (1 lb.) force and the total available length of the cable was 0.9 meters (36 in.). The potentiometer pulley diameter and the potentiometer specifications depend upon the nature of the measurement. Various torque motors with various torque values and cable lengths are commercially available to meet specific requirements. Three such units were used in the final device to provide each of the line segments in Fig. 45. The final device is shown in Fig. 47. The cables were connected to the point on model frame where displacement was to be measured. A four channel FM instrumentation tape recorder was used to record potentiometer output during a test. A strip chart recorder was then used to produce a hard paper copy of the displacements. Initial length l_o , l_x and l_y were measured before test and the changes in the lengths were obtained from the recorded signal. Total lengths at any time during a test were computed by adding the length change to the initial lengths. Using the above equations, new positions of point p were then calculated. The deflection of point p was computed from the initial

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position of point p, the initial lengths and the change in the three dimensional three lengths.

Two such devices were constructed. The device of the 1/8-scale model was much smaller and utilized a smaller negator motor rated at 1/3 lb. pull. Figure 48 shows a typical recording of the potentiometer output of the transducer designed for the 1/4-scale model system. Following is a sample calculation to illustrate how deflection was computed. For this transducer $s_x = 18.4$ cm and $s_y = 7.6$ cm. The initial l_o , l_x and l_y were measured to be 35.5, 31.3, and 37.4 cm respectively. Using Eq. (115), (116), and (117) the initial location of the upper left hand corner of the model ROPS was calculated.

$$x_o = [(18.4)^2 + (35.5)^2 - (31.3)^2] / (2 \cdot 18.4) = 16.8 \text{ cm}$$

$$y_o = [(7.6)^2 + (35.5)^2 - (37.4)^2] / (2 \cdot 7.6) = -5.3 \text{ cm}$$

and

$$\{(35.5)^2 - (16.8)^2 - (-5.3)^2\}^{\frac{1}{2}} = 31.3 \text{ cm}$$

From Fig. 47 the changes in the line segments were computed by noting that the calibration was 71.65 mv/cm.

$$\Delta l_o = 5.024 \text{ cm}$$

$$\Delta l_x = 2.024 \text{ cm}$$

$$\Delta l_y = 3.907 \text{ cm}$$

Using the initial lengths and the change in each length, final lengths were computed.

$$l_o = 40.524 \text{ cm}$$

$$l_x = 29.276 \text{ cm}$$

$$l_y = 41.307 \text{ cm}$$

Again using now the new lengths and Equations (115), (116), and (117) the new position of the deflected point was computed. Note that the values of s_x and s_y remain unchanged.

$$x' = 30.5 \text{ cm}$$

$$y' = -0.45 \text{ cm}$$

$$z' = 26.6 \text{ cm}$$

The difference in the new and the initial position components resulted in the following deflection in three dimensional space.

$$\Delta x = 13.7 \text{ cm}$$

$$\Delta y = 14.9 \text{ cm}$$

$$\Delta z = -4.7 \text{ cm}$$

Following the procedure described above the final deformation in the x and y directions were computed. Final voltage levels were used from Fig. 47 to compute the permanent deformation which were found to be,

$$\Delta x|_{\text{final}} = 4.0 \text{ cm}$$

and

$$\Delta y|_{\text{final}} = 0.25 \text{ cm}$$

It is seen from the foregoing example that the three-dimensional displacement transducer could be used to measure the maximum instantaneous as well as the permanent frame deformation. However, it was deemed desirable to verify the accuracy of the instrument. It was rather difficult to measure the maximum deflection by another means therefore, it was decided to measure the final deformation by an alternate technique.

To measure final deformation a special reference fixture was designed. Three pins were fixed on a heavy base and corresponded to three tapered holes in the chassis of the model tractor. A reference frame, made of channel sections, were also affixed to the same base plate. Thus, the tractor was in the same position with respect to the reference frames whenever placed on the pins. To measure the final deformation of frames, the model tractor was placed on the pins and two mutually orthogonal measurements were made to locate the impact point of the frame with respect to the reference frame. This process was repeated after overturning

the tractor. The difference in the two readings was the resulting frame deflection. This arrangement was provided for both model systems. Fig. 49 shows the 1/4-scale model tractor frame being measured before an overturn test. The static measurements were made using an inside micrometer length gage. Using this method the plastic deformation of the frame in the previous example was measured to be,

$$\Delta x|_{\text{final}} = 3.94 \text{ cm}$$

$$\Delta y|_{\text{final}} = 0.38 \text{ cm}$$

Considering the static measurements as the true values, the error in measuring Δx was found to be 1.5% where the same for Δy was 34%. The 34% error was considered to be too high.

As indicated earlier, the displacement transducer for 1/8 scale tractor was built and tested. Several mechanical problems were encountered with this transducer. The pulleys were equipped with fishing line as opposed to fine steel cable in the case of 1/4-scale one. It was suspected that there might be some stretching of the nylon strings which may cause error in the deflection readings. The nylon strings were replaced by fine steel wires. The tension in the negator motor was not enough to keep the wires taut which may also result in some error. Measurements were made to get an estimate of the error which was found to be approximately 28 percent. This was considered high for the experimental verification of the prediction equation.

Due to the large values of the errors in measuring dynamic deflections it was decided not to use the transducers. Since the purpose of the overturn testing was mainly to verify the modelling equations, it was decided to measure the permanent deflection using the static measurement technique.

The model frames for overturn testing were simple two dimensional welded or bolted portal frames of the same type as tested in static testing. Difficulties were encountered in building the welded type of frames because of the small

dimensions for the 1/8-scale frames. All frames were made of cold rolled mild steel.

Several frames of both welded and bolted types were tested. The dimensions of the frames were modified to achieve large deflections in the overturn tests. In general, bolted frames gave better results than welded frames. The problem of welding in building scale models was critical and should be studied further. It was not possible to find standard size stock for both model (1/8-scale) and the prototype (1/4-scale) frames which met the scaling requirements, so frame components for either the 1/8-scale or 1/4-scale models had to be machined to the desired dimensions. Frame height and widths and the outer dimensions of the mounting plates followed the general geometric scaling relationships. The cross-sectional dimensions for the frame components, thickness of the mounting plate and the size of the mounting bolts followed the scaling laws developed using the compensated distortion analysis. Fig. 33 shows a general drawing for the frames that were tested and Table 9 shows the dimensions of the frames that were tested.

Average frame deflection values obtained in the model overturn tests were presented in Table 10. Predicted values were computed using Eq. (101). It was assumed that $\sigma_{om} = \sigma_o$ and $A = 0.1$ for mild steel. Assumption that the yield stress for the model and prototype frame material was the same was not entirely correct. Tests to determine material yield stress indicated that some variability in this value was present. Yield stress for the prototype material was consistently higher. In Table 5 frames No. 2 and 3 were welded structures whereas 1, 4 and 5 were of the bolted type. A good general agreement is found between the predicted and the prototype data except for frame 3. Frame 3 was of the welded design and a careful inspection of frame 3 indicated that the inconsistency in the data may have been due to improper welding of the model frame. Frame No. 5 was a four-post type frame. It can be seen that the prediction equation can be used to predict prototype frame deflection using a properly designed model tractor and ROPS system.

CONCLUSIONS

From the previously presented analysis and the experimental data, the following conclusions were drawn:

1. In similitude modelling of ROPS deformation in the overturn testing, force scale is distorted if the same material is used for the model and the prototype.
2. Distortion in force scale can be compensated by distorting the scale factor for the frame area moment of inertia.
3. Distortion in the design condition related to material yield stress can be accounted for by developing a prediction factor using elastic theory of simple beams.
4. Results of the static testing support the hypothesis that prediction equations developed to predict behavior at the yield point can be used to predict complete elastic-plastic ROPS behavior.
5. Distortion in the time scale due to the strain rate dependence of the material yield stress can be accounted for by developing a prediction factor. However, if strain rate effects are neglected then an error of 6.7 percent may exist in predicting the ROPS load-deflection behavior from a 1/4-scale model.
6. It is feasible to develop scale-models of dissimilar cross-sectional members by including the shape factors in the analysis.
7. Tractor overturning motion can be predicted using small scale models.
8. Deflection of ROPS in an overturn test can be predicted using small models.

DISCUSSION AND SUMMARY

Part of the overall objectives of the project were to develop procedures to perform standard testing of ROPS using small scale models. Specifically, the objectives were to develop scaling relationships to build tractor and ROPS models, conduct the static, pendulum and overturn testing and predict the performance of the full scale ROPS. The previous sections presented the analyses according to the type of tests.

Equations (15a) and (15b) are the prediction equations for static testing. Due to the nature of loading, strain rate effects of steel do not pose any difficulty. By choosing a force scale factor as the square of the length scale factor it is possible to select the same material for model and prototype. This would not result in distortion of the force scale and a true model can be built. This approach may be appropriate because most manufacturers of ROPS are using the static test procedure for the reasons of ease of testing and better control of the test conditions. However, this approach is not applicable to either the pendulum test or the overturn test.

To develop appropriate scaling relationships to test models of ROPS in the pendulum test, strain rate effects had to be taken into consideration. Inclusion of these effects resulted in distortion of the time scaling. A prediction factor was developed to account for the errors due to such distortion. The strain rate effects were, however, found to be small. Force scaling was the same as for the static testing, i.e., a square of the length scale factor. The same prototype material is to be used for building model frames. Equation 30 is to be used to predict full scale frame deflection in pendulum test. Values of the prediction factor, δ , in Eq. (30) may be obtained from Fig. 18.

The foregoing analysis is applicable only if the model frames are not mounted on the model tractors. The frames should be mounted on model tractor chassis which in turn should be mounted on a rigid base. The standards recommend that the frame

be mounted on the tractor which should be tied down appropriately. Hence, a ROPS design based on model test results would be over designed because the pendulum energy which would otherwise be absorbed by the 'tie-down' and tractor tires would have to be absorbed by the ROPS. This is one of the areas where the test standards need improvements. To maintain the integrity of the ROPS mounting one must mount the model on appropriately designed model tractors. Design of a model tractor will be discussed later in the context of modelling of the overturn test. Design of the model impact pendulum for the pendulum test may pose some problems. The problem will be in trying to maintain both geometric and mass similarities in the model pendulum. The standard full scale pendulum weighs 4400 lb. and is a 27 in. cube which results in the density equal to 387 lb/ft^3 . For example, a quarter scale pendulum would have to weigh $4410/(4)^2 = 275.6 \text{ lb.}$ and would be 6.75 in. cube in size. This corresponds to a density of 1548.5 lb/ft^3 . Density of lead is only 710 lb/ft^3 . This problem, however, can be resolved by following the pendulum design used by Brown, et al., (1974).

Overturn testing of ROPS models imposed an additional requirement on the force scaling which was not encountered in the static or the pendulum tests. The scale factor for tractor weight must be the cube of the length scale factor. This requirement implied that the force scaling for the model frame design should follow the same relationship. This introduced a distortion for force scale if the same prototype material had to be used to design the model ROPS frame. Compensated distortion analysis technique was applied to compensate the distortion in the force scale. This was done by assuming that the frame deflection during tractor rollover was mainly due to bending. Since the compensated distortion approach requires knowledge of the phenomenon involved., theory of bending of simple elastic beams was employed in the analysis. Design conditions related to the area moment of inertia were deliberately distorted to compensate for distortion in the force scale. The elastic beam analysis was extended to the yield point by including material yield stress. This resulted in ...

an additional model design condition which could not be satisfied by the model. A prediction factor was derived analytically to account for this. It was then hypothesized that if the strain hardening effects were neglected then the prediction equations for ROPS behavior at yield point would be sufficient to predict the behavior throughout the entire plastic region of frame deformation. Two sets of prediction equations were developed for load-deformation behavior of the prototype frame. The first set was related to the elastic range while the second dealt with the yield point and the elastic-plastic region. Equations (97a) and (97b) are the prediction equations for the elastic deformation and Eq. (98a) and (98b) apply to the yield point and the elastic-plastic behavior.

In deriving the above equation, the force was considered static and, therefore, these equations could be used to predict behavior of the full size frame in the standard static test. The equations were modified to include dynamic loading. This involved replacing the static yield stress by dynamic yield stress. Effects of rate of strain were included in the analysis. Equations (99a) and (99b) are the prediction equations for dynamic load deformation in the overturn test was derived for dynamic plastic or elastic-plastic deformation (Eq. 101). This prediction equation applies to rear as well as to the sideways overturns.

Most ROPS are manufactured from high strength hollow-thin walled tubing which are available in certain standard sizes. Design of model ROPS while satisfying the design conditions would require sections of non-standard sizes and of extremely thin wall thicknesses. Because of this problem it was decided to consider a model of solid sectional steel, even though the prototypes were made of hollow sections. This obviously introduced distortion of the shape. A prediction equation was developed for this case, however, by introducing the shape factor into the analysis. Equations (108a) and (108b) are the prediction equations for static elasto-plastic load deformation behavior for dissimilar models. Equation (109) is the prediction equation for elasto-plastic deformation for this case in overturn tests.

It must be noted that the development of the prediction equations for frame deformation in an overturn test began by first developing equations for static behavior which were later modified to include dynamic loading. Therefore, this approach can be taken to perform static, pendulum and the overturn test. This would be advantageous if a comparison of the various testing procedures was desired.

Experimental testing of the frames to verify the static and the overturn testing generally supported the modelling approach. No testing was done to verify Eq. (15a) and (15b) because no difficulty in the similitude modelling was encountered. The modelling approach for overturn testing was first verified by static testing of a frame of similar cross-sectional members. Difficulty in building frames for 1/8-scale model requiring welding was experienced. This was shown by the predicted data when compared with that of the 1/4-scale frame. Models of dissimilar section were also built and showed a good correlation between the predicted and the measured data in the static testing. In order to verify model design and prediction equations against full scale frame test data, design specification and static test results were obtained from a manufacturer of one commercially manufactured two-post type ROPS. A model of solid sections was designed and tested. The results were used to predict the prototype behavior and the results compared well with the load-deformation behavior observed by the manufacturer.

Verification of the prediction equations for the overturn test was completed by overturning a 1/4-scale and a 1/8-scale model tractor on their respective ramps. The impact surface for both models was concrete floor which was assumed to be a non-energy absorbing surface. This satisfied the scaling requirements for surface characteristics for the impact surface. Simple two-post type of frames were built and tested. Both welded and bolted type of designs were tested. The predicted results for the bolted type of frames compared well with those of the prototype. Welded frames, due to fabrication problems, showed large prediction errors. The fabrication problem in case of the welded frames was found only in the 1/8-scale model. The

problem was due to the small sizes of the frame components. Sometimes steel sections as small as 0.125 x 0.125 in. were welded by arc welding process. This problem will not be present when building 1/4-scale model frames to predict the deflection of full scale ROPS. Permanent plastic deformation was measured for both the 1/4-scale and the 1/8-scale model frames. Tests with the four-post types of model ROPS indicated that the modelling approach was applicable to various frame types.

SUGGESTIONS FOR FURTHER RESEARCH

1. Although the strain hardening effects were considered small and were not included, implications of such assumption should be analyzed in detail.
2. Similitude modelling may be further verified by measuring maximum frame deflections during the overturn tests by designing suitable instrumentation. This may be done by either perfecting the design already discussed earlier or by taking a completely new approach.
3. The compensated distortion modelling approach was verified against full scale ROPS only in the static testing. Verification with full scale prototypes in overturn test will add further to the analysis.
4. To understand effects of the various system variables affecting tractor overturning and the resulting frame behavior, an experimental schedule should be developed. Frame deflections should be measured for a range of values of the dimensionless system variables.
5. Similitude modelling may be applied to investigate the relationship of the required amount of strain energy in the standard static and pendulum test with tractor weight. The scale model testing approach may provide an excellent way to study this relationship for an increasingly important class of four-wheel drive agricultural tractors.
6. Relative severity of the standard static, the pendulum and the overturn tests using scale models of tractors and ROPS should be studied. This study will provide basis for making these testing procedures more compatible.

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LIST OF SYMBOLS

<u>Symbol</u>	<u>Description</u>	<u>Dimension</u>
A	Strain rate parameter	(-)
B	Bending moment	(FL)
B _p	Plastic moment	(FL)
B _y	Yield moment	(FL)
b	Width of beam	(L)
c	Strain rate parameter	(T ⁻¹)
C	Tire damping coefficient	(FL ⁻¹ T)
D	Strain rate parameter	(T)
d	ROPS deflection	(L)
E	Modulus of elasticity	(FL ⁻²)
E _{ir} (E' _{ir})	Input energy in rear static test, lb-in(Kg-cm)	(FL)
E _{is} (E' _{is})	Input energy in side static test, lb-in(Kg-cm)	(FL)
E _u	Strain energy absorbed by the structure	(-)
E _w	Pendulum kinetic energy	{FL}
F	Dimension of force	(-)
f	Shape factor	(-)
G	Modulus of rigidity	(FL ⁻²)
g	Acceleration due to gravity	(LT ⁻²)
H, H'	Pendulum height before impact, in (cm)	(L)
h	Thickness of beam	(L)
I _α	Tractor mass moment of inertia	(FLT ²)
I	Area moment of inertia	(L ⁴)
K ₁ , K ₂	Constants in theoretically derived equations	(-)
K' ₁ , K' ₂	Constants in experimental equation	(-)
k	Tire spring constant	(FL ⁻¹)

L	Load	(F)
L	Dimension of length	(-)
ℓ	Characteristic length	(L)
$\ell_{c.g.}$	Pivot point to center of gravity distance	(L)
ℓ_i	Distance of impact point to pivot point	(L)
M	Tractor or pendulum mass	$(FL^{-1}T^2)$
M_{po}	Dynamic yield moment	(FL)
M_{pso}	Static yield moment	(FL)
m	Subscript - denotes model system	(-)
N	Strain energy	(FL)
n	Geometric scale factor	(-)
P	Applied force	
p	Strain rate parameter	(-)
s	Soil surface characteristic	$(F^a L^b T^c)$
s	Size of impact piece	(L)
T	Dimension of time	(-)
t	Characteristic time	(T)
t_o	Time to reach yield	(T)
v	Tractor or pendulum velocity	(LT^{-1})
W, (W')	Tractor weight, lb (Kg)	(F)
x	Distance along beam length	(L)
y	Generalized beam deflection	(L)
Z	Section modulus	(L^3)
α, β, γ	Distortion factors	(-)
δ, δ'	Prediction factors	(-)
Δ	Prefix denotes error in the variable	(-)

ϵ	Strain	(-)
$\dot{\epsilon}$	Rate of strain	(F ¹)
λ	Geometry	(L)
θ	Angular displacement of pendulum	(-)
θ_i	Pendulum angular position at maximum beam deflection	(-)
μ	Tire-surface friction coefficient	(-)
ρ	Radius of curvature	(L)
ρ_r	ROPS material density	(FL ⁻⁴ T ²)
ρ_t	Tractor density	(FL ⁻⁴ T ²)
σ_o	Static yield stress	(FL ⁻²)
σ_y	Dynamic yield stress	(FL ⁻²)
ϕ	Angle of impact	(-)
Φ	Function	(-)
τ_o	Static shear stress	(FL ⁻²)
π	Dimensionless term	(-)

Table 2. List of pertinent variables to the ROPS pendulum test.

Symbols	Variables	Dimensions
ℓ	Height of beam	L
b	Width of beam	L
h	Thickness of beam	L
ℓ_i	Distance of impact point to pivot point	L
s	Size of semi-cylindrical impact piece	L
M	Pendulum mass	$FL^{-1}T^2$
E	Modulus of elasticity of beam material	FL^{-2}
σ_o	Static yield stress of beam material	FL^{-2}
D	Strain rate parameter	T
A	Strain rate parameter	-
v	Impact velocity	LT^{-1}
ϕ	Angle of impact	-
g	Acceleration due to gravity	LT^{-2}
t	Characteristics time	T
d	Beam deflection	L

Table 3. Levels of independent π -terms in the experiments.

$\pi_1 = \frac{d}{\ell}$	$\pi_2 = \frac{b}{\ell}$	$\pi_3 = \frac{h}{b}$	$\pi_4 = \frac{\ell}{D^2 g}$	$\pi_5 = \frac{EL^2}{mg} \times 10^{-6}$	$\pi_6 = \frac{v^2}{g\ell}$
Observed Parameter	0.1 0.125 0.150 0.200	0.4	0.448	16.34	0.96
	0.125	0.3 0.4 0.5 0.6	0.448	16.34	0.96
	0.125	0.4	0.224 0.448 0.672 0.896	16.34	0.96
	0.125	0.4	0.448	9.8 10.57 12.37 16.34	0.96
	0.125	0.4	0.448	16.34	0.63 0.96 1.30 1.65

Table 4. Values of the system variables.

Deflection	Beam Length	Beam Width	Beam Depth	Pendulum Weight	Pendulum Velocity
d	ℓ (cm)	c(cm)	h(cm)	Mg(kg)	v(cm/sec)
Observed Variable	12.70	1.270	0.508	20.52	109.22
		1.587	0.635		
		1.905	0.762		
		2.540	1.016		
	12.70	1.587	0.476	20.52	109.22
			0.635		
			0.794		
			0.952		
	6.34	0.794	0.317	5.13	77.22
	12.70	1.587	0.635	20.52	109.22
	19.05	2.381	0.952	46.19	133.76
	25.40	3.175	1.219	82.10	154.46
	12.70	1.587	0.635	34.72	109.22
				32.18	
				27.50	
				20.52	
	12.70	1.587	0.635	20.52	88.89
					109.22
					126.99
					143.25

$\phi = 90^\circ$; $D = 0.17 \text{ sec}^{-1}$

Table 5. List of pertinent variables.

Symbols	Variables	Dimensions
λ	Geometry	L
ℓ	Characteristic length	L
M	Tractor mass	$FL^{-1}T^2$
I_{α}	Tractor mass moment of inertia	FLT^2
ρ_t	Tractor density	$FL^{-4}T^2$
k	Tire spring constant	FL^{-1}
c	Tire damping coefficient	$FL^{-1}T$
μ	Tire-surface frictional coefficient	-
E	Modulus of elasticity of ROPS	FT^{-2}
σ_y	Yield stress	FL^{-2}
ρ_r	Material density	$FL^{-4}T^2$
d	Frame deflection	L
s	Surface characteristic	$F^a L^b T^c$
v	Tractor velocity	LT^{-1}
t	Time	T
g	Acceleration due to gravity	LT^{-2}

L = Dimension of Length, T = Dimension of Time, F = Dimension of Force

Table 6. Variable scale factors.

Symbols	Variable	Scale Factor
λ	Geometry	n^{-1}
M	Mass	n^{-2}
I_{α}	Mass moment of inertia	n^{-4}
ρ_t	Tractor density	n
k	Tire spring constant	n^{-1}
c	Tire damping coefficient	$n^{-\frac{3}{2}}$
μ	Tire surface friction coefficient	1
v	Velocity	$n^{-\frac{1}{2}}$
t	Time	$n^{-\frac{1}{2}}$

Table 7. Summary of the relationships of the various physical tractor variables with the PTO horsepower.

Physical variable (in)	a	b	r
Front Tread, Min	0.12	45.09	0.72
Front Tread, Max	0.13	71.82	0.70
Rear Tread, Min	0.16	45.17	0.74
Rear Tread, Max	0.35	64.29	0.68
Front Tire Width	0.04	5.39	0.17
Front Tire Dia.	0.01	15.35	0.32
Rear Tire Width	0.06	11.71	0.75
Rear Tire Dia.	0.08	26.34	0.54
Wheel Base	0.39	63.19	0.85
Tractor Weight (lb)	80.97	756.19	0.92

Model: $Y = ax + b$
 Y = Physical variable
 X = Horsepower (PTO)
 a, b = Regression constants
 r = Regression coefficients

Table 8. Design variables for full size and model tractors.

Variable	Full Size	Scale Factors	1/4 Scale Model	1/8 Scale Model
<u>Weights (kg)</u>				
Total	5150	n^3	80.5	10.06
Front-end	316	n^3	4.94	0.617
Rear Tire	465	n^3	7.26	0.908
Front Tire	56	n^3	0.87	0.109
<u>Location of Center of Gravity (cm)</u>				
Ahead of rear axle	78	n	19.5	9.75
Above rear axle	19.6	n	4.9	2.45
<u>Mass Moments of Inertia* (kg-cm-sec²)</u>				
I_{xx}	28134	n^5	27.47	0.858
I_{yy}	64659	n^5	63.14	1.973
I_{zz}	59175	n^5	67.56	2.111
I_{xz}	-3227	n^5	-3.15	-0.098
<u>Geometry (cm)</u>				
Wheel Base	269.2	n	67.3	33.65
Rear Track	203.2	n	50.8	25.4
Rear Tire	18.4-39	n	4.0 x 8	2.50 x 4
Front Tire	11L-15	n	2.50 x 4	1.25 x 2

*For definition see Davis (1974).

Table 9. Specifications of experimental frames.

Frame No.	Dimensions (cm)										Remarks
		A	B	C	D	E	F	G	H	I	
1	1/8 scale	22.86	12.7	5.08	2.54	1.27	0.315	0.635	0.317		Bolted design type.
	1/4 scale	48.72	25.4	10.16	5.08	2.54	0.754	1.51	0.754		
2	1/8 scale	22.86	12.7	5.08	2.54		0.317	0.317	0.317		Welded upper corners with no reinforcement
	1/4 scale	48.72	25.4	10.16	5.08		0.754	0.754	0.754		
3	1/8 scale	25.4	12.7	5.08			0.317	0.317	0.317		Welded upper corners with no reinforcement
	1/4 scale	50.8	25.4	10.16			0.754	0.754	0.754		
4	1/8 scale	22.86	12.7	5.08	2.56	1.27	0.267	0.534			Bolted type
	1/4 scale	48.72	25.4	10.16	5.08	2.54	0.635	1.27			
5	1/8 scale	22.86	12.7	5.08	2.56	1.27	0.267	0.534	0.165	0.317	Fourpost bolted type
	1/4 scale	48.72	25.4	10.16	5.08	2.54	0.635	1.27	0.376	0.635	

Table 10. Predicted and measured frame deflections in the model overturn tests.

Frame No.	Average Deflection (cm)		Remarks
	Predicted	Measured	
1	1.93	2.59	Maximum Deflection
2	3.12	3.61	Permanent Deflection
3	0.99	2.54	Permanent Deflection
4	3.78	3.79	Permanent Deflection
5	2.49	2.62	Permanent Deflection

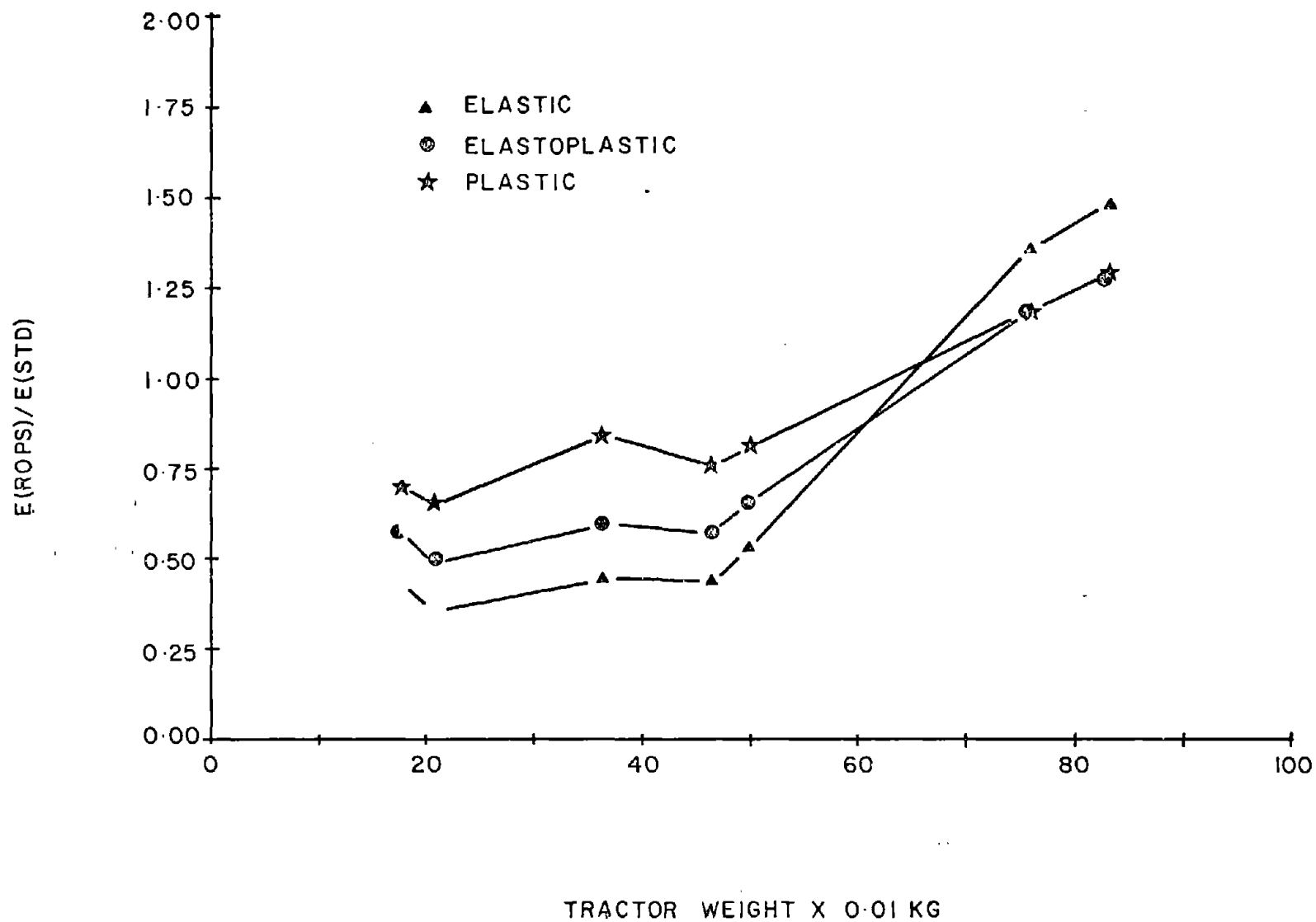


Figure 1. Effect of tractor weight and the type of frame deformation on the energy absorption with respect to the standard requirements.

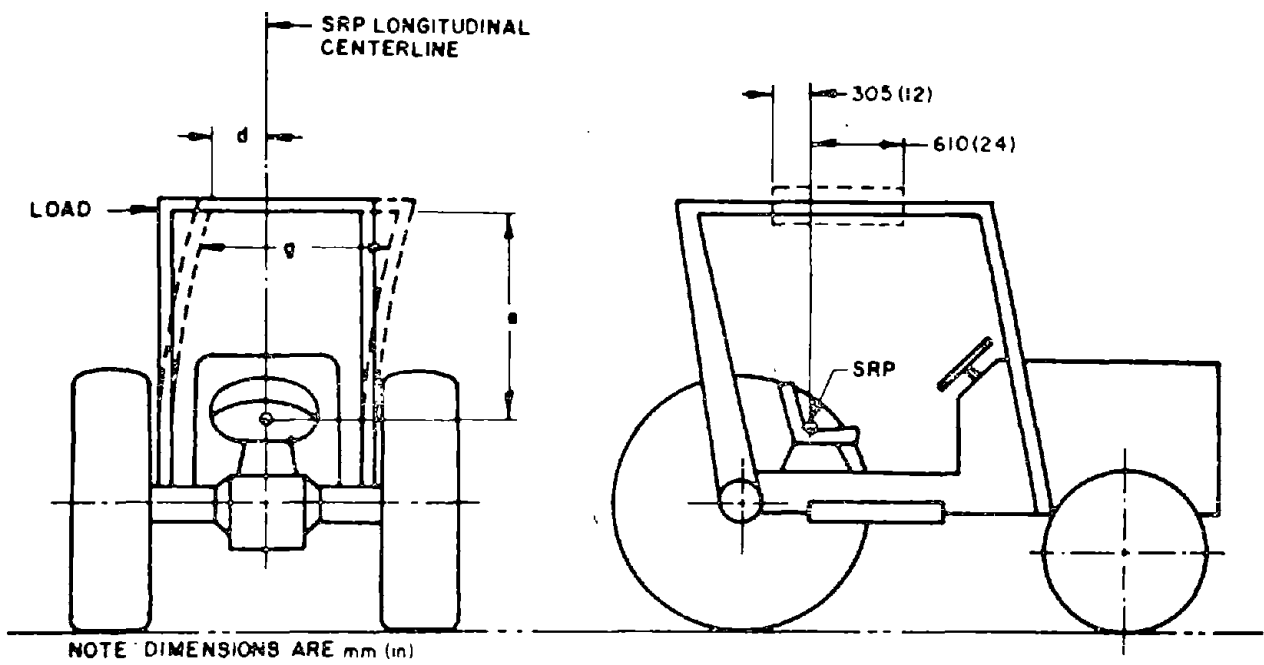


Figure 2. Typical side load application (Ref. ASAE Standard S383).

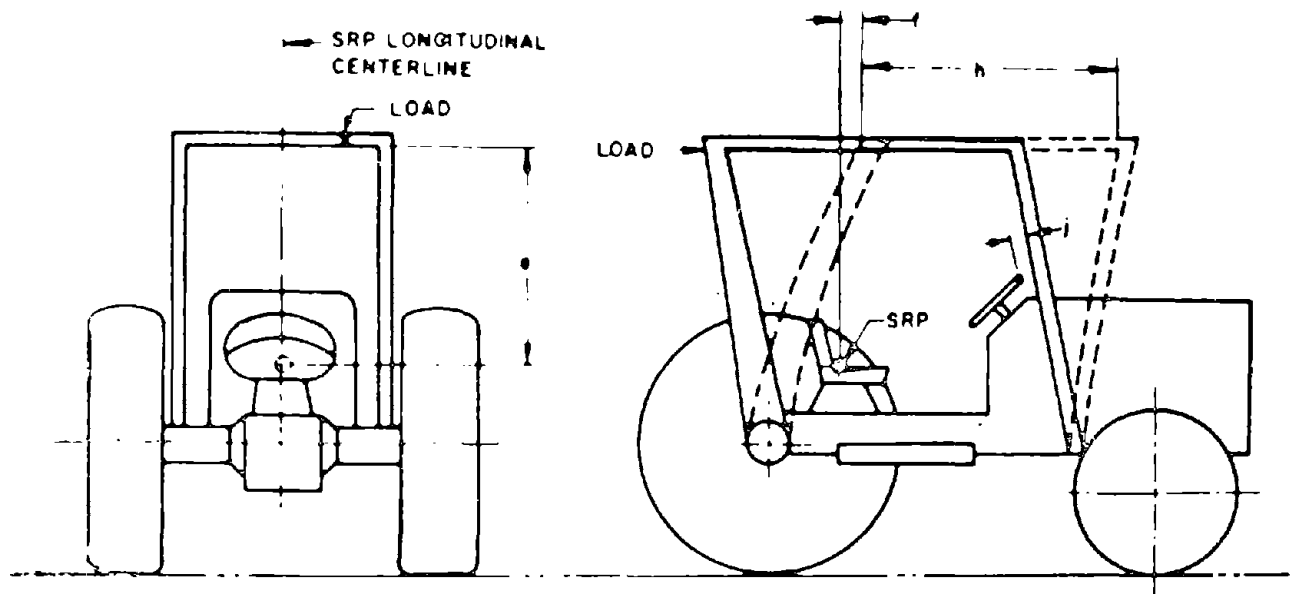


Figure 3. Typical rear load application (Ref. ASAE Standard S383).

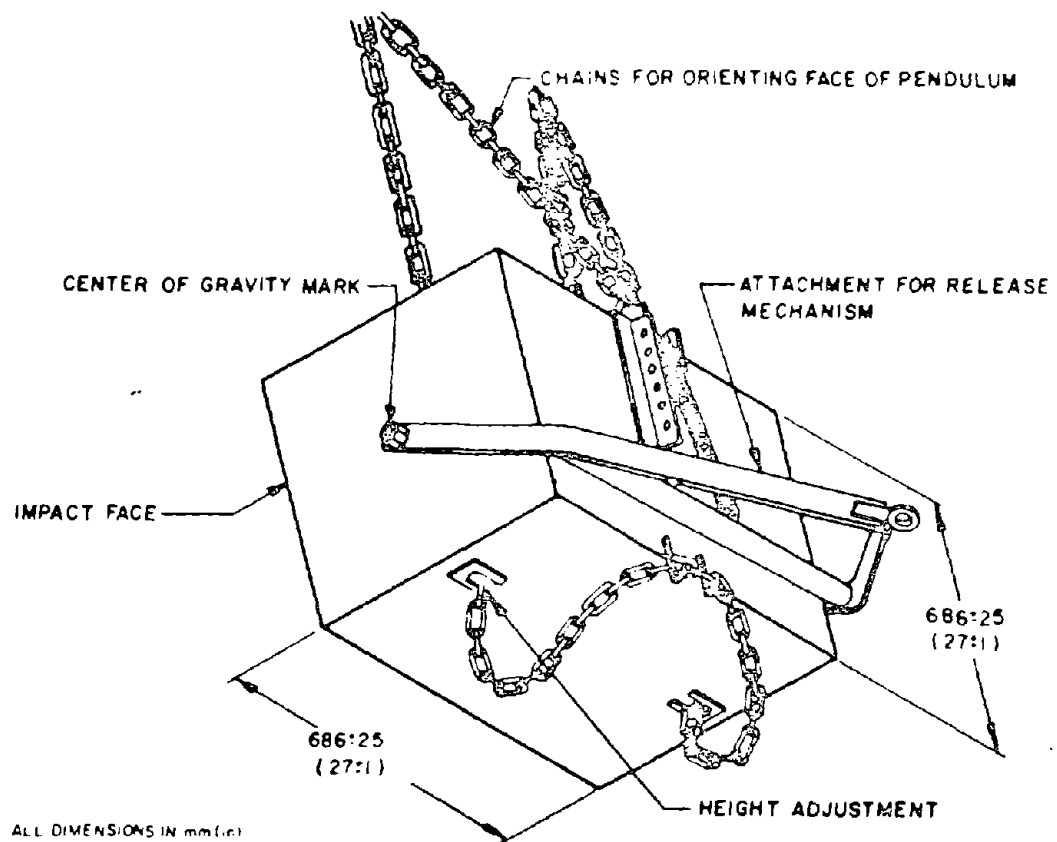


Figure 4. Standard impact pendulum (Ref. ASAE Standard S383).

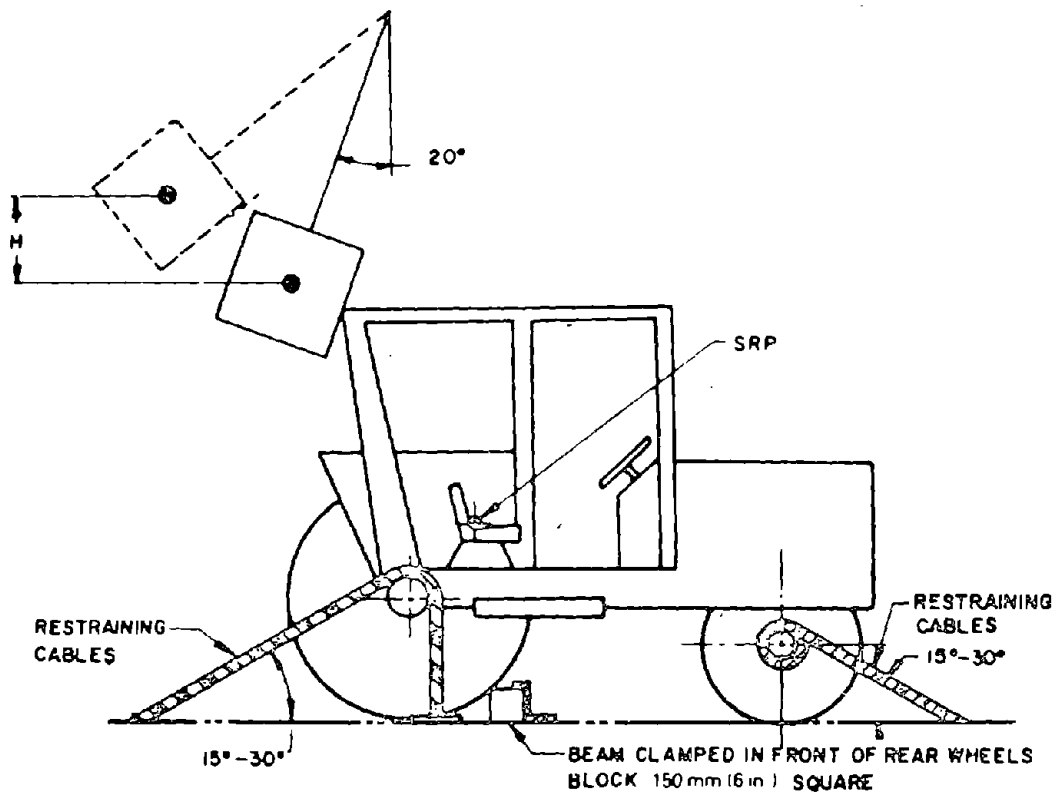


Figure 5. Typical rear impact application (Ref. ASAE Standard S383).

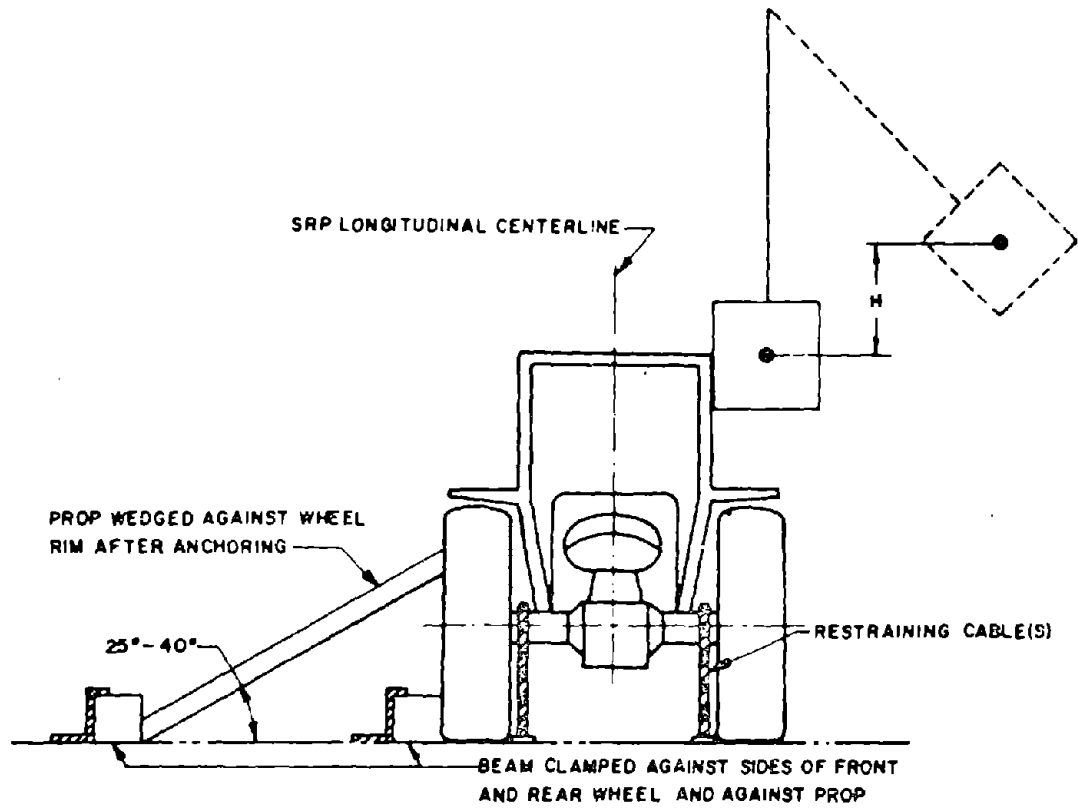


Figure 6. Typical side impact application (Ref. ASAE Standard S383).

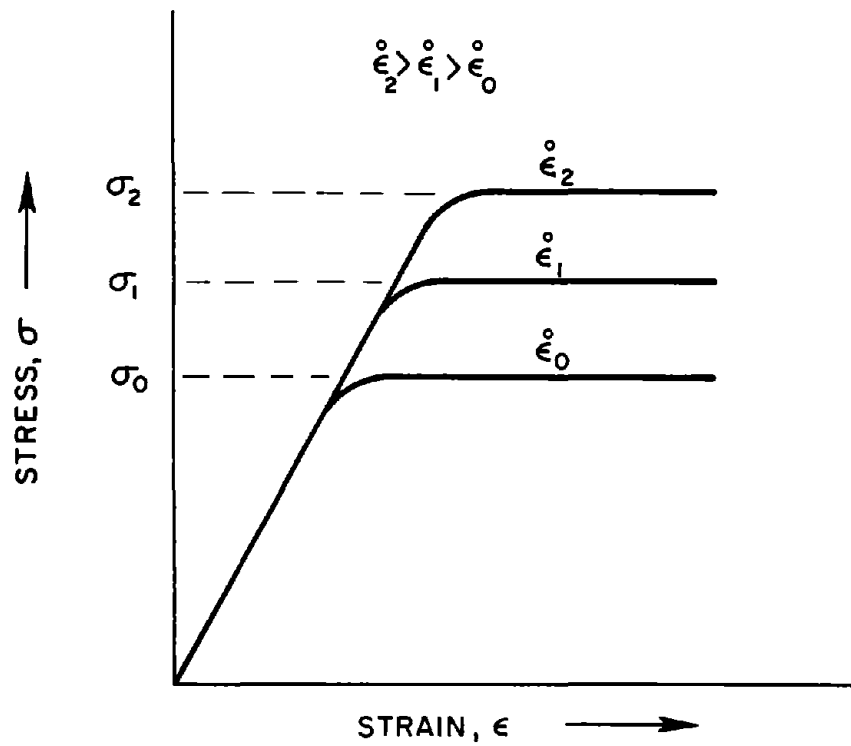


Figure 7. An idealized stress-strain diagram showing the effects of rate of strain on yield stress.

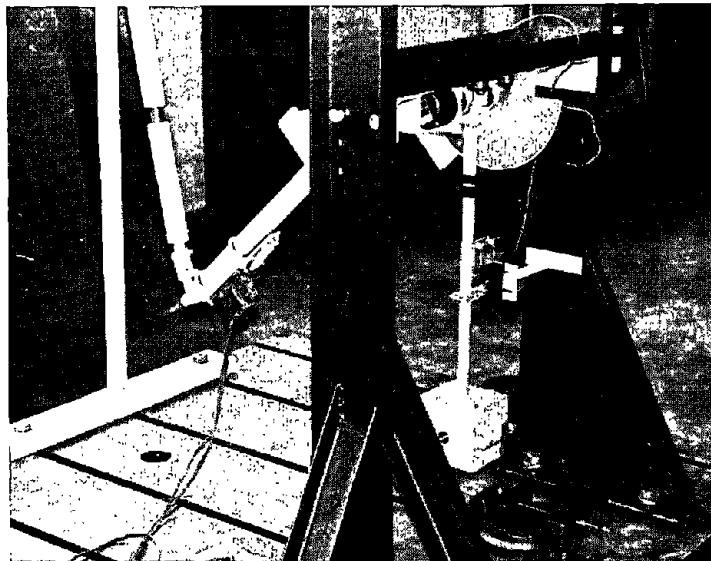


Figure 8. A general view of the experimental equipment showing the impact pendulum, the release mechanism, an angular displacement transducer and a cantilever beam.

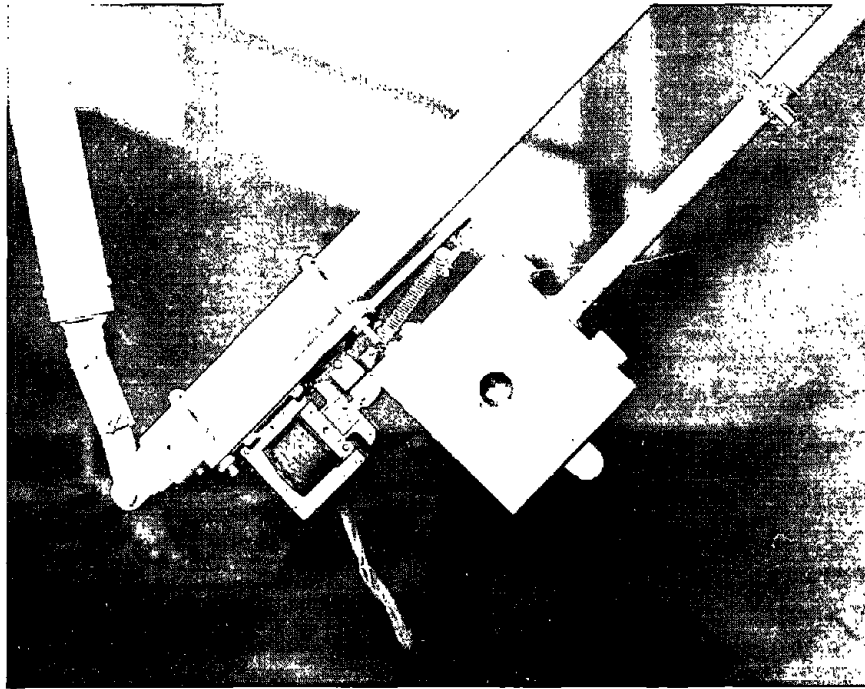


Figure 9. Pendulum release mechanism.

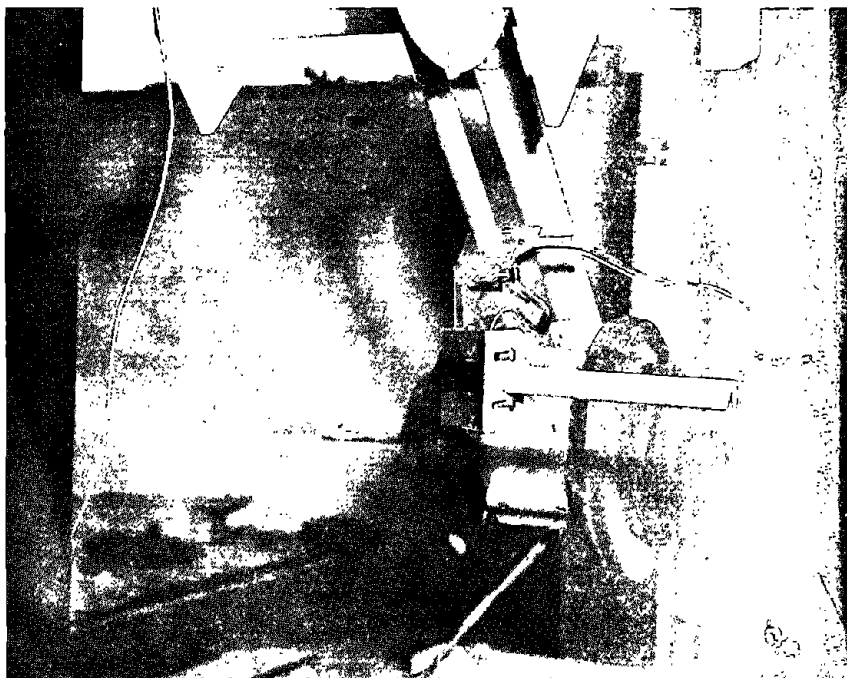


Figure 10. Method of adding weights to the pendulum.

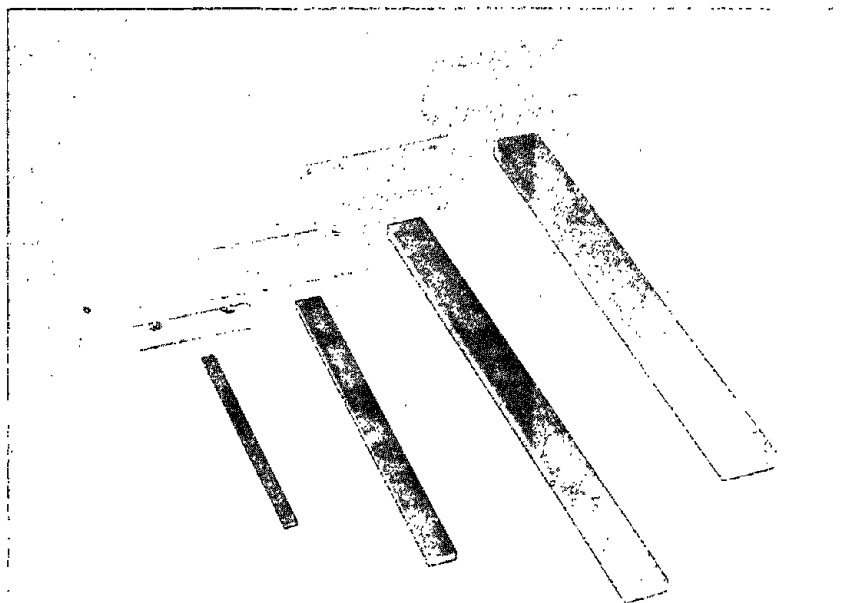


Figure 11. Different size beams and the impact pieces.



Figure 12. A typically deformed beam after impact by the pendulum. Figure also illustrates method of measuring permanent deflection.

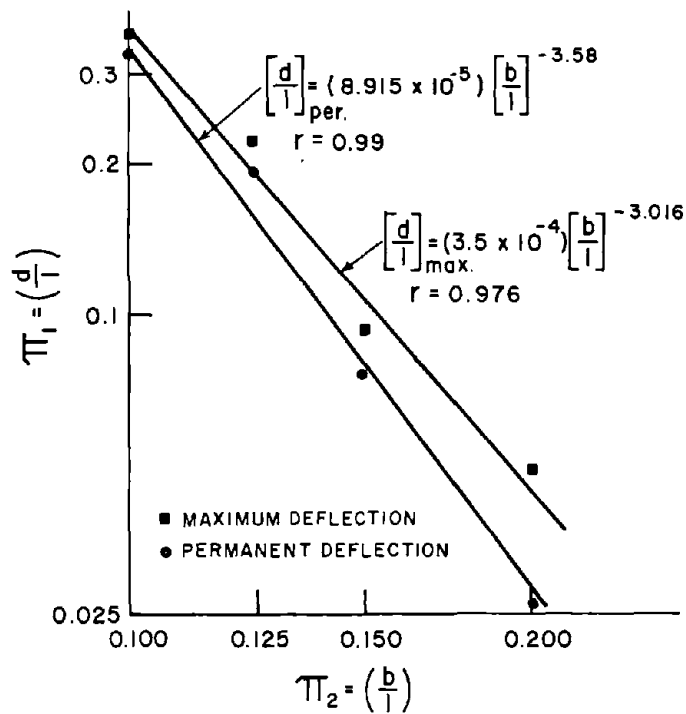


Figure 13. Effects of changing b/l on beam deflection. Other π -terms were held constant ($\pi_3 = 0.4$; $\pi_4 = 0.448$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

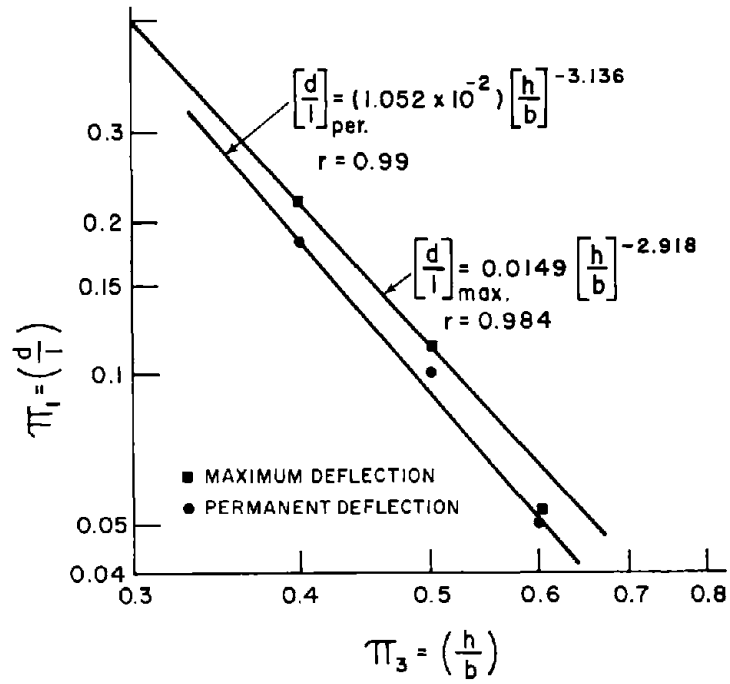


Figure 14. Effects of changing h/b on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_4 = 0.448$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

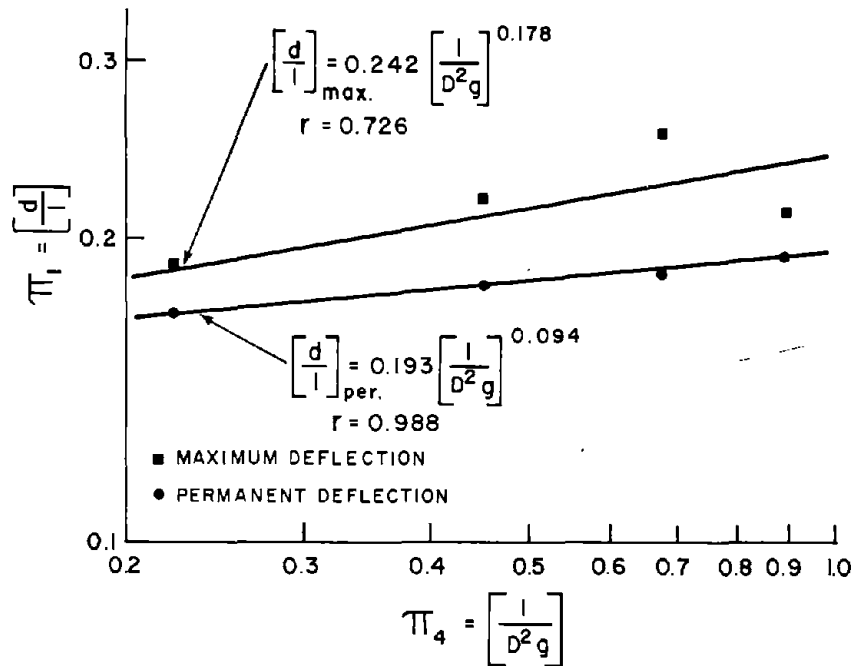


Figure 15. Effects of changing l/D^2g on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

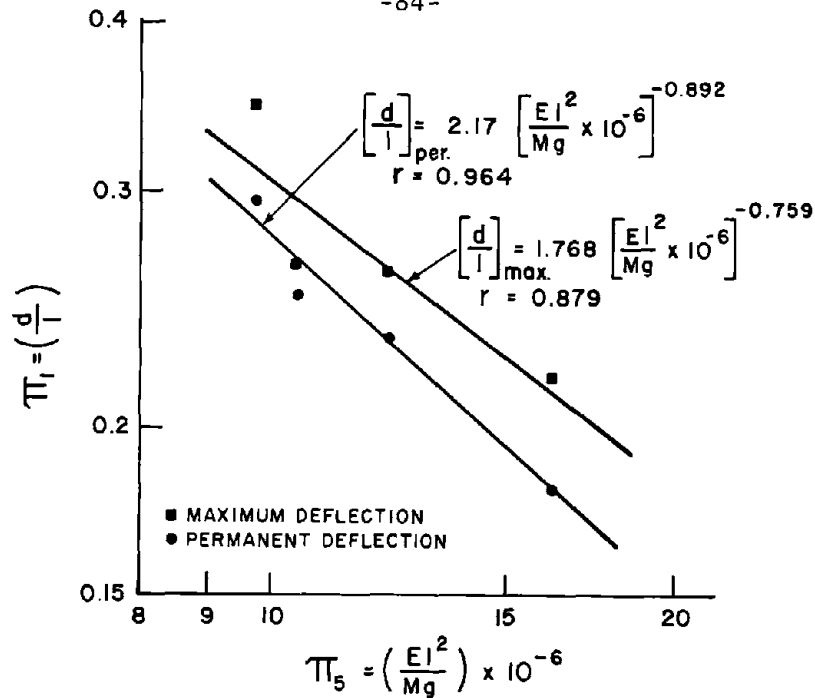


Figure 16. Effects of changing EI^2/Mg on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_4 = 0.448$; $\pi_6 = 0.96$).

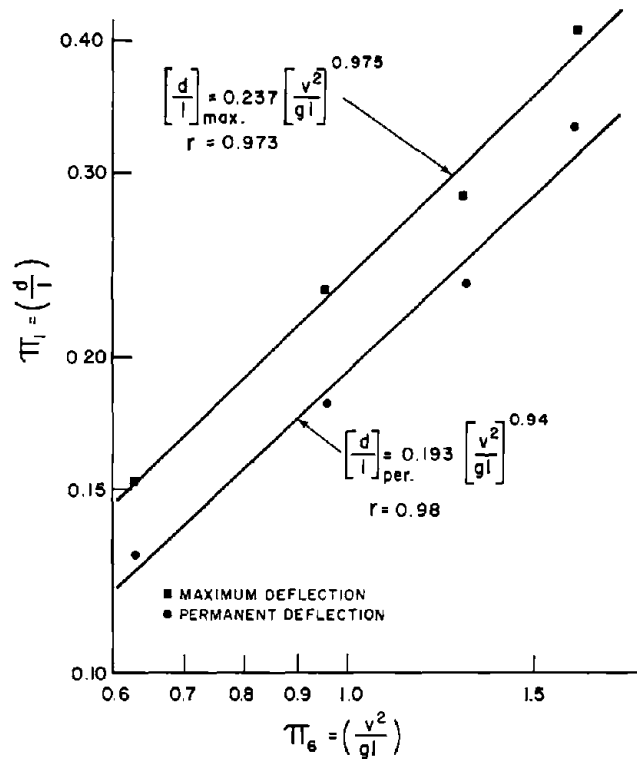


Figure 17. Effects of changing v^2/gl on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_4 = 0.448$, $\pi_5 = 16.34$).

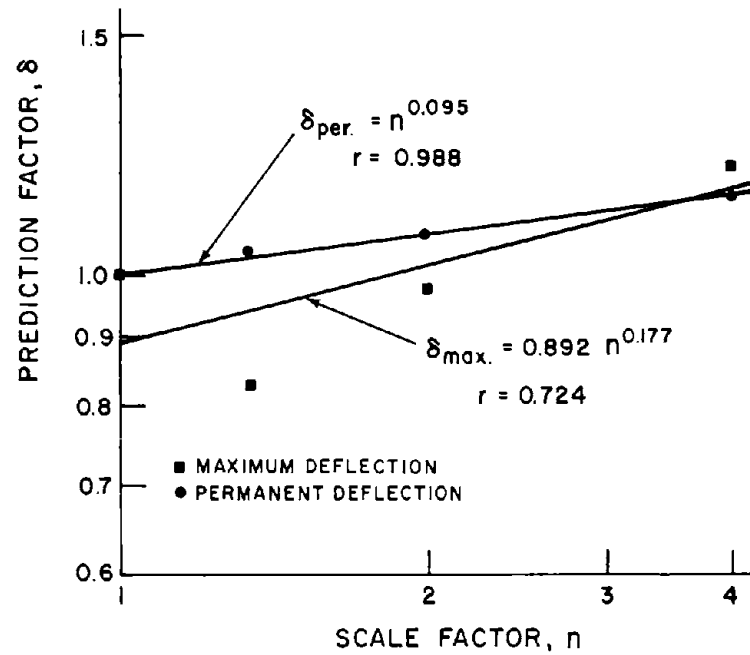


Figure 18. Effects of distortion in the strain rate design condition on prediction factors for dimensionless maximum and permanent deflections.

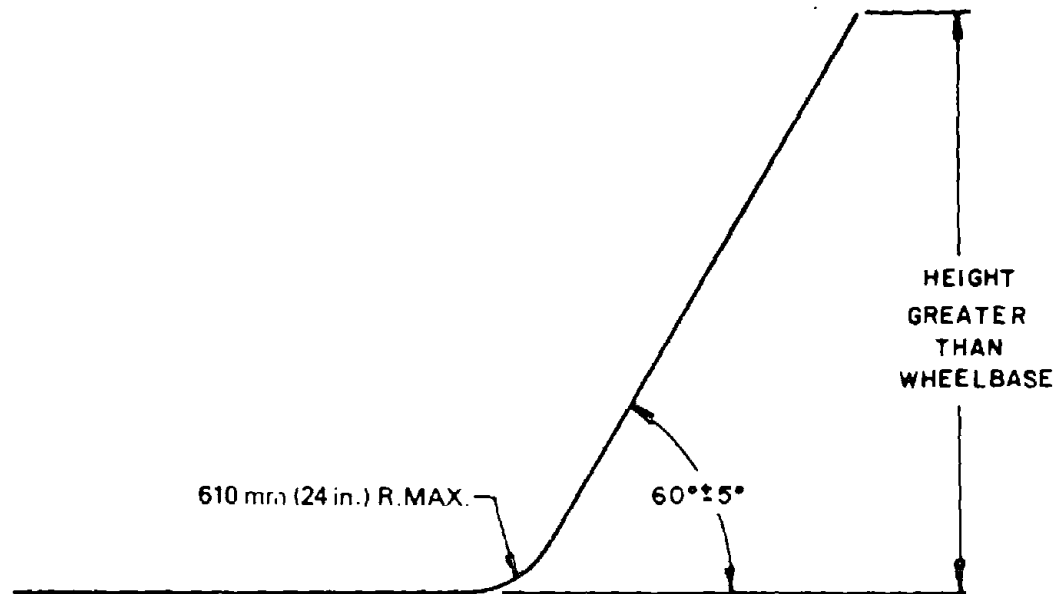


Figure 20. Rear overturn bank (ASAE Standard S383).

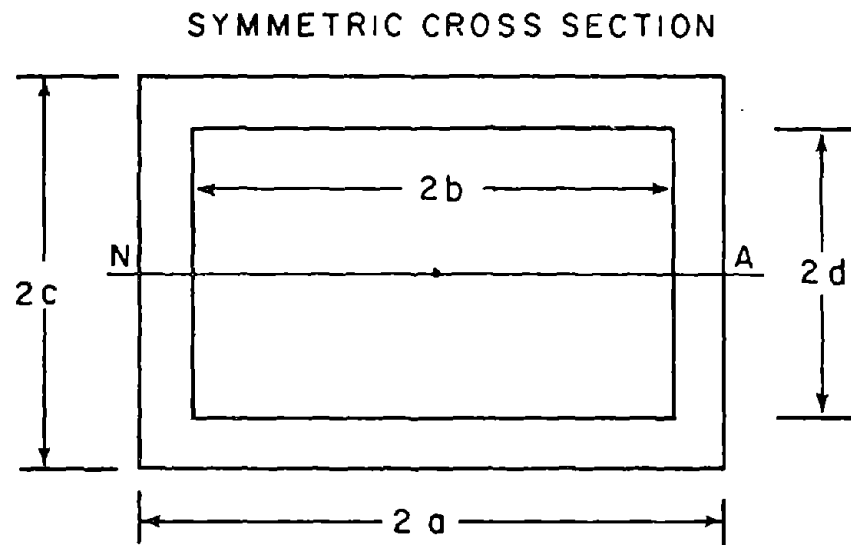


Figure 21. Symmetric rectangular hollow cross-section for calculation of shape factor.

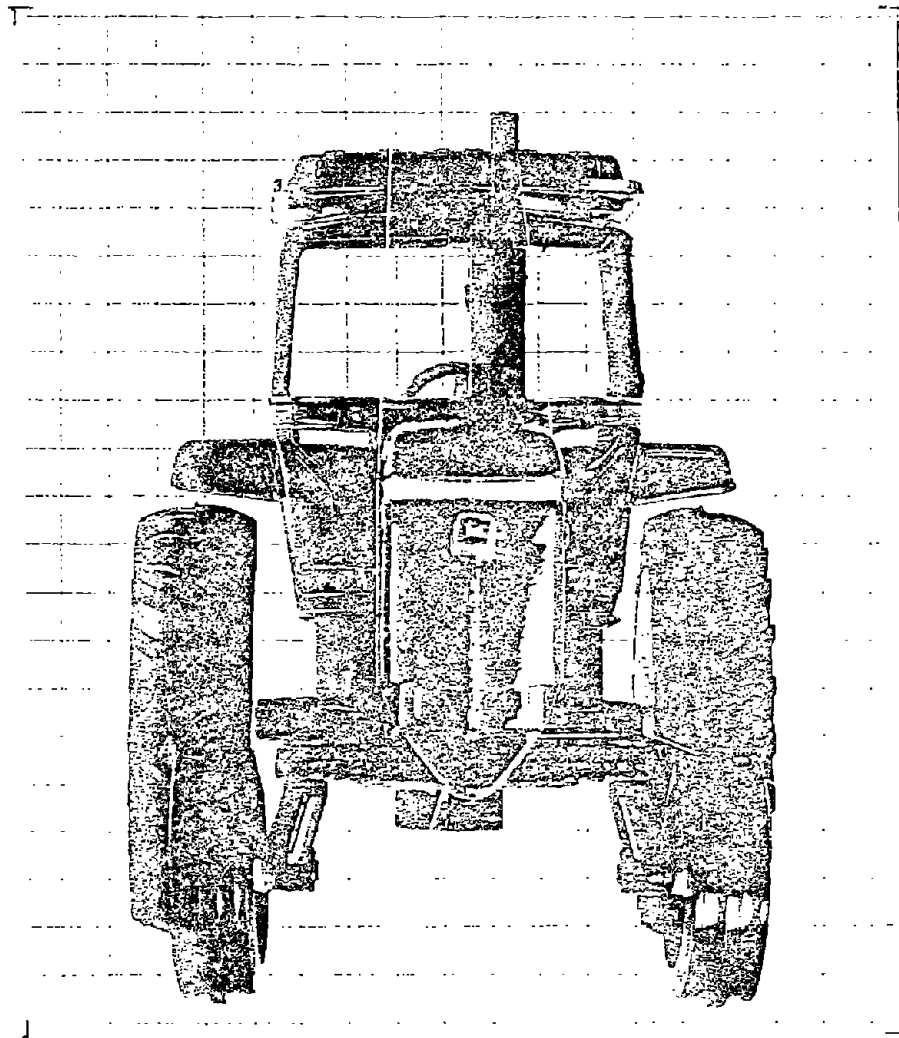


Figure 22. Front view of a 1/16-scale model toy tractor.

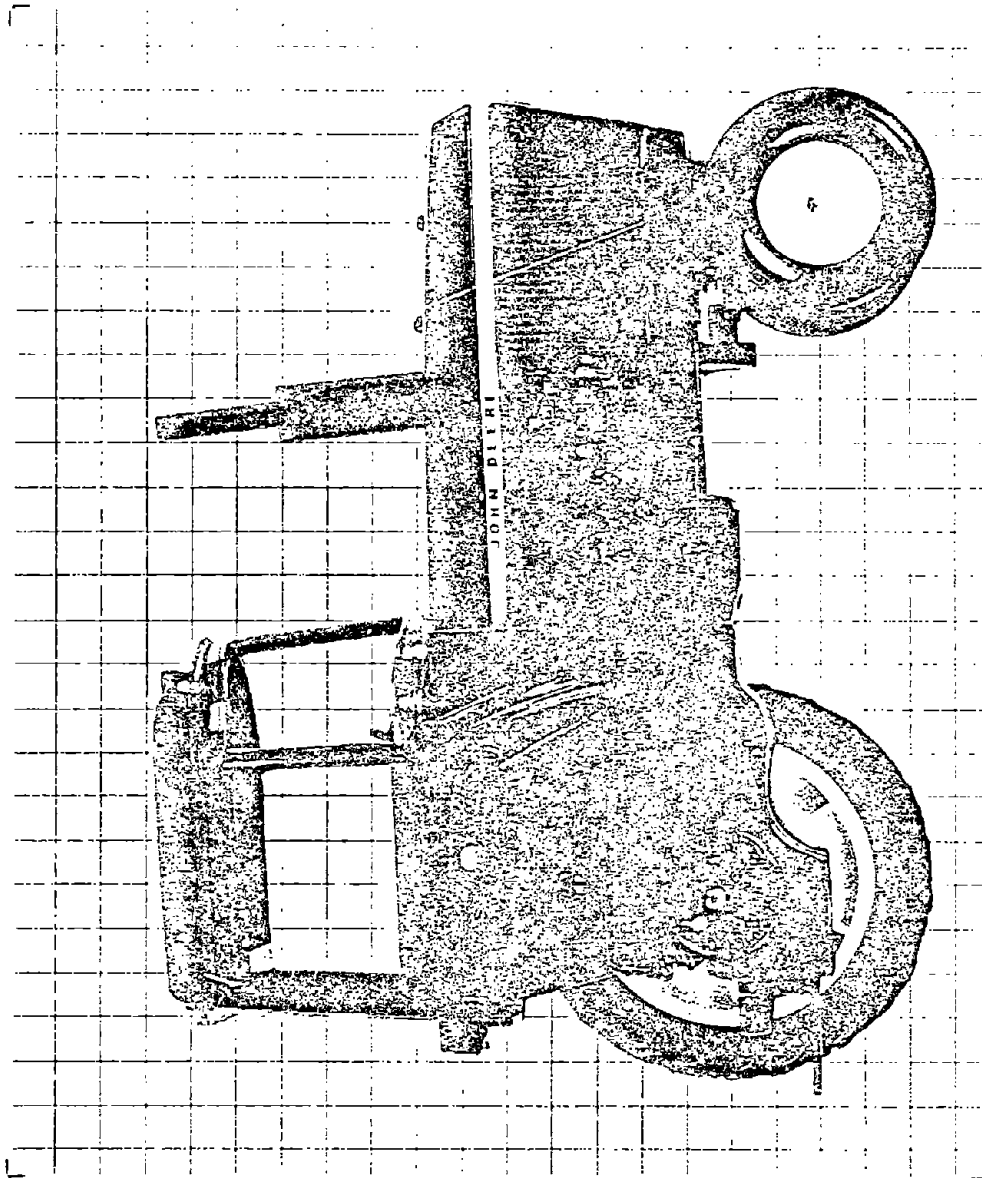


Figure 23. Side view of a 1/16-scale model toy tractor.

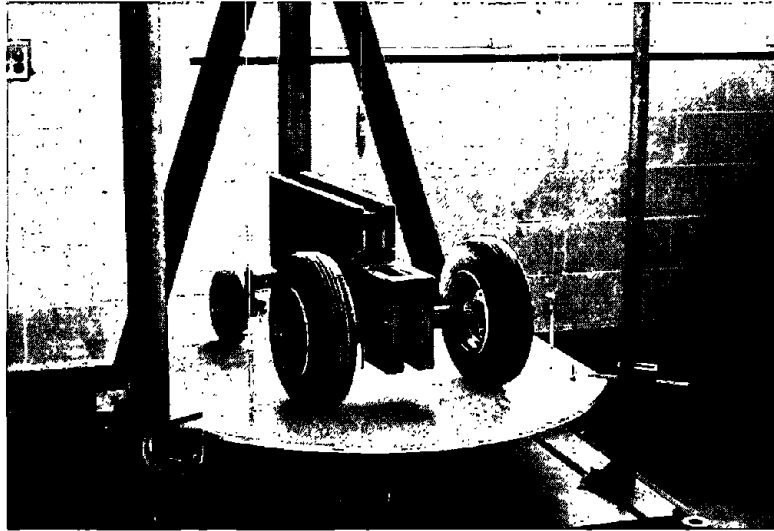


Figure 24. Determination of mass moments of inertia for model tractors using trifilar pendulum.

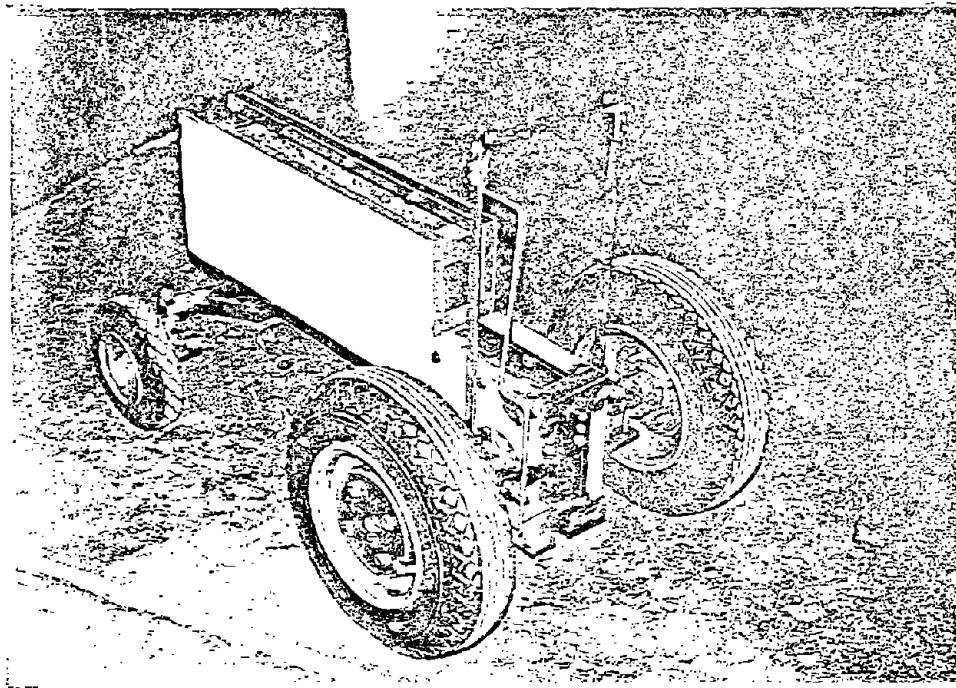


Figure 25. The quarter scale model tractor and ROPS.

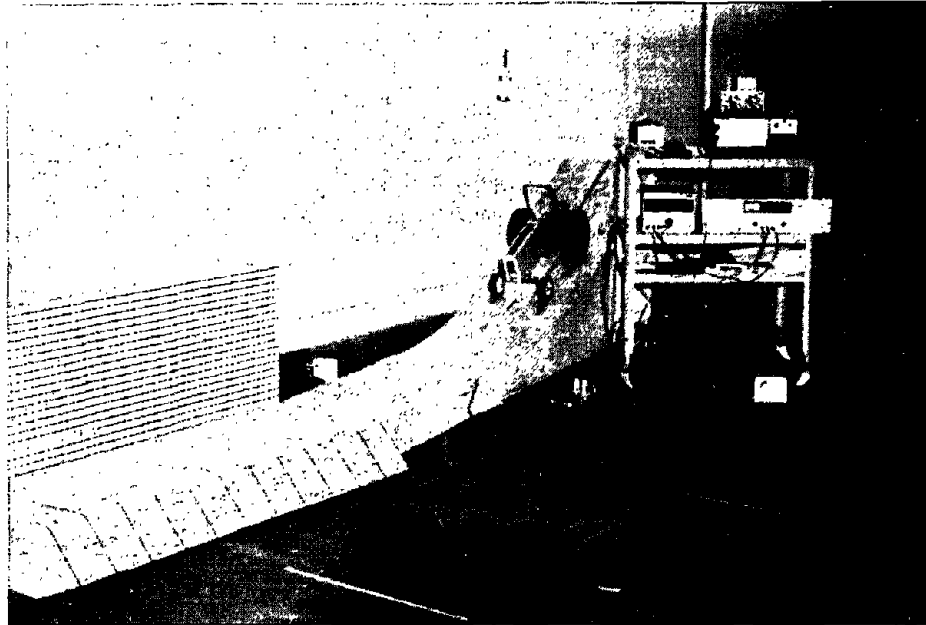


Figure 26. Experimental setup for model overturn test. Shown are the 1/8-scale tractor model on the starting ramp in the release position, the overturn ramp, the grid board for the motion analysis and the deflection measuring instrumentation.

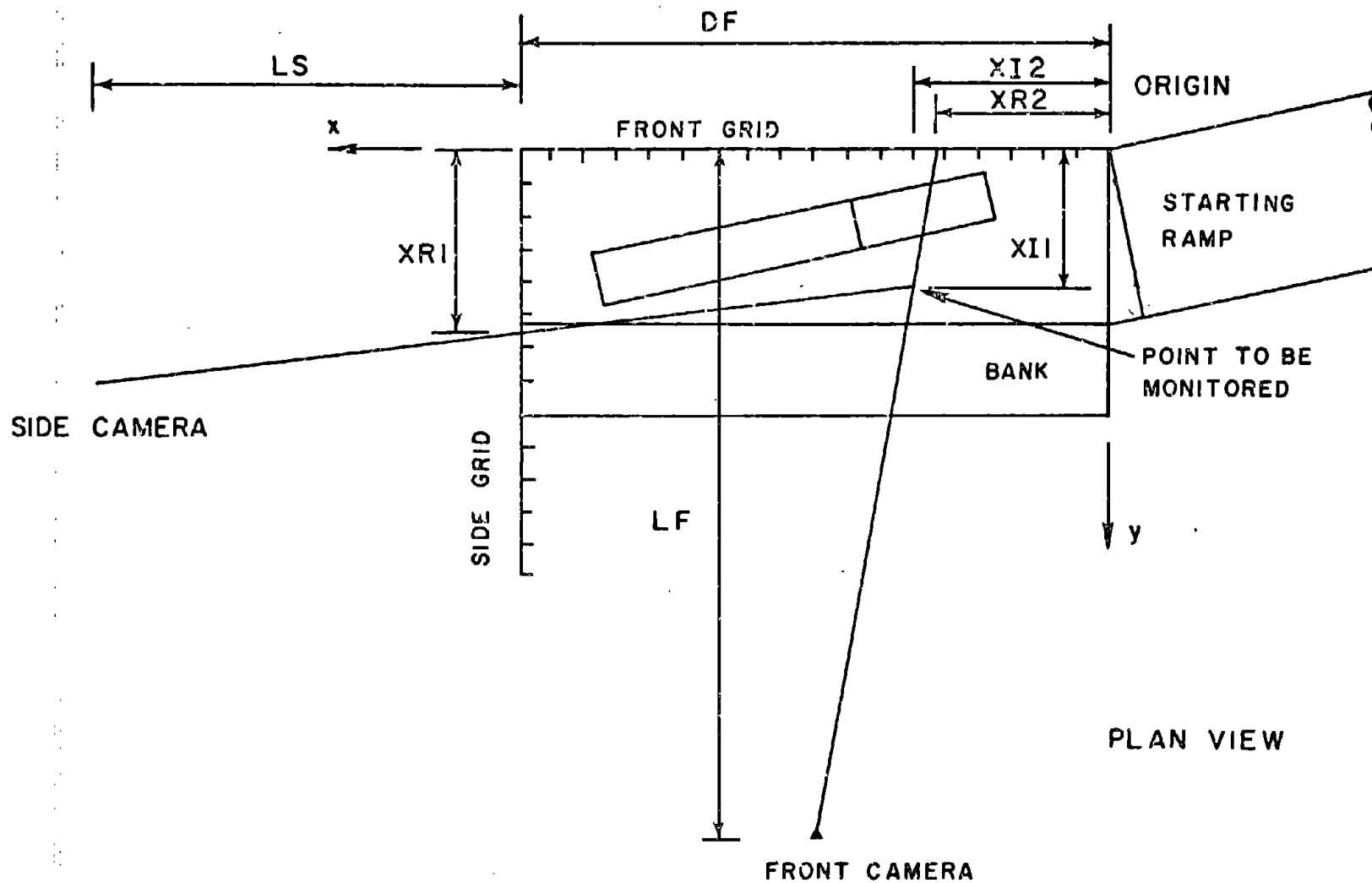


Figure 27. Plan view of the layout for high speed motion picture analysis.

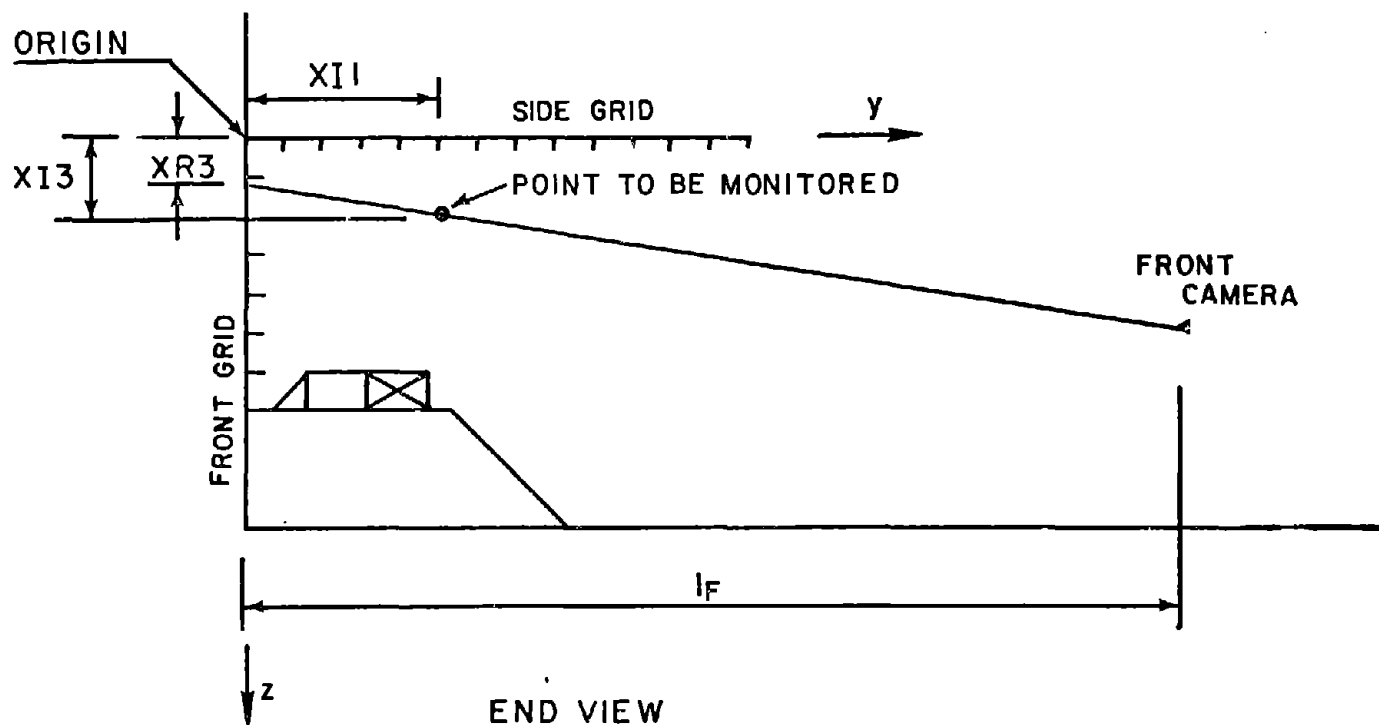


Figure 28. End view of the layout for high speed motion picture analysis.

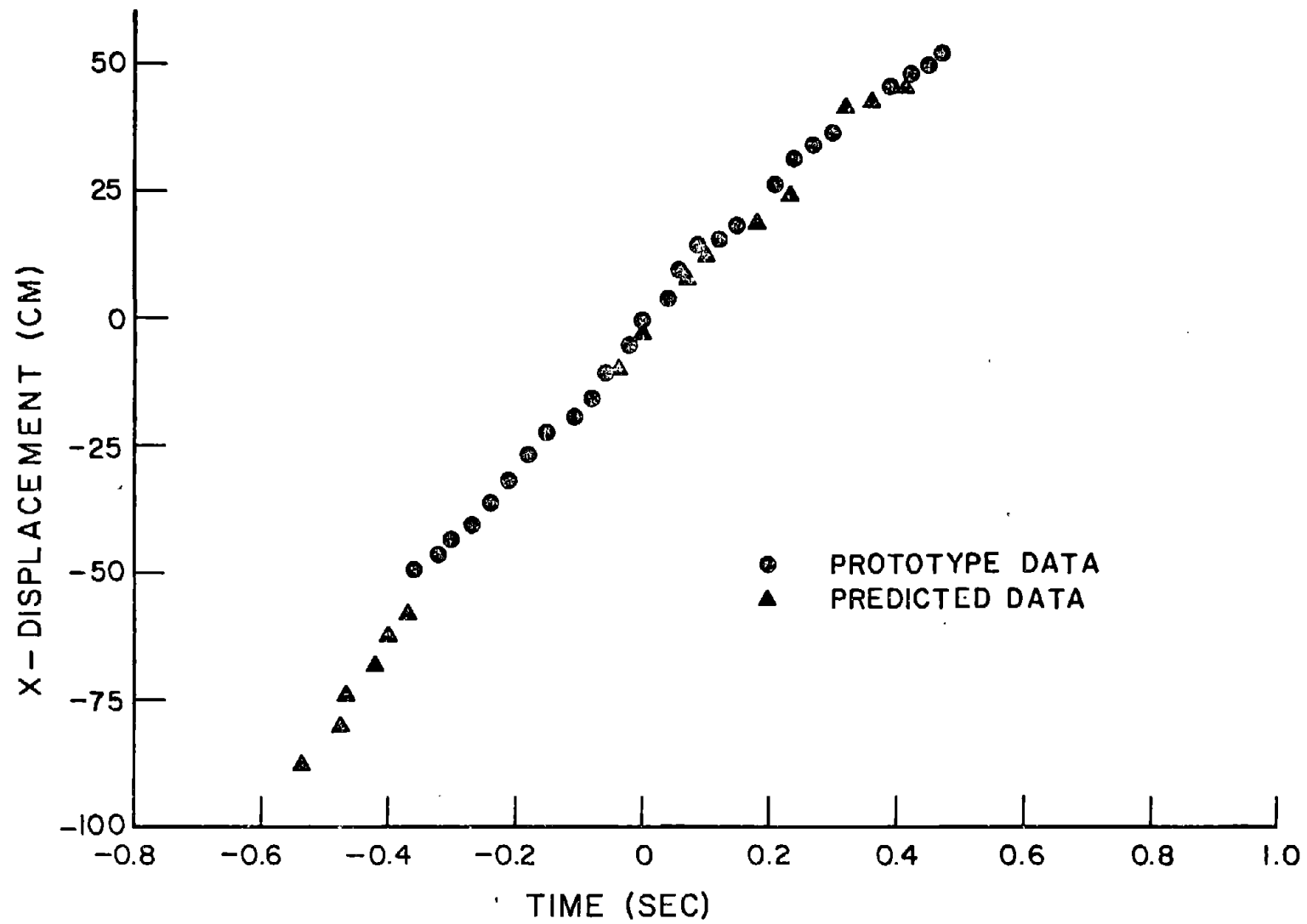


Figure 29. Results of high speed film analysis showing the displacement of left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

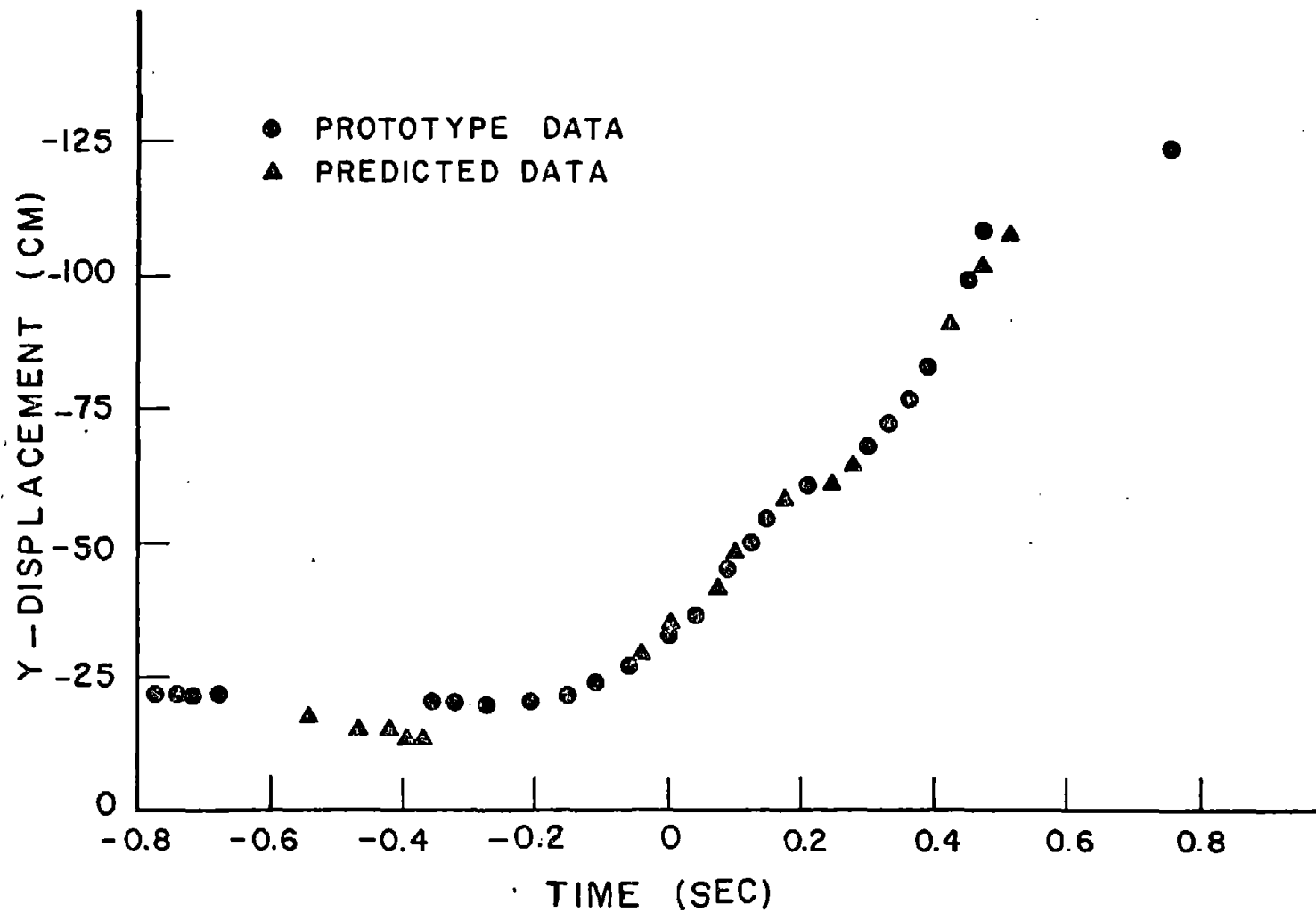


Figure 30. Results of high speed film analysis showing the displacement of left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

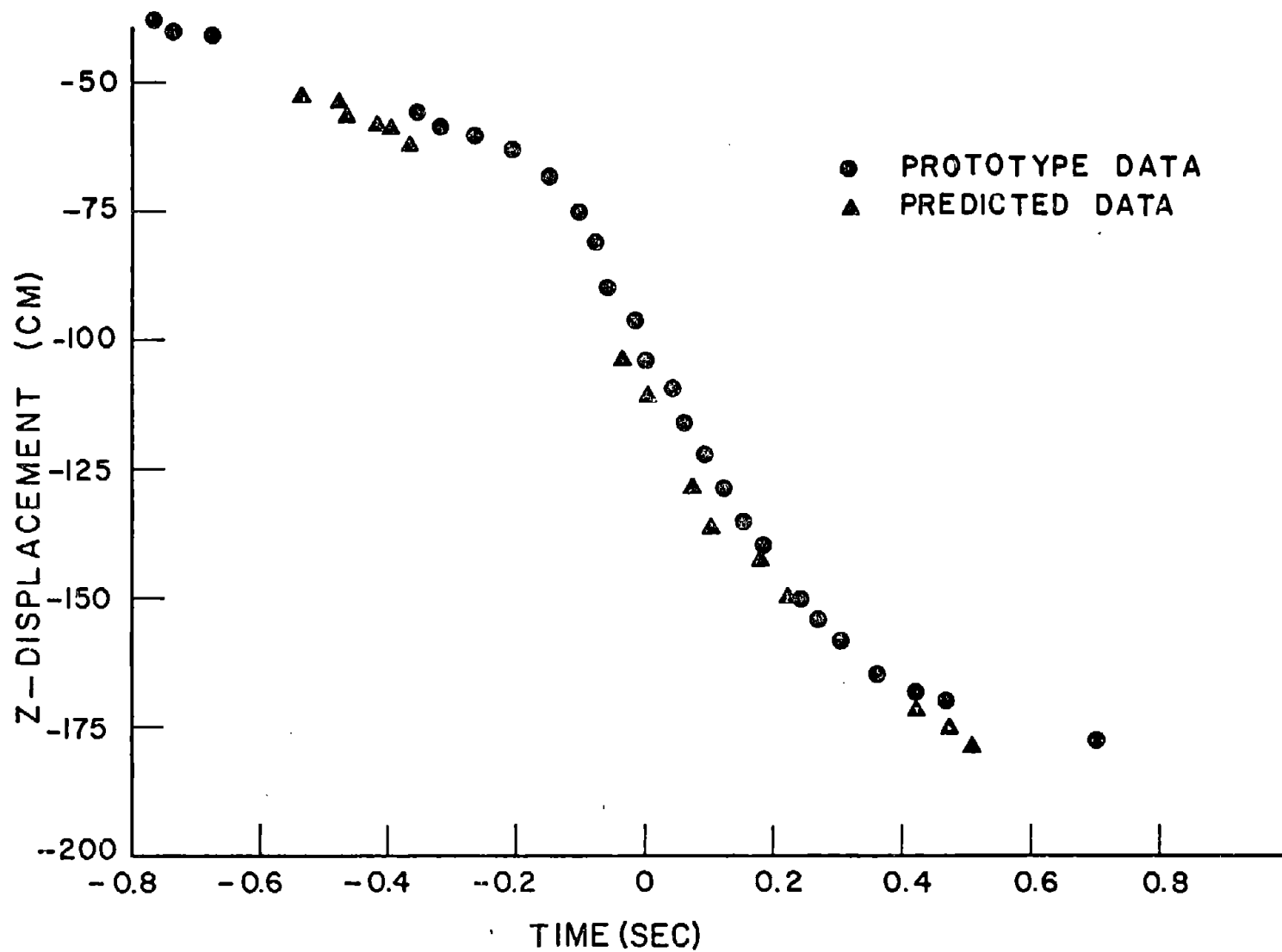
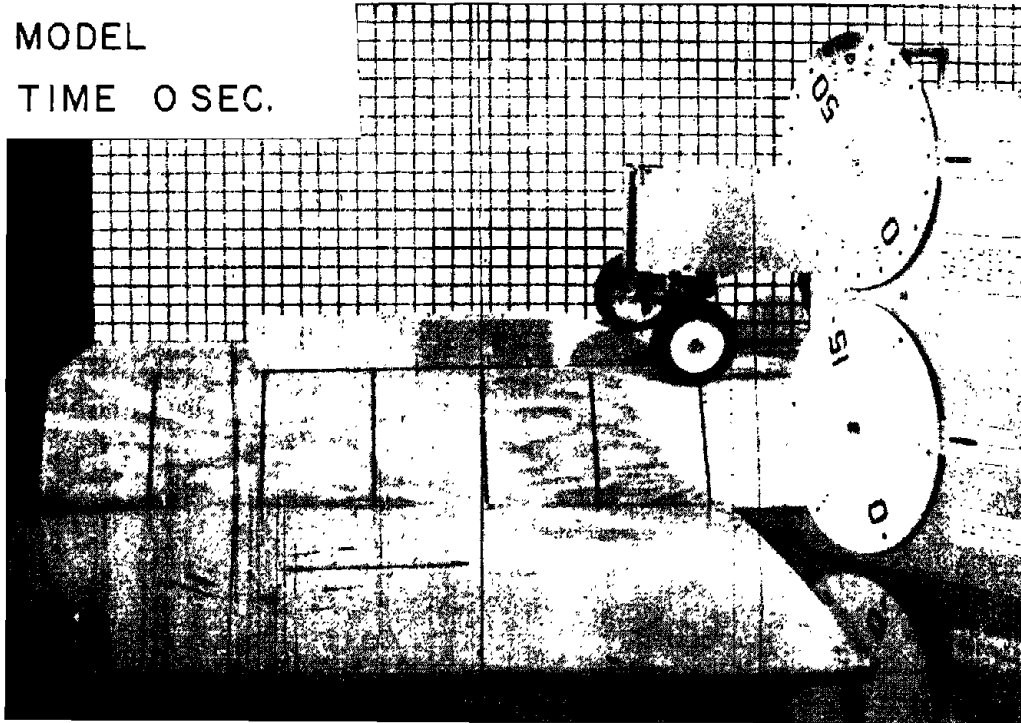


Figure 31. Results of high speed film analysis showing the displacement of the left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

MODEL
TIME 0 SEC.



PROTOTYPE
TIME 0 SEC

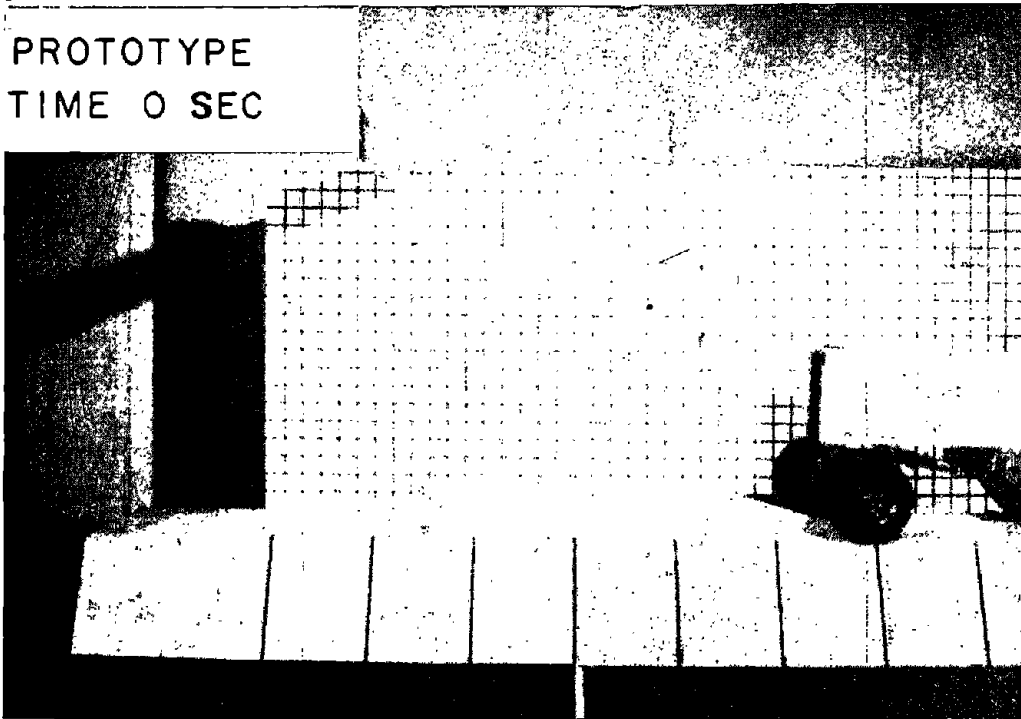
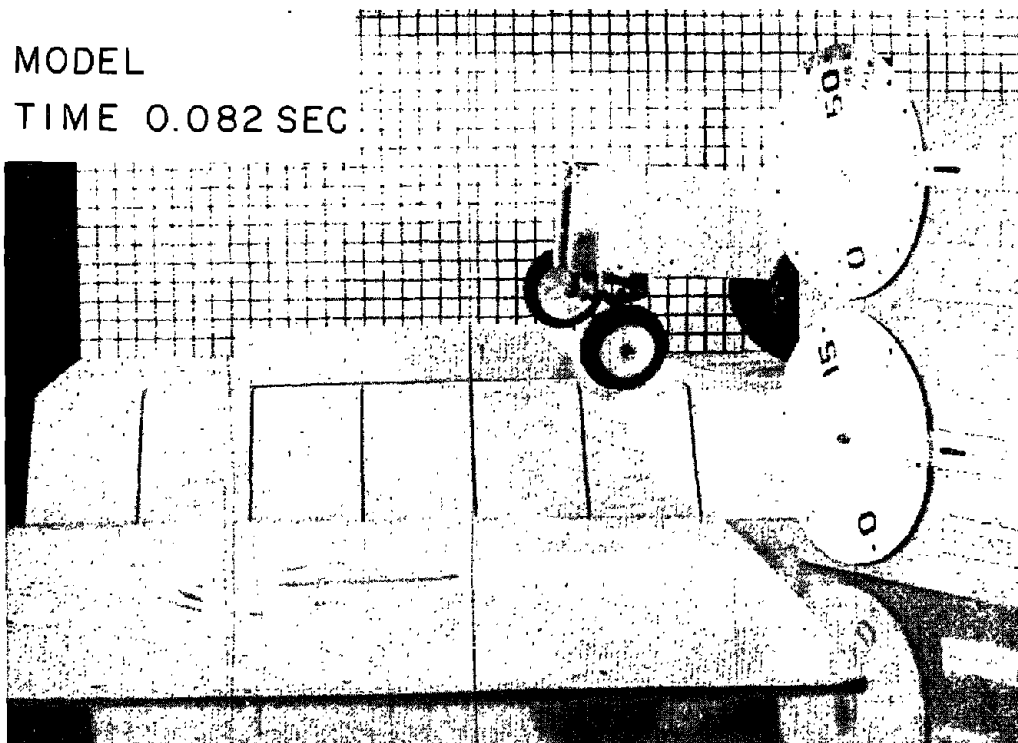


Figure 32. Sequence of photographs made from the high speed motion film to show similarity in the overturning behavior of the 1/4-scale and the 1/8-scale tractor models. The time interval of these photographs corresponds to the time scale factor, $t = \sqrt{n} \ t_m$.

MODEL
TIME 0.082 SEC.



PROTOTYPE
TIME 0.115 SEC

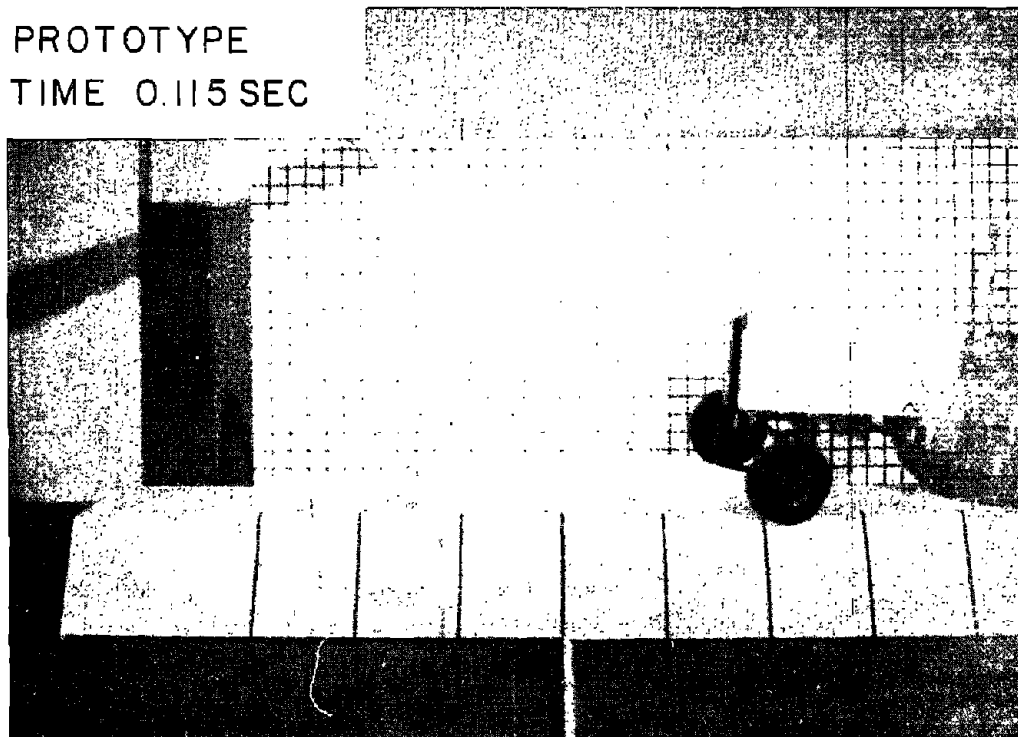
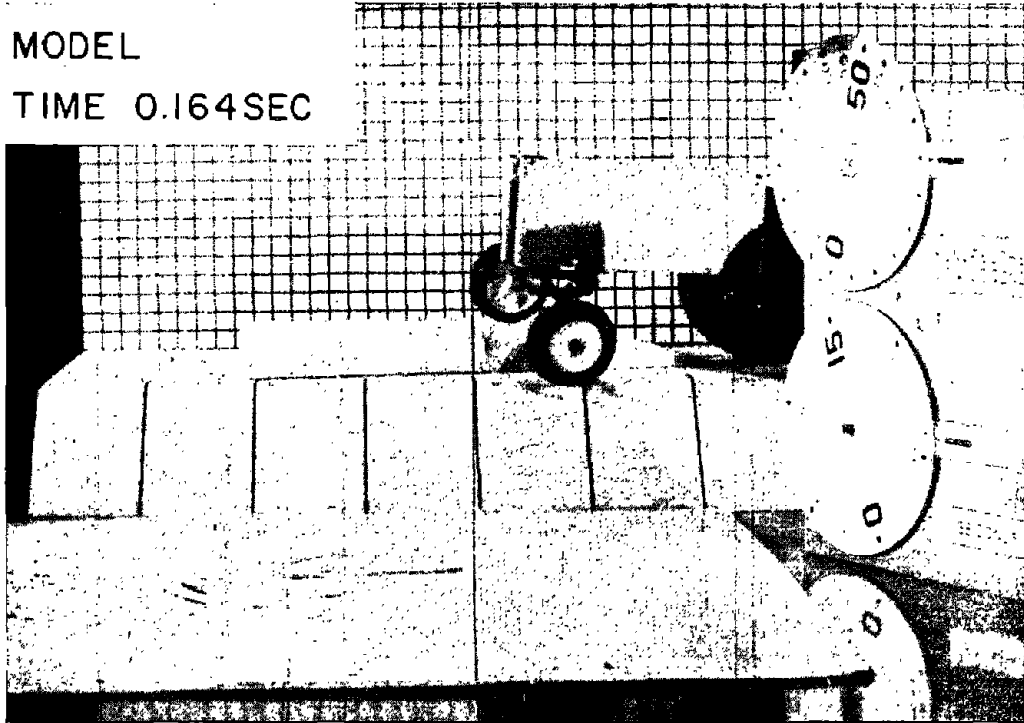


Figure 32. cont'd.

MODEL
TIME 0.164 SEC



PROTOTYPE
TIME 0.230 SEC

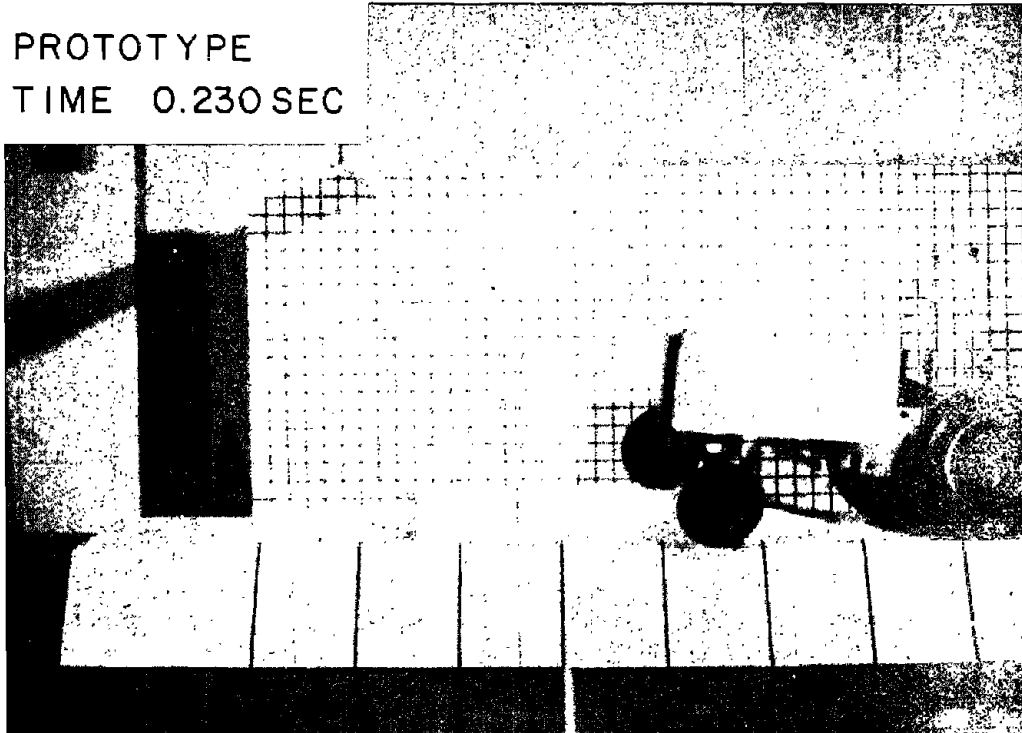
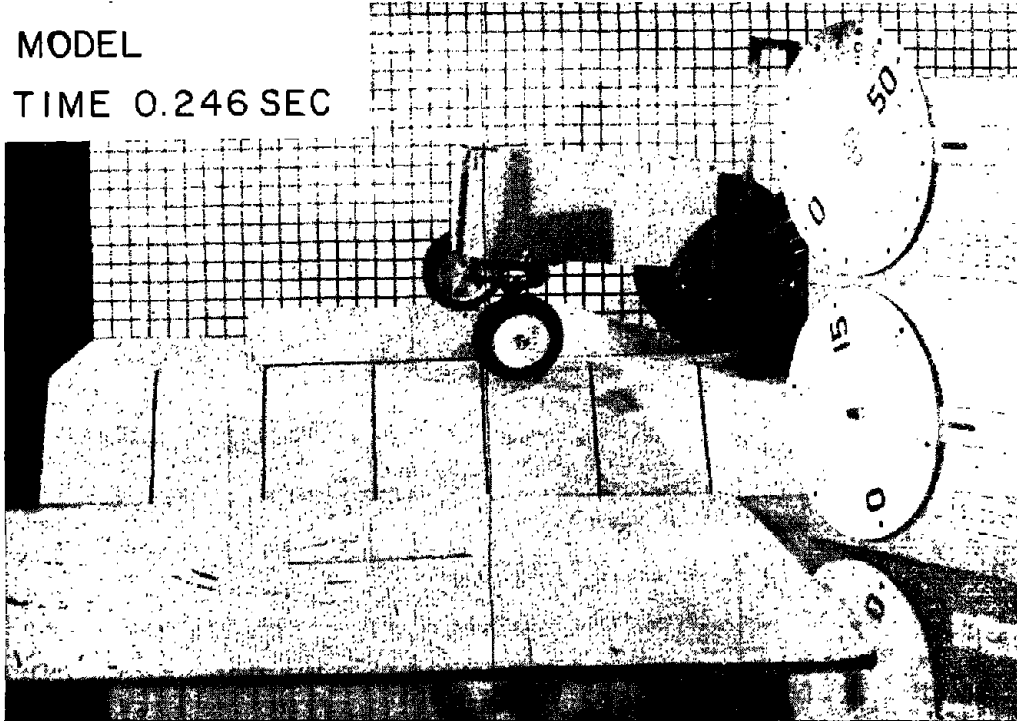


Figure 32. cont'd.

MODEL
TIME 0.246 SEC



PROTOTYPE
TIME 0.344 SEC

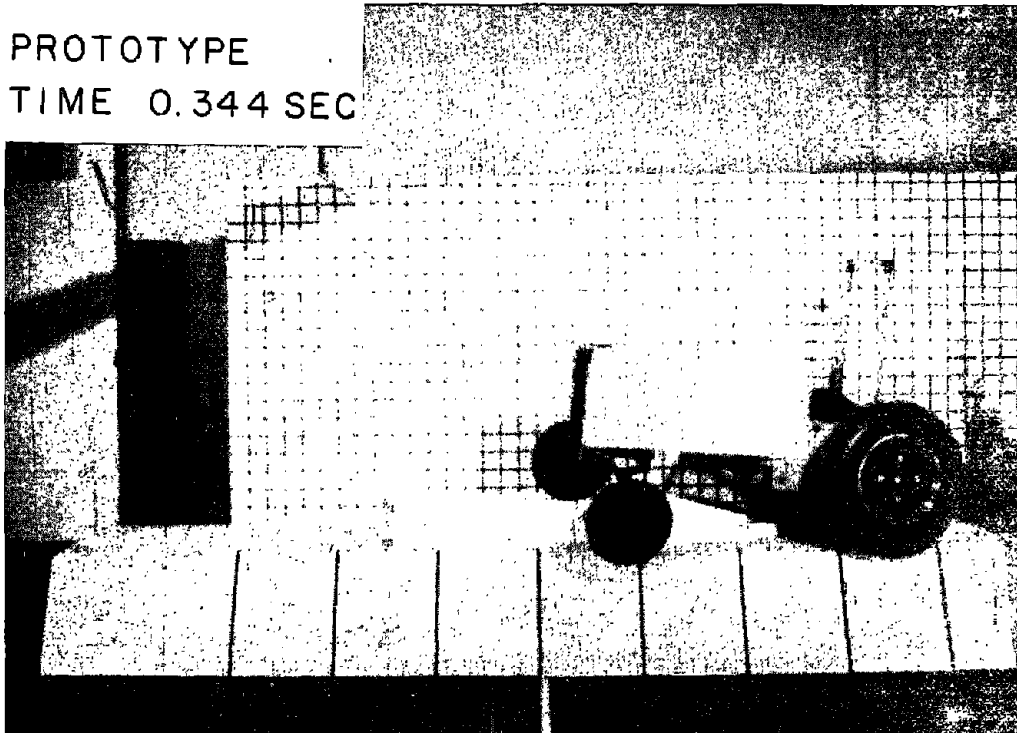
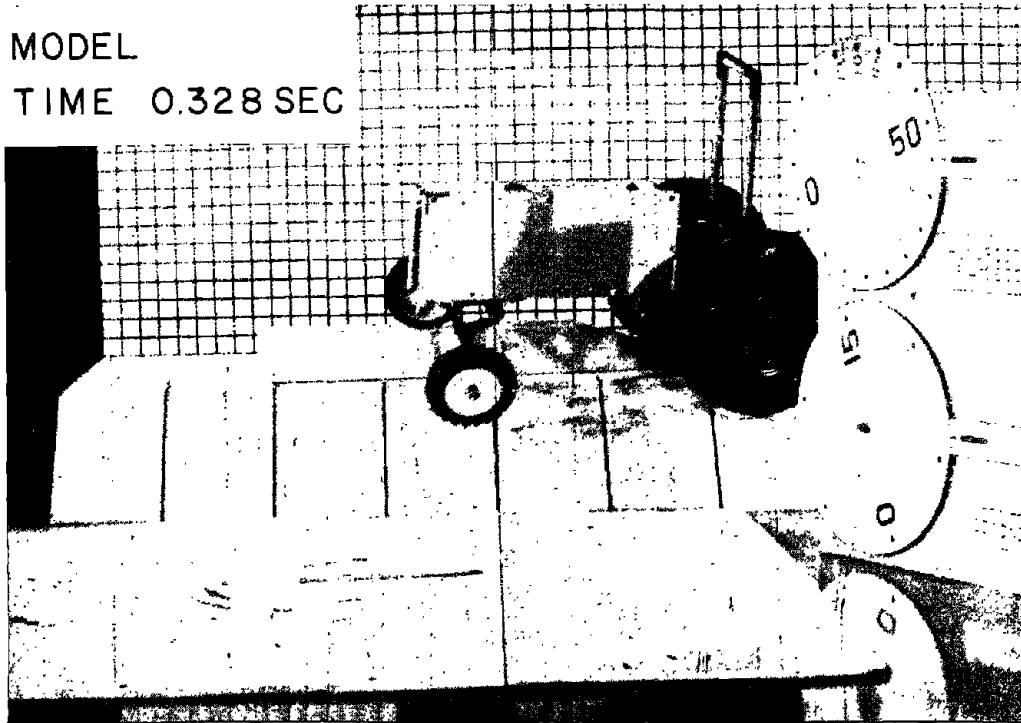


Figure 32. cont'd.

MODEL
TIME 0.328 SEC



PROTOTYPE
TIME 0.459 SEC

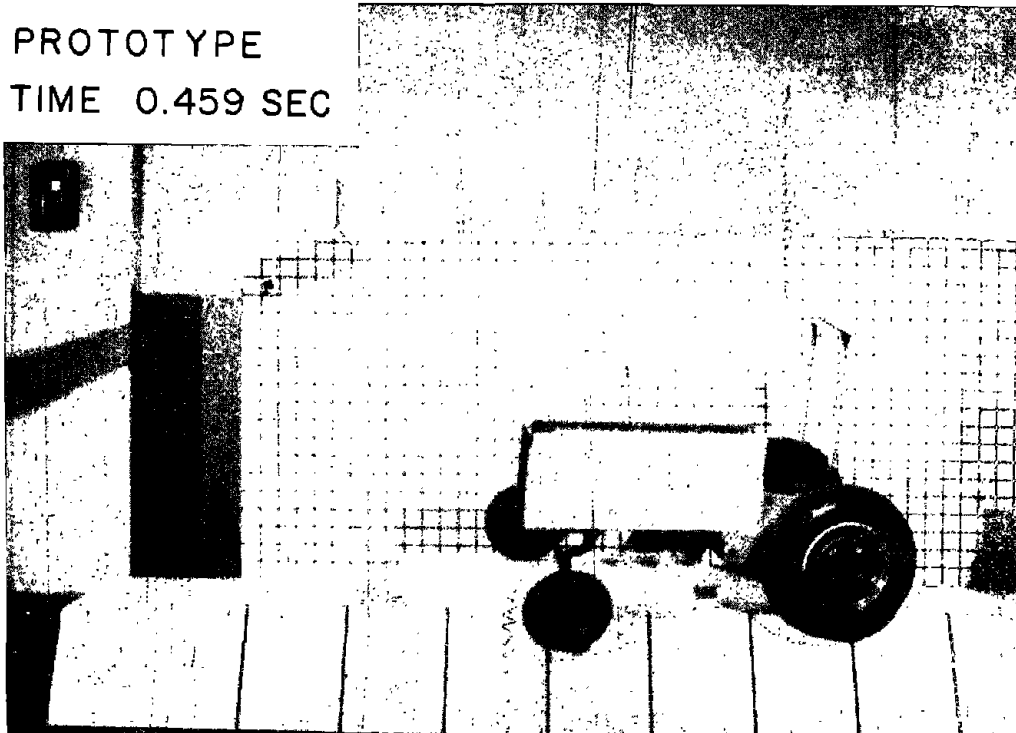
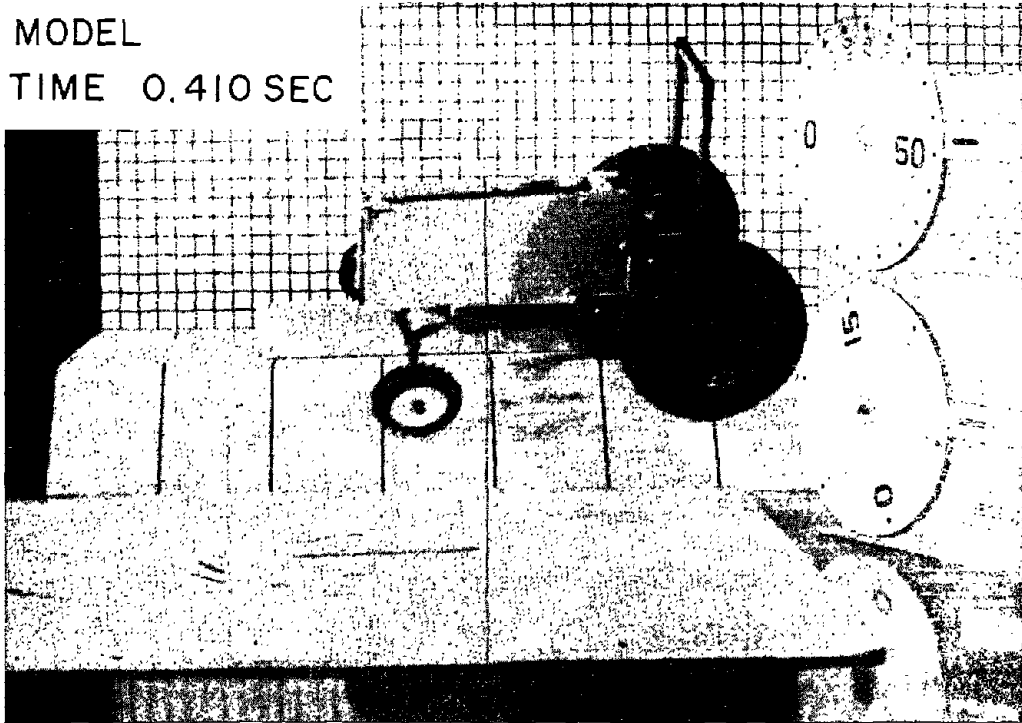


Figure 32. cont'd.

MODEL
TIME 0.410 SEC



PROTOTYPE
TIME 0.574 SEC

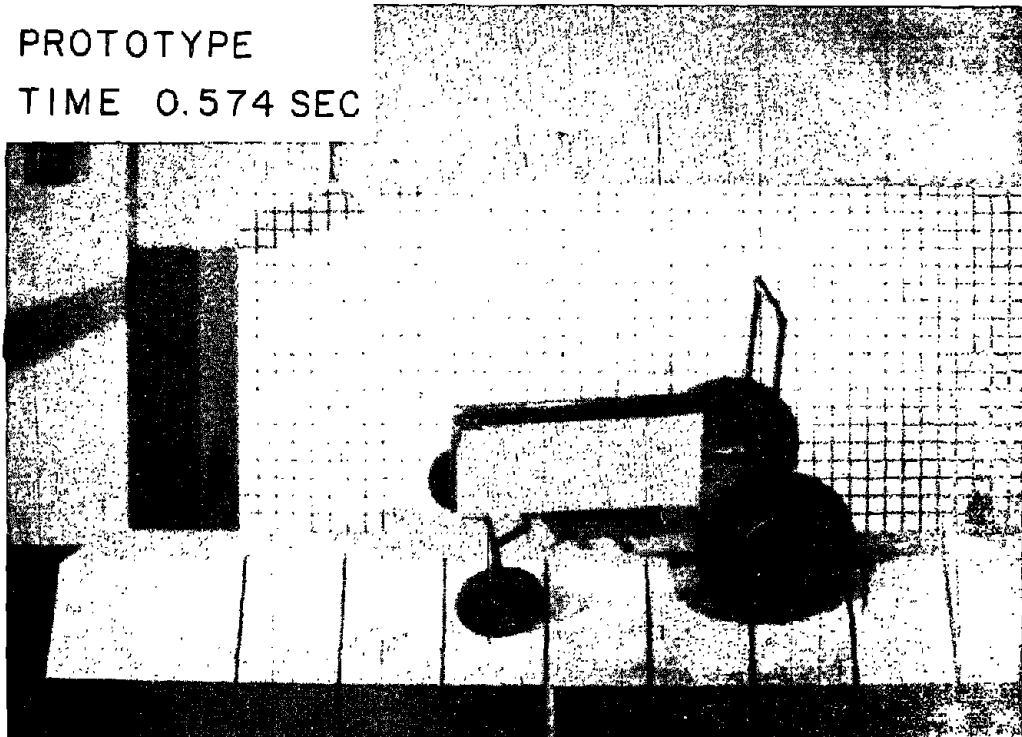
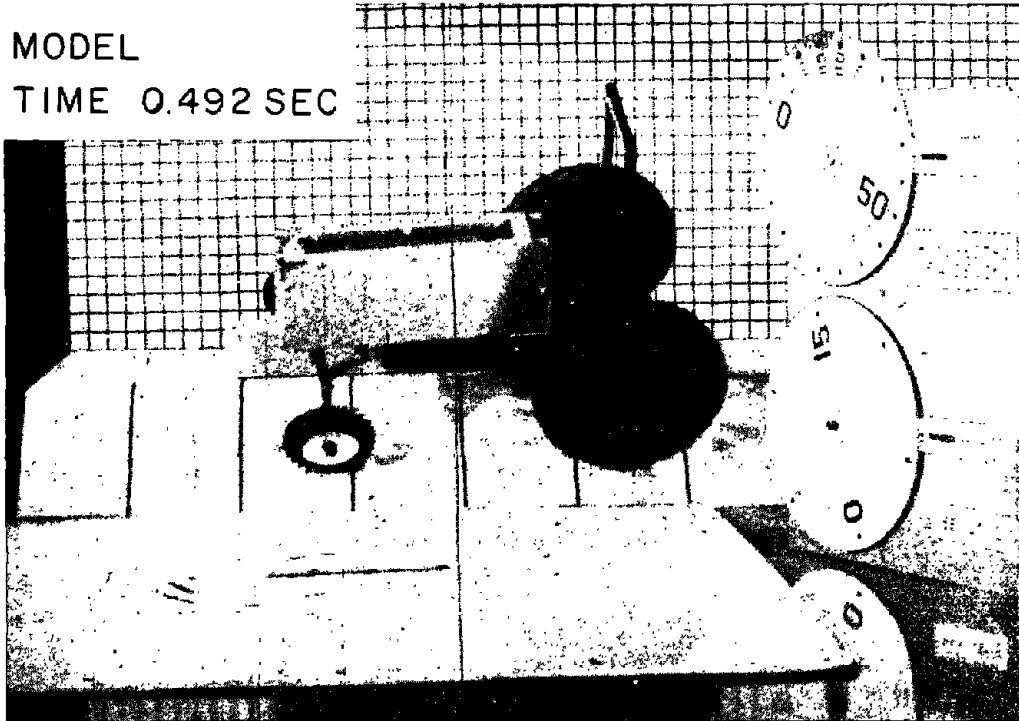


Figure 32. cont'd.

MODEL
TIME 0.492 SEC



PROTOTYPE
TIME 0.689 SEC

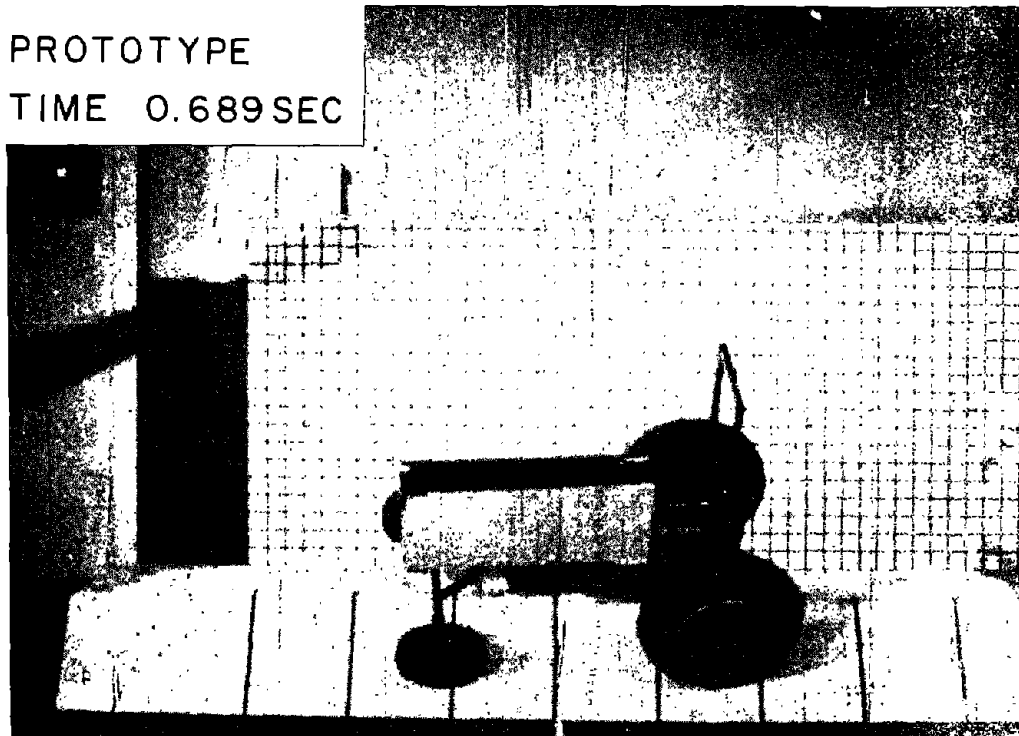
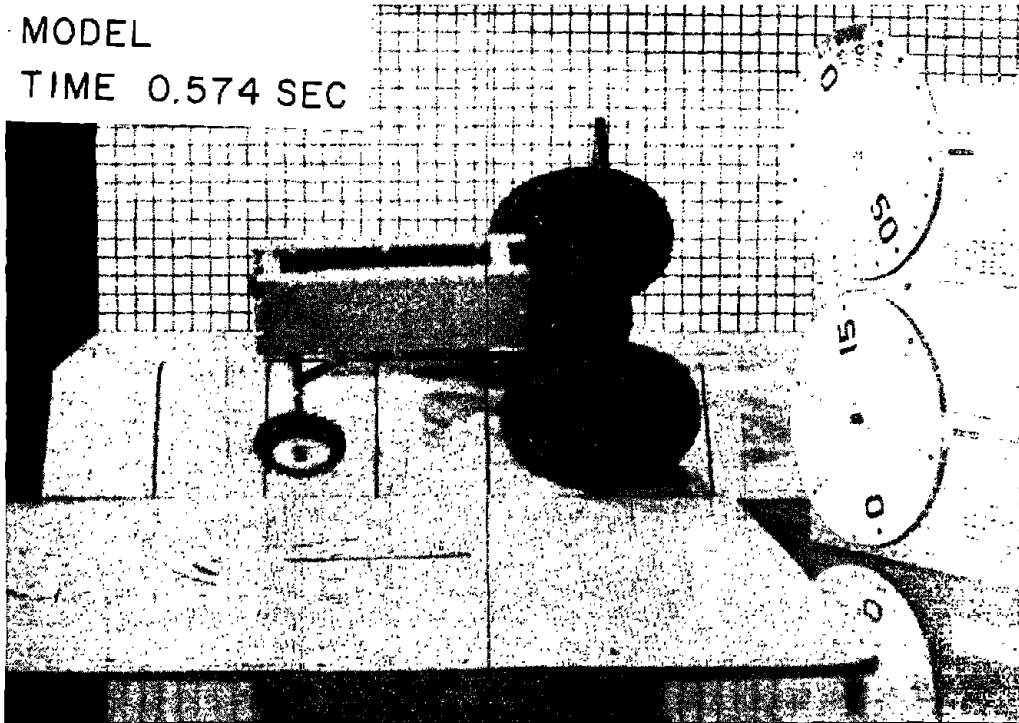


Figure 32. cont'd.

MODEL
TIME 0.574 SEC



PROTOTYPE
TIME 0.803 SEC

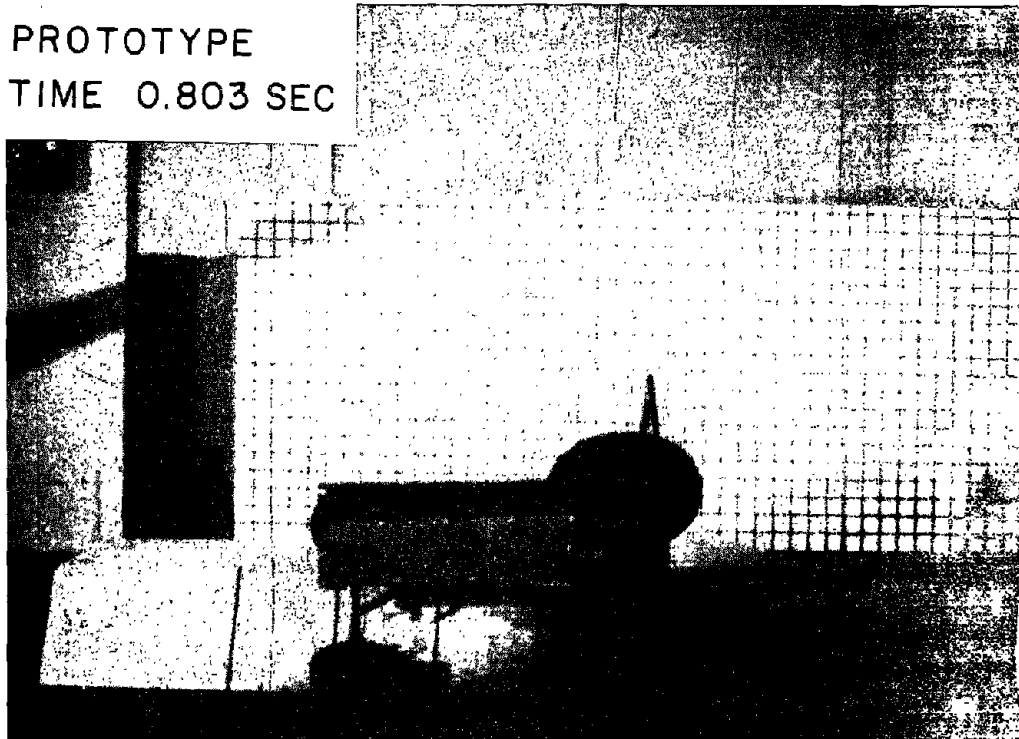
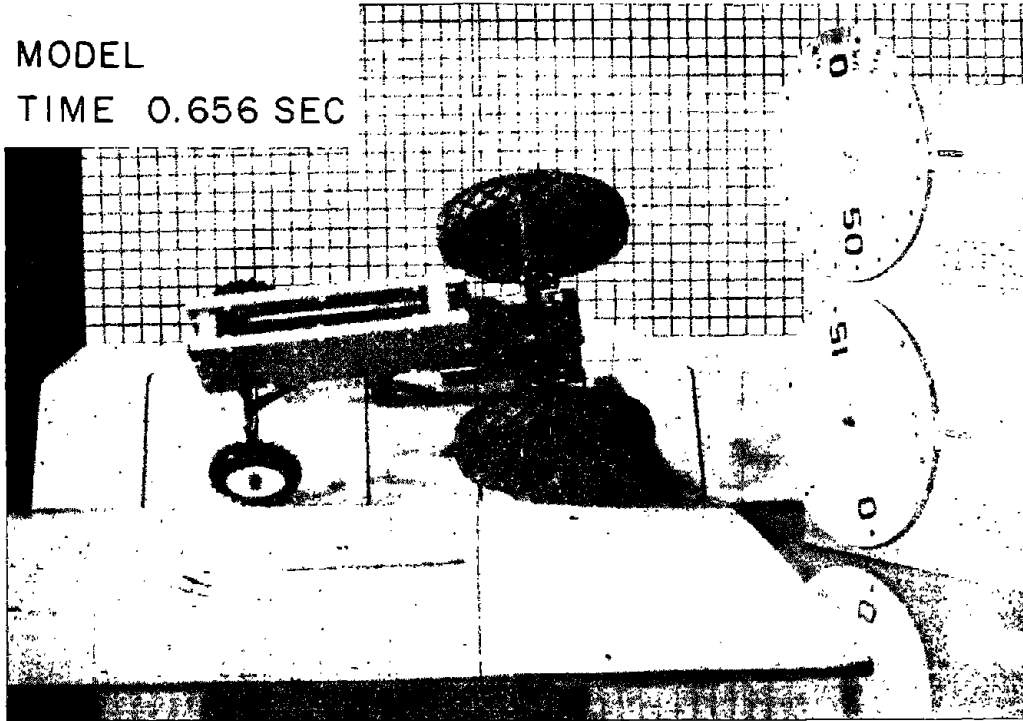


Figure 32. cont'd.

MODEL
TIME 0.656 SEC



PROTOTYPE
TIME 0.918 SEC

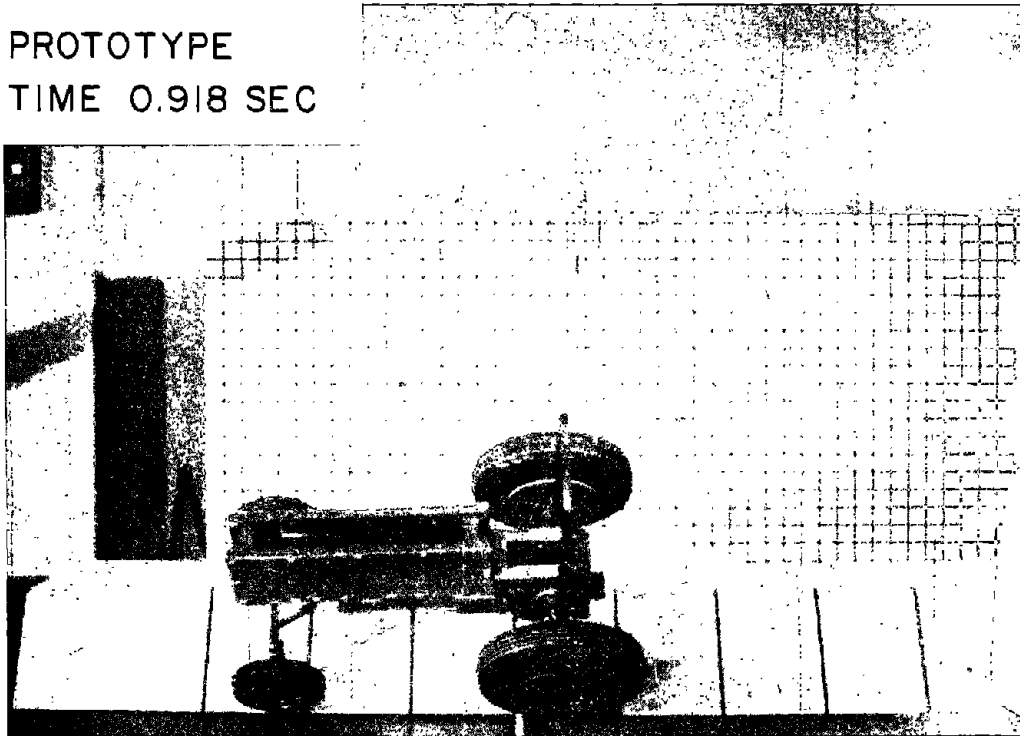
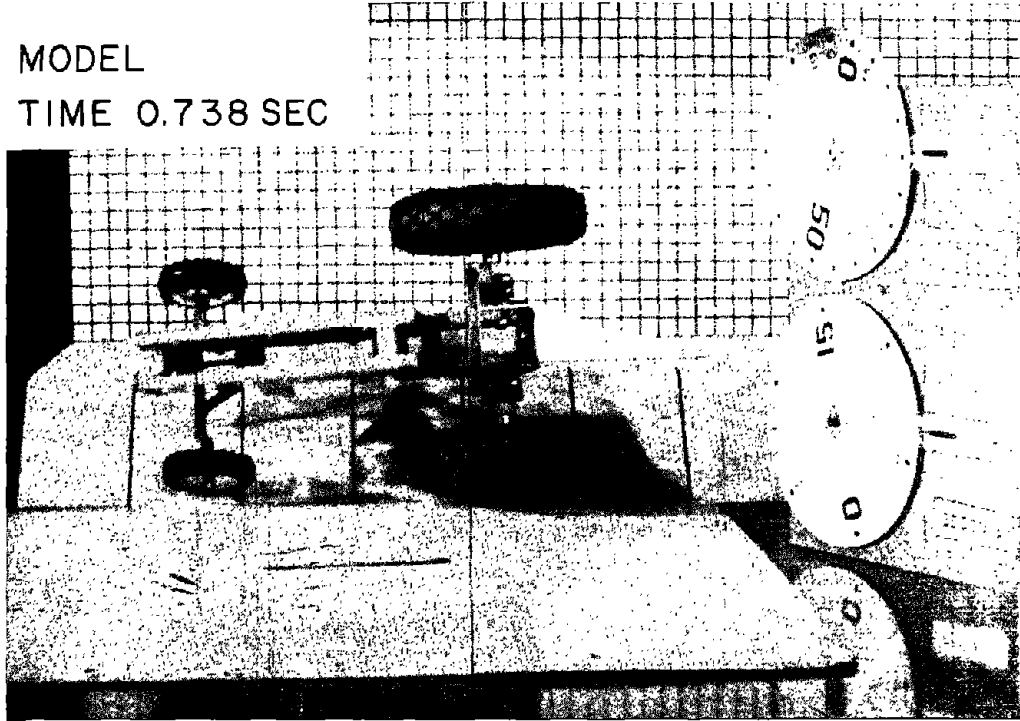


Figure 32. cont'd.

MODEL
TIME 0.738 SEC



PROTOTYPE
TIME 1.033 SEC

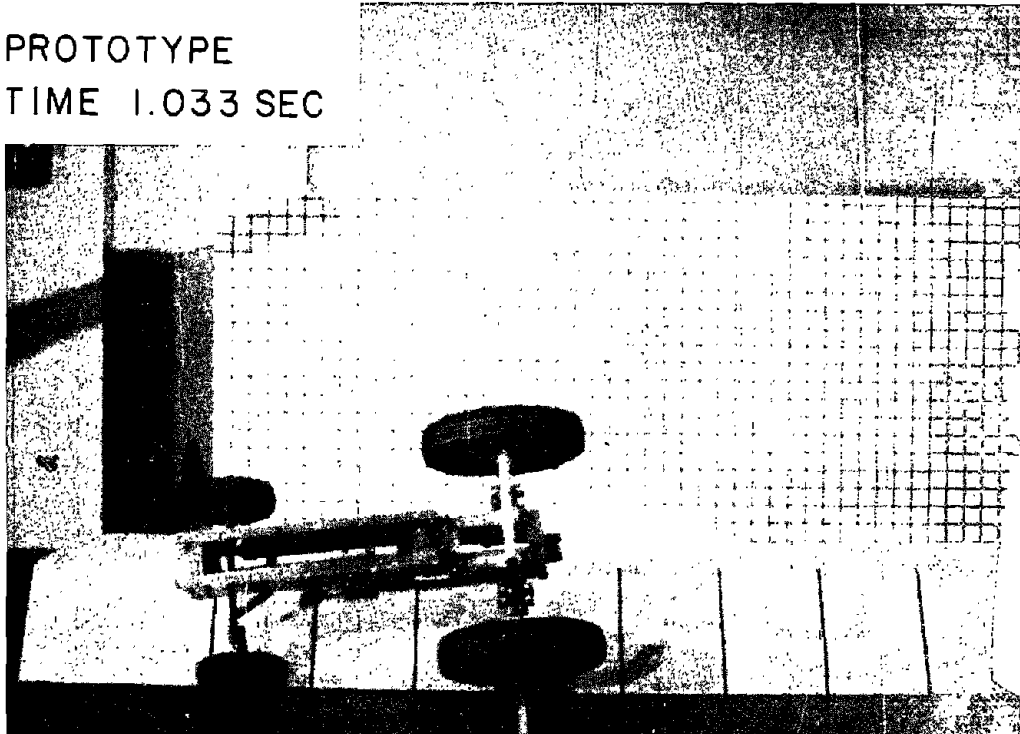
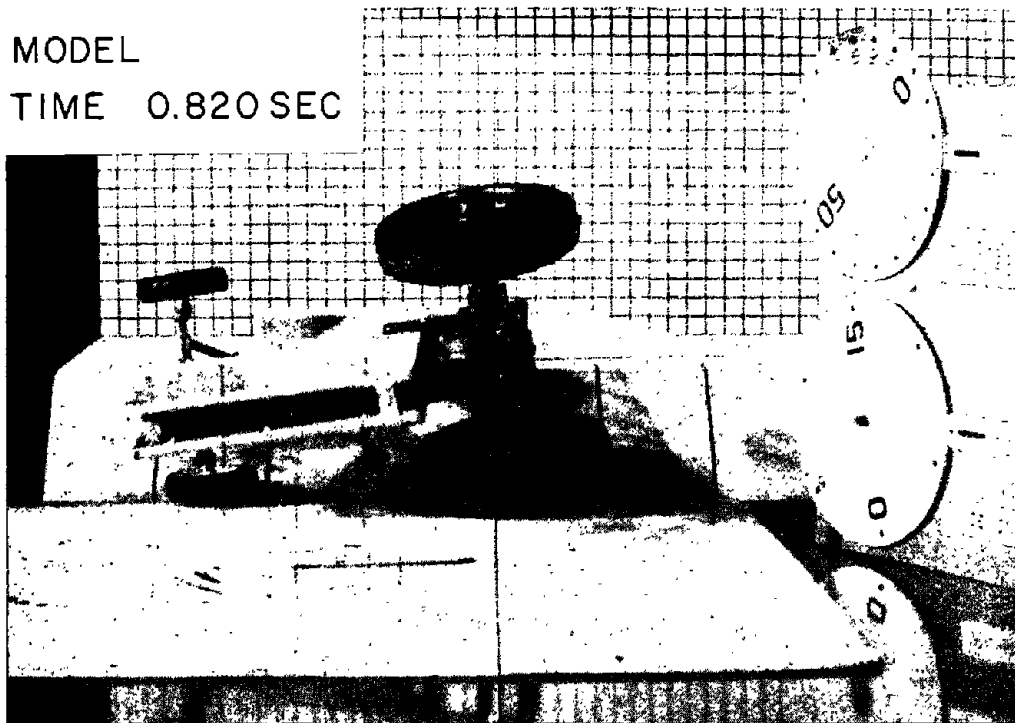


Figure 32. cont'd.

MODEL
TIME 0.820 SEC



PROTOTYPE
TIME 1.148 SEC

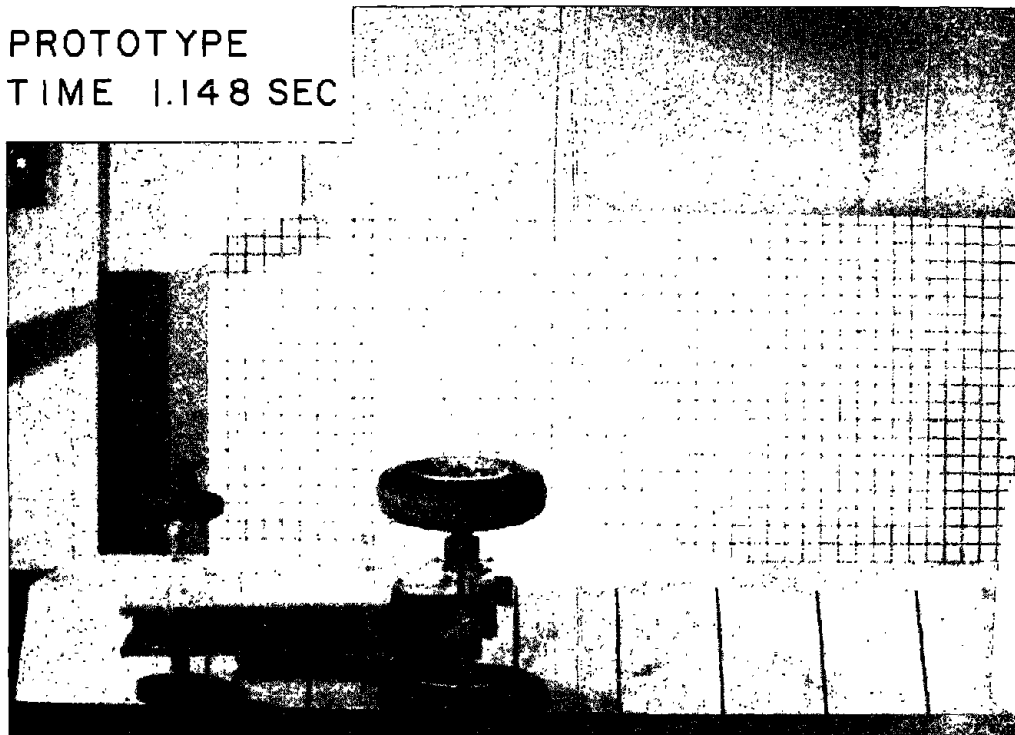
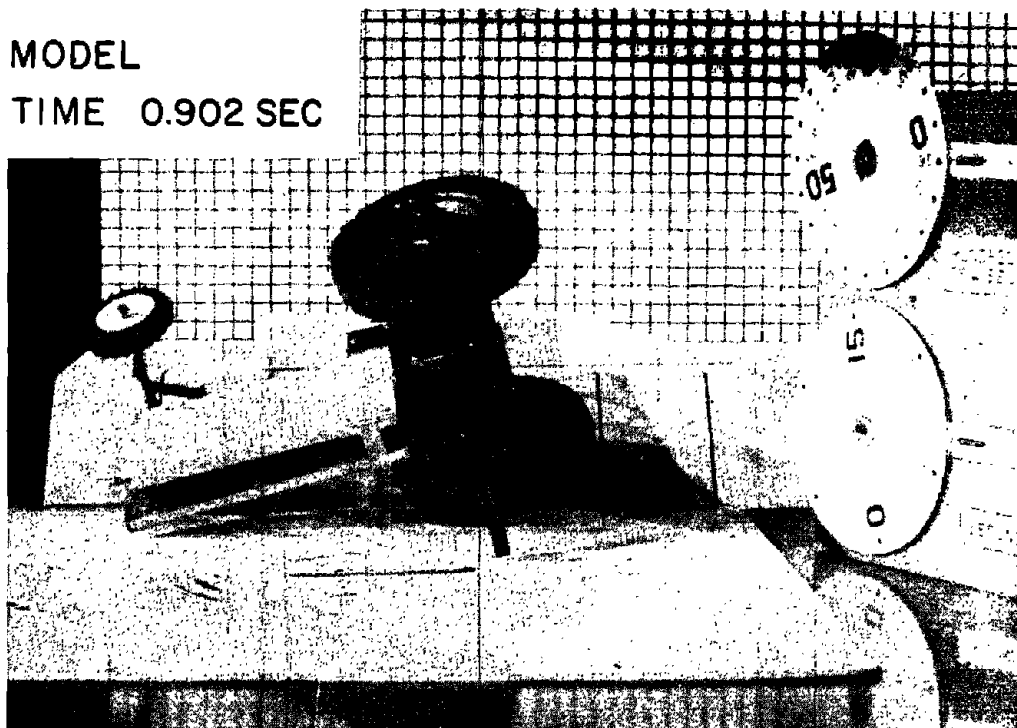


Figure 32. cont'd.

MODEL
TIME 0.902 SEC



PROTOTYPE
TIME 1.262 SEC

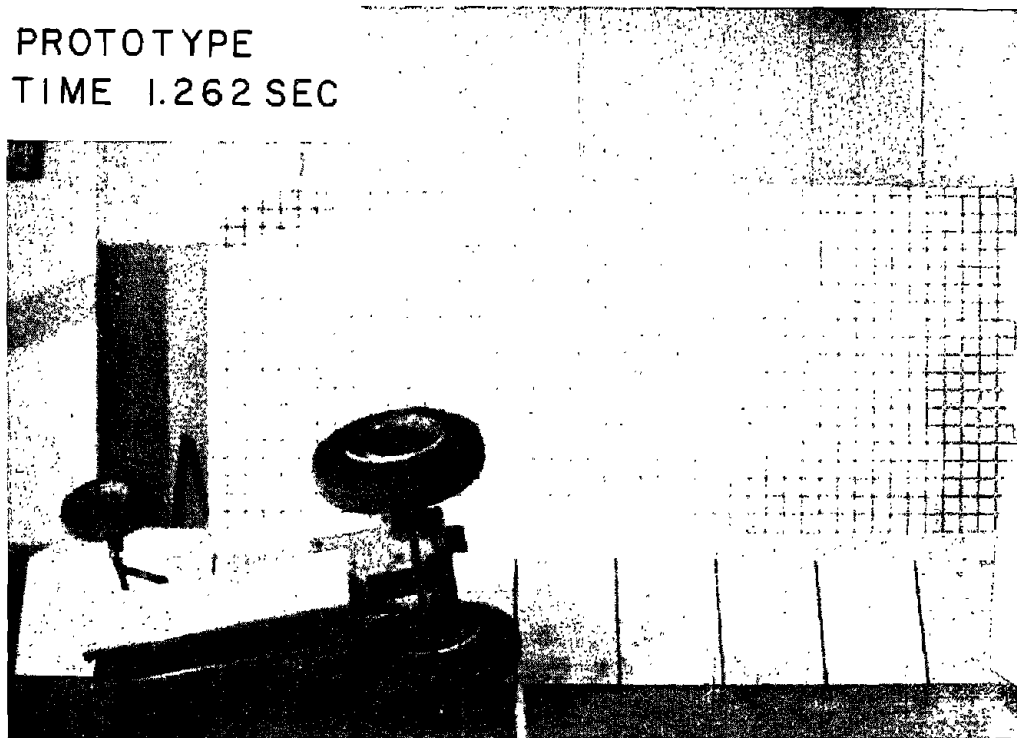


Figure 32. cont'd.

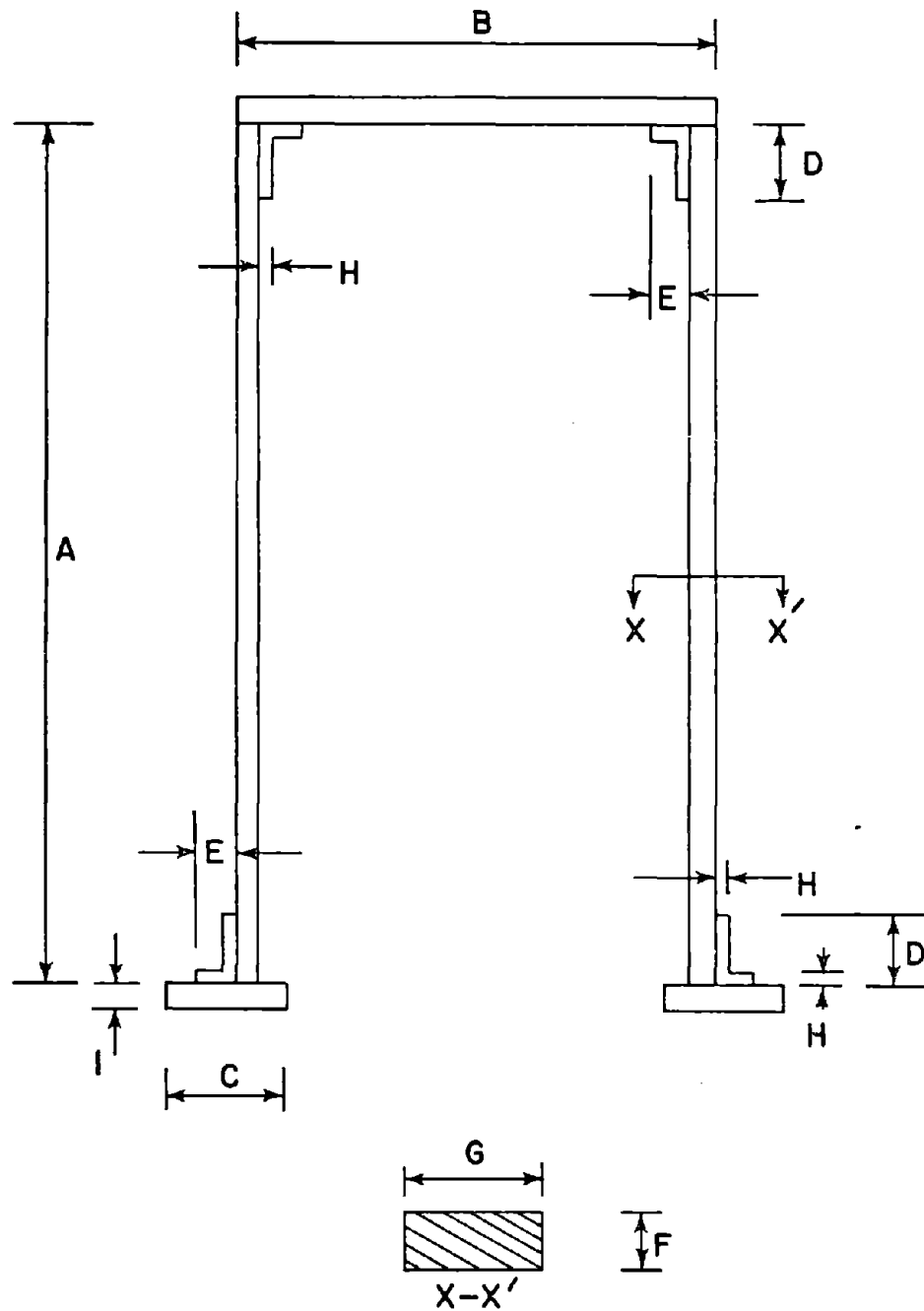


Figure 33. Generalized diagram of two-post type frame used in the model studies.

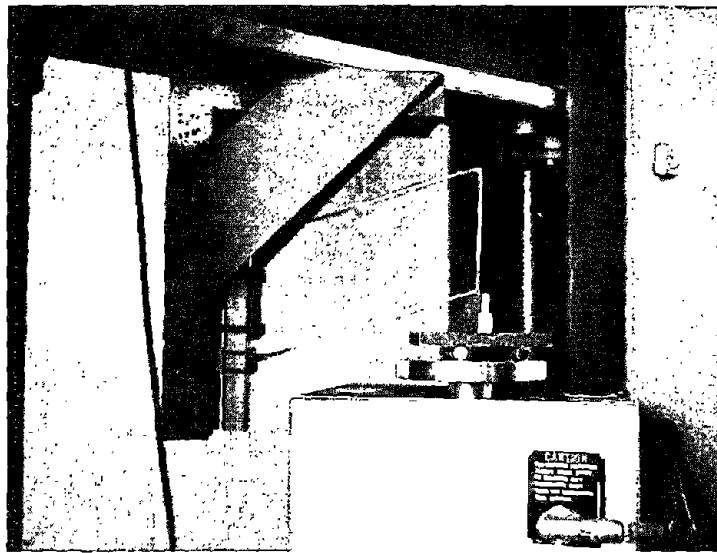


Figure 34. A two-post frame being tested in the Instron testing machine.

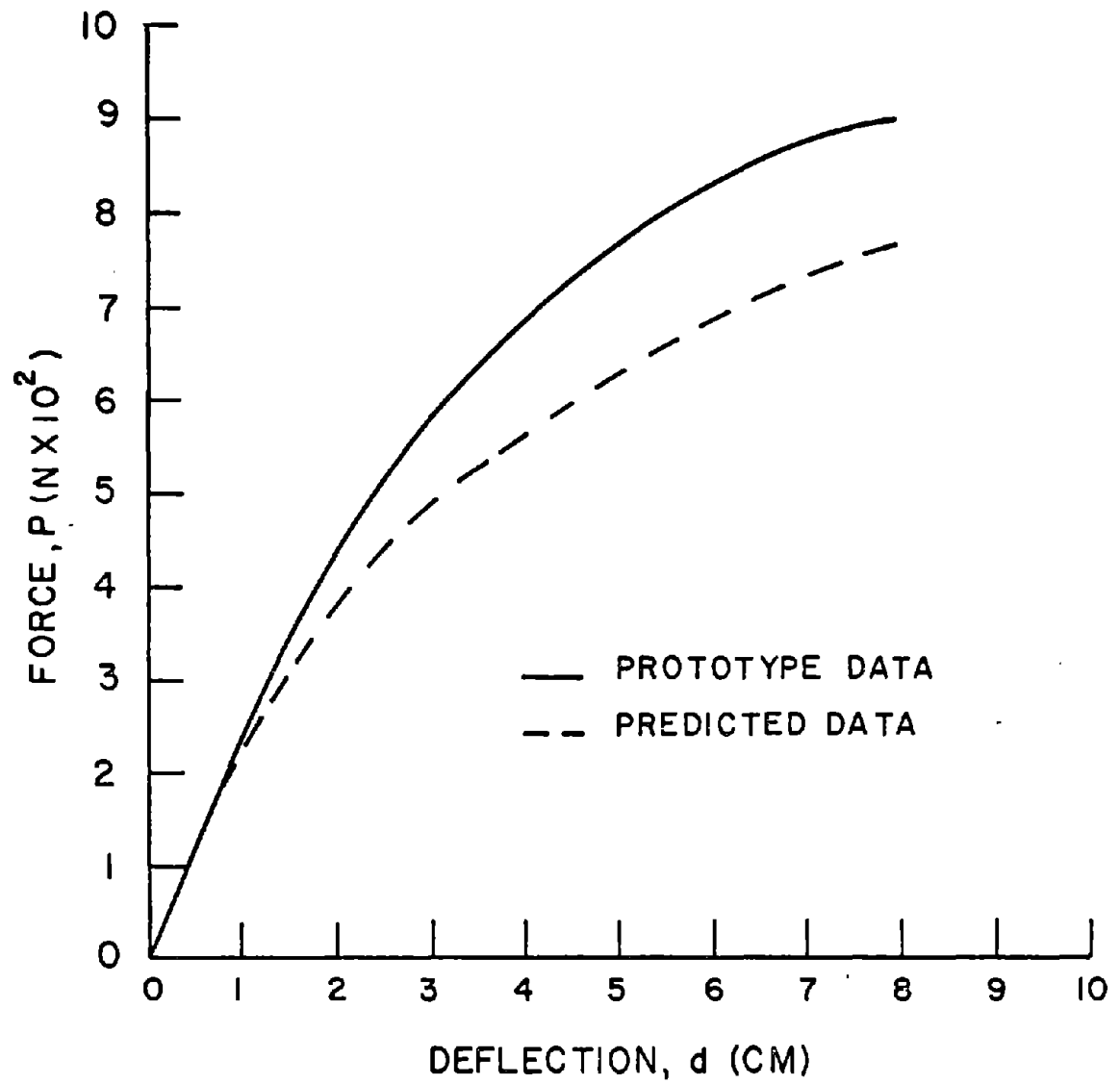


Figure 35. Static test results of the two-post type bolted frame.

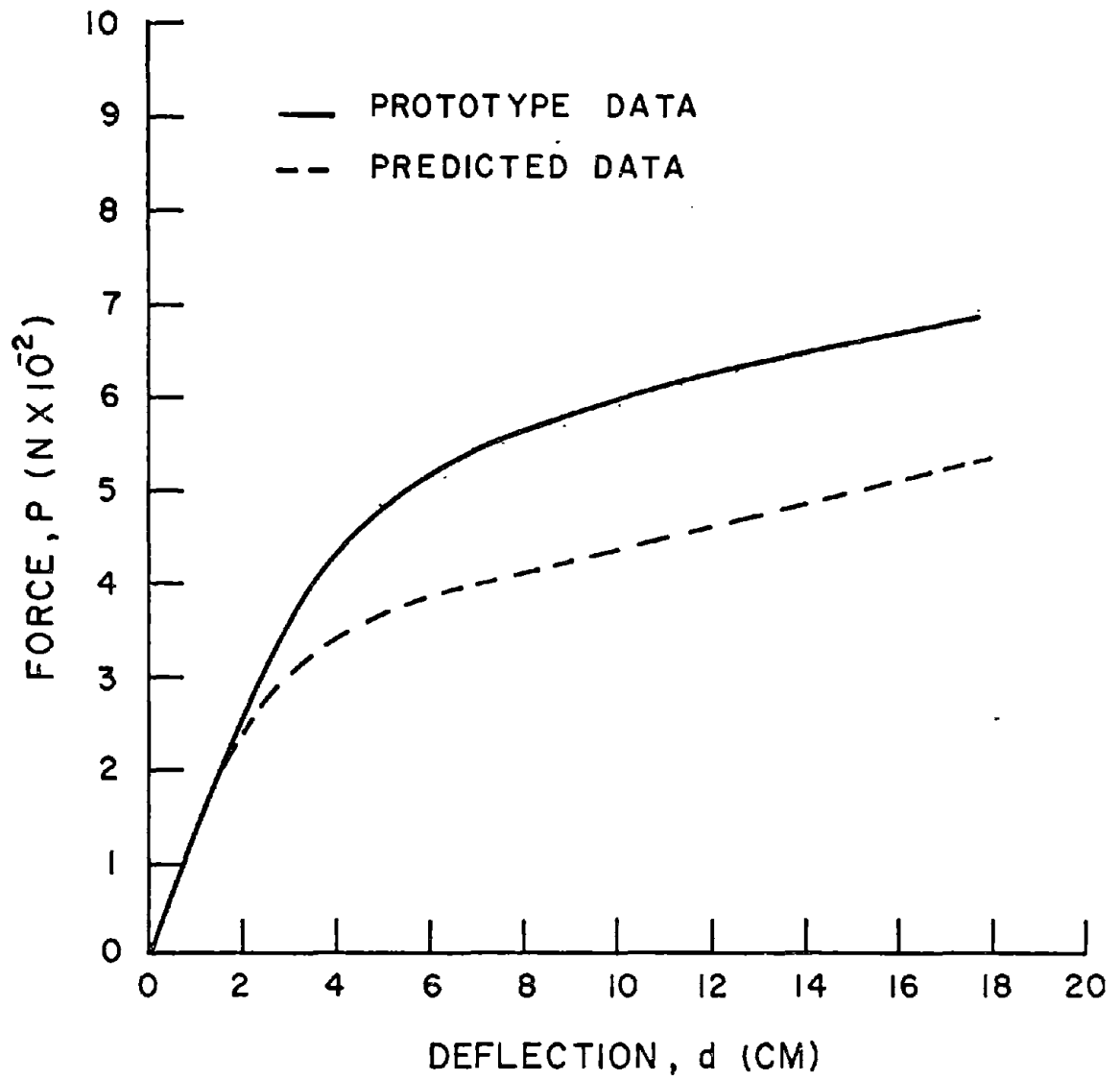


Figure 36. Static test results of the two-post type welded frames.

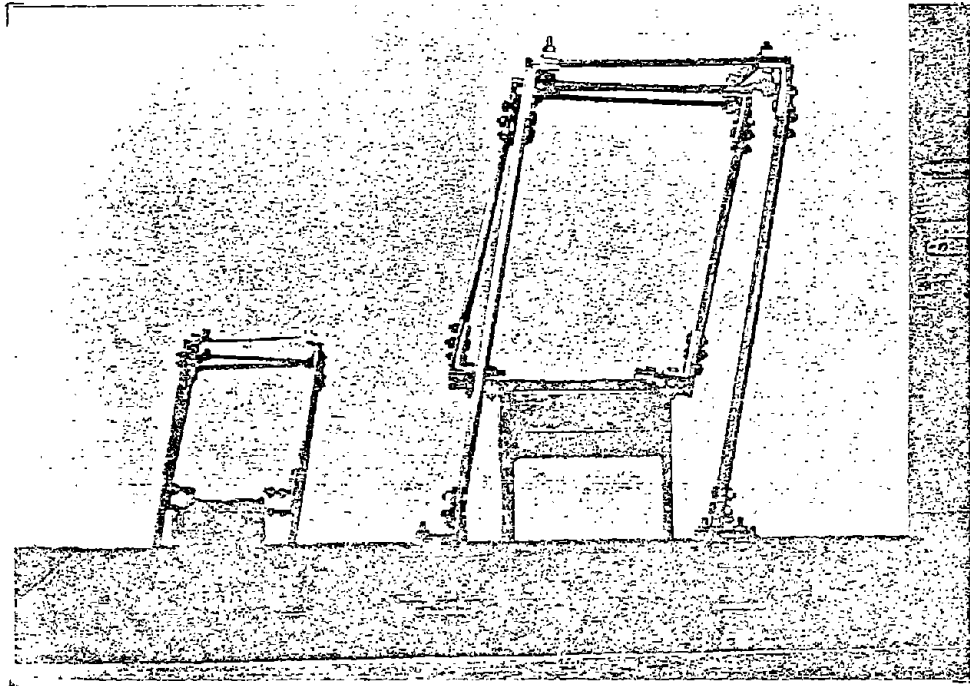


Figure 37. Typically bent 1/4-scale and 1/8-scale four-post, type model ROPS after static testing.

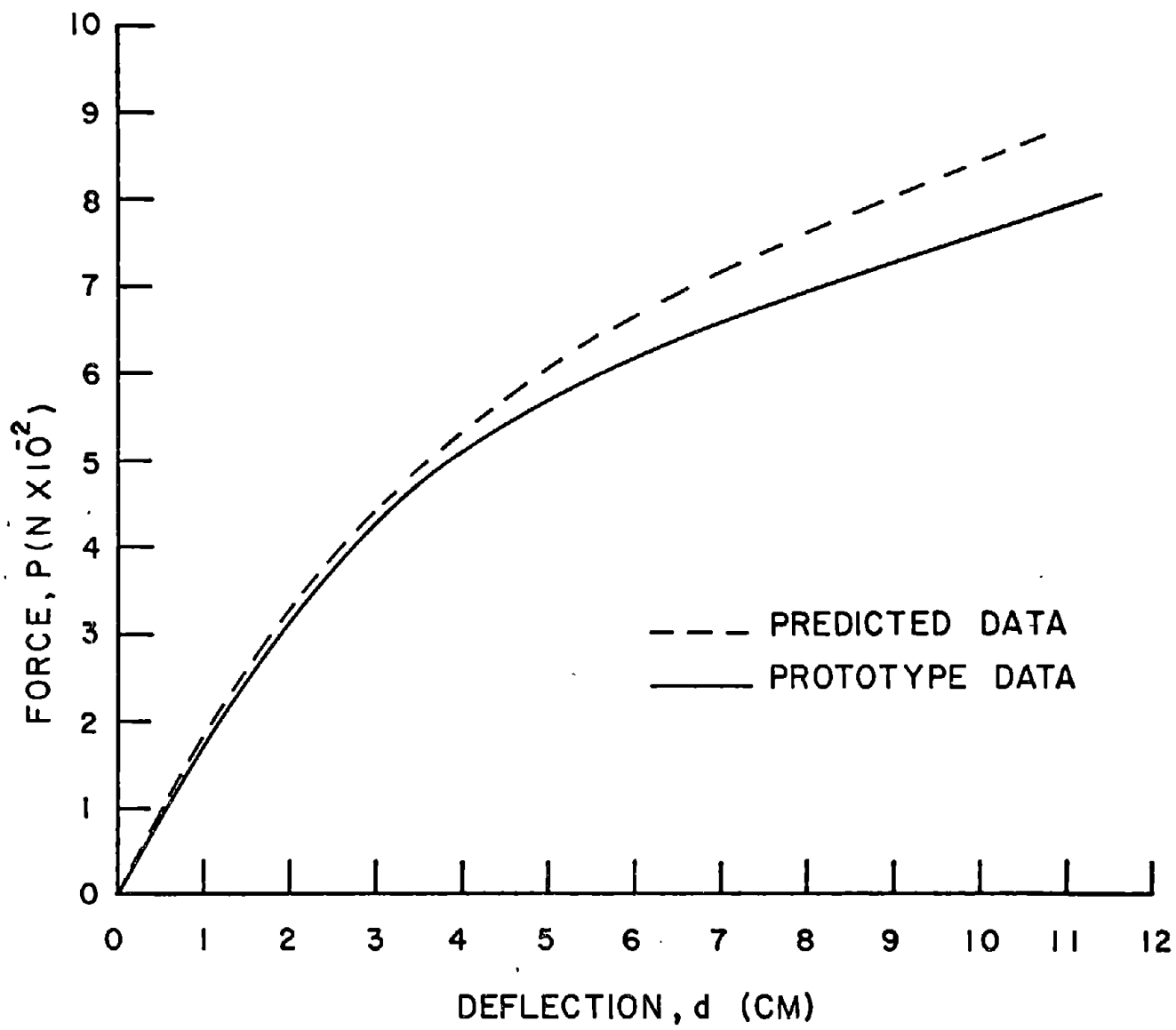


Figure 38. Static test results of four-post type model ROPS.

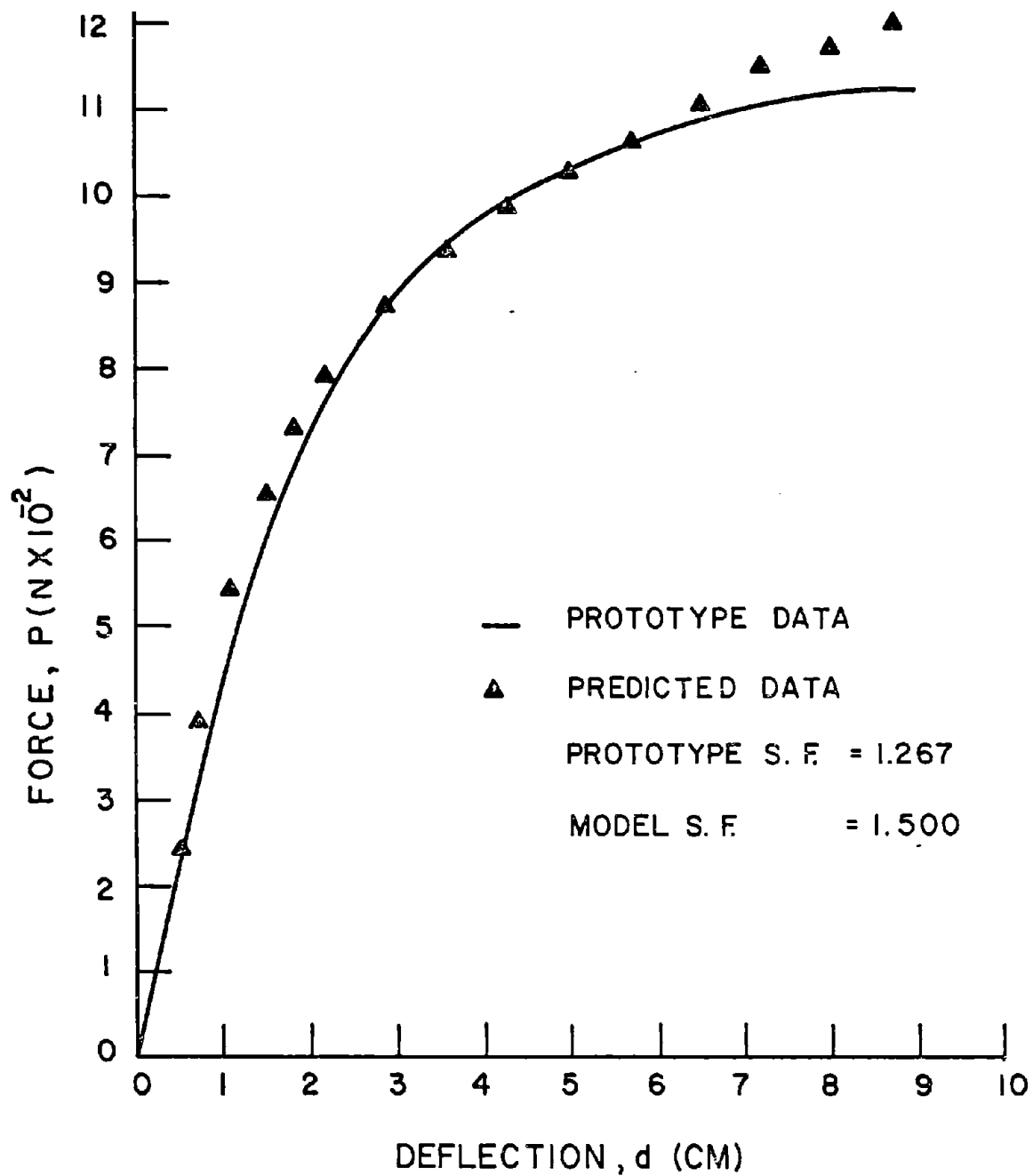


Figure 39. Static test results of dissimilar cross-section model frames.

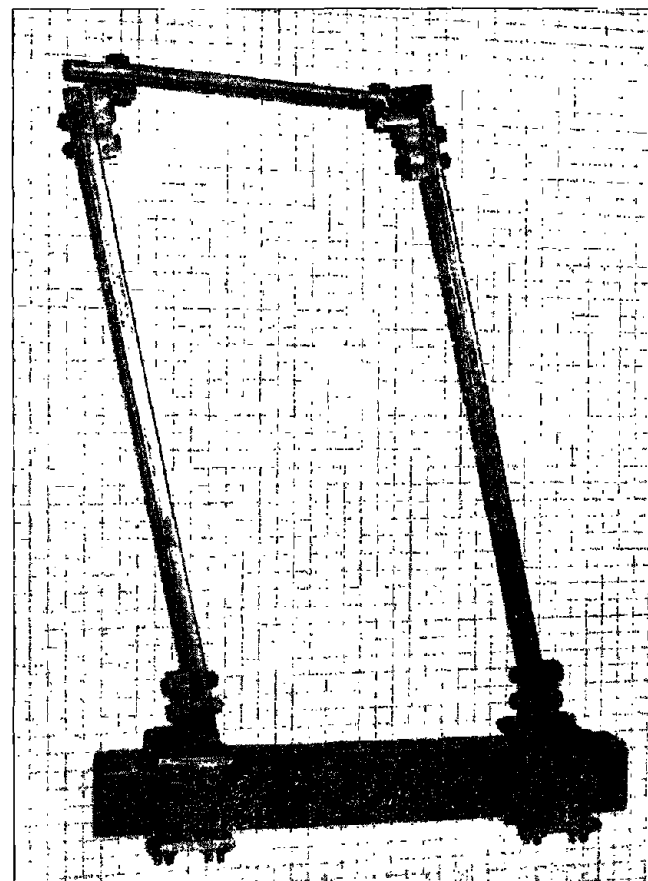
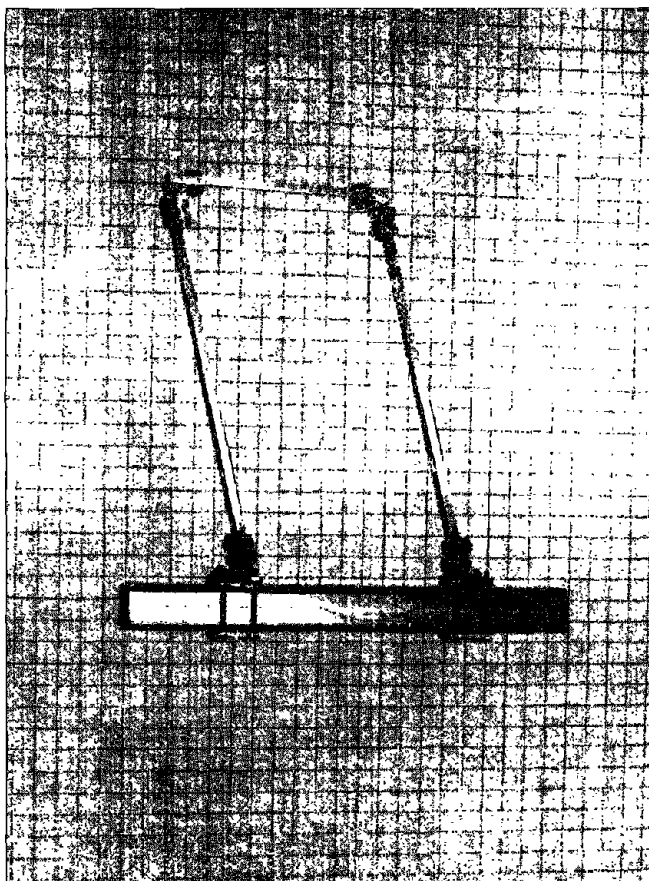


Figure 40. Photograph showing the similarity in deformation between the solid-section model '(a)' and the hollow-section prototype '(b)' frames.

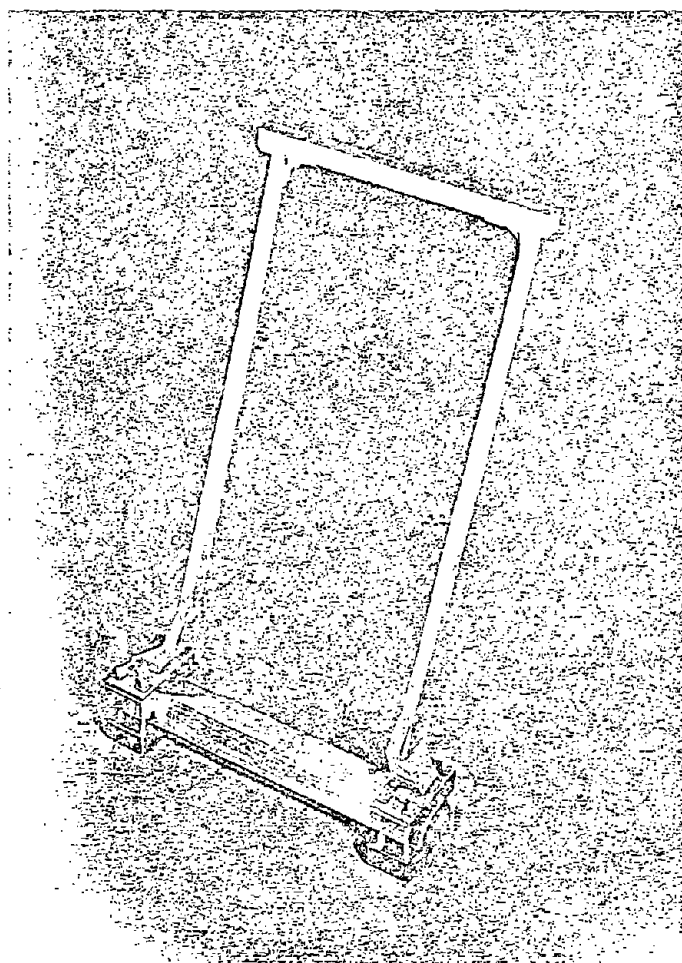


Figure 42. Quarter scale model of the Ill-frame.

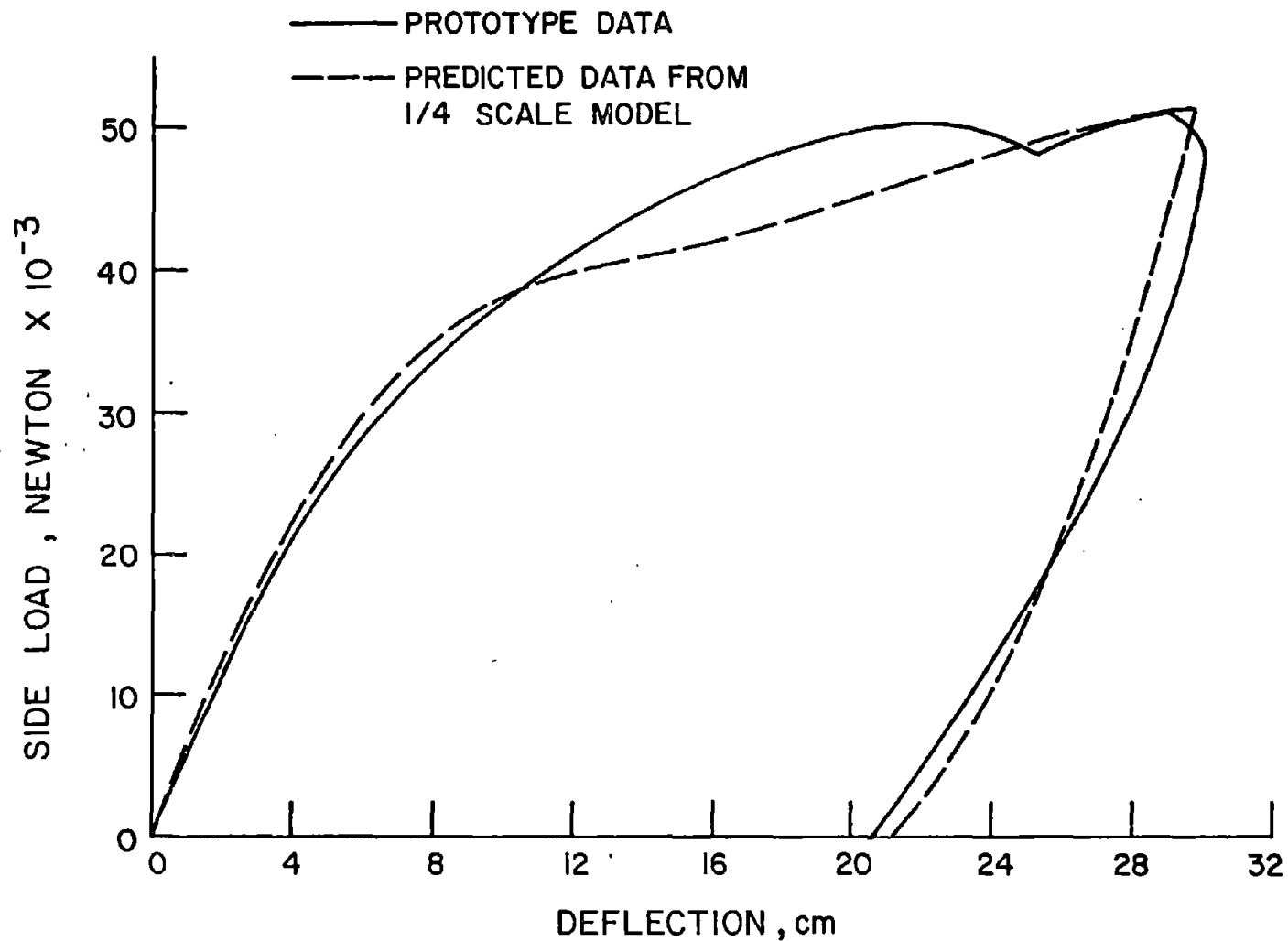


Figure 43. Similarity in the predicted load-deflection data using a 1/4-scale model frame and the static test data of a commercially manufactured full size ROPS provided by the International Harvester Company.

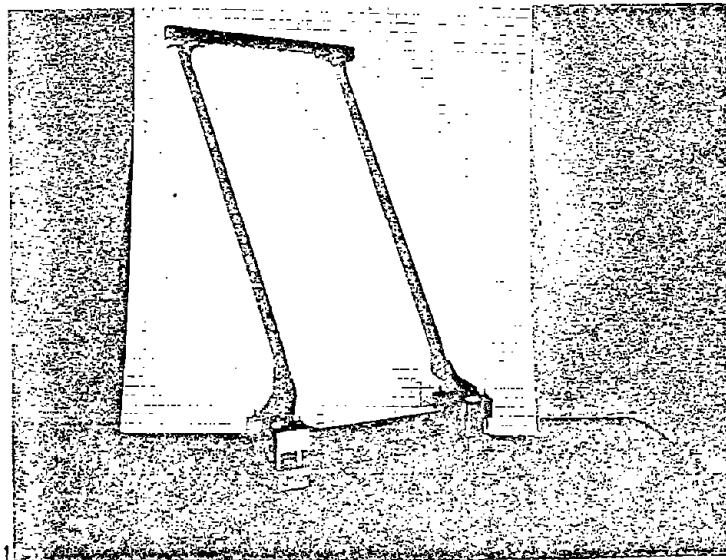
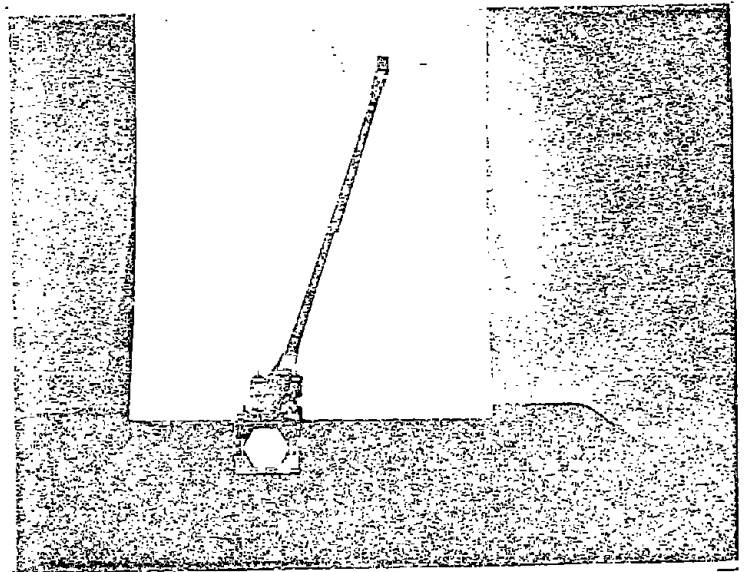
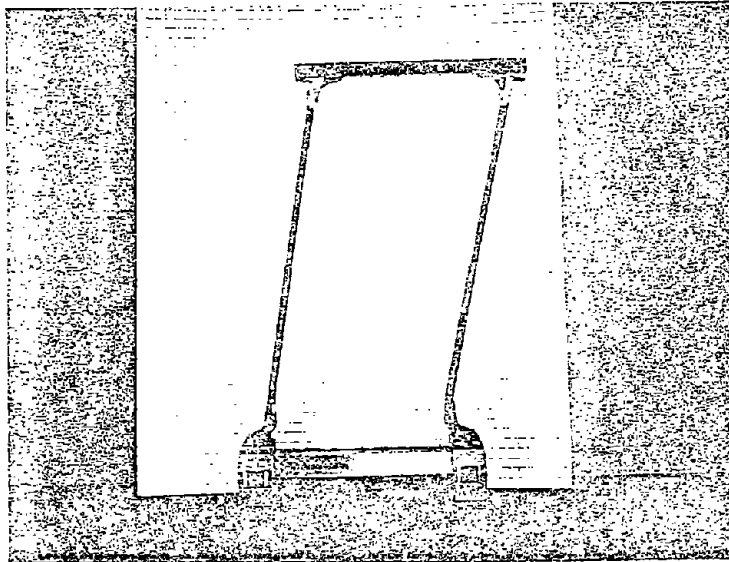


Figure Figure 44. The quarter scale frame after static testing.

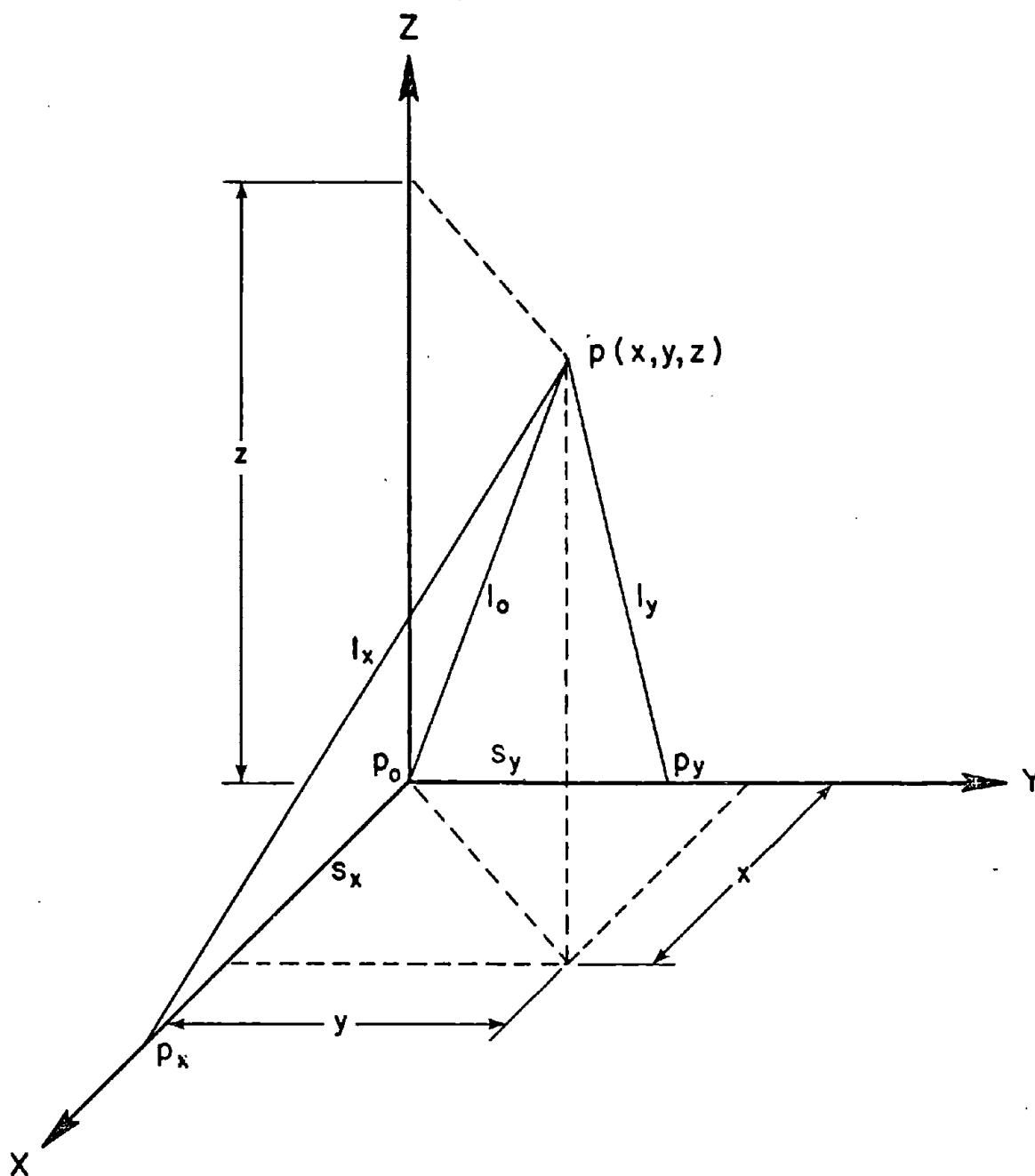


Figure 45. Coordinate system used to illustrate the principle of the 3-dimensional frame deflection measuring transducer.

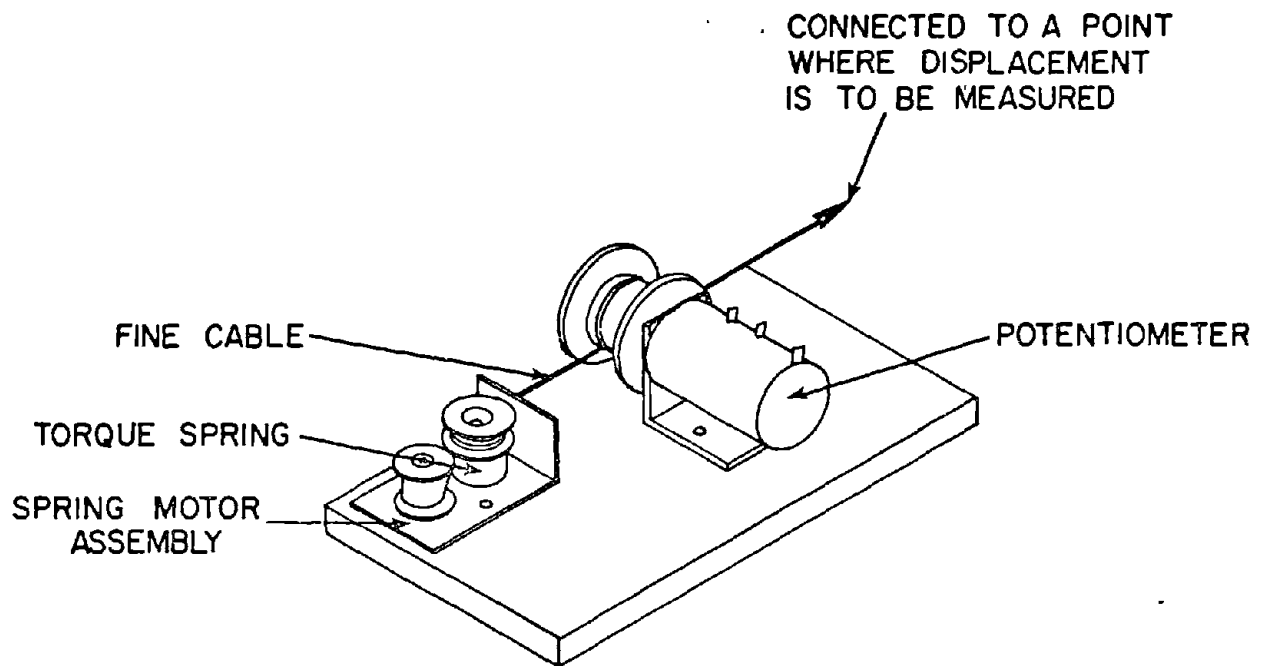


Figure 46. Constant torque motor and potentiometer unit as used to measure change in each line segment.

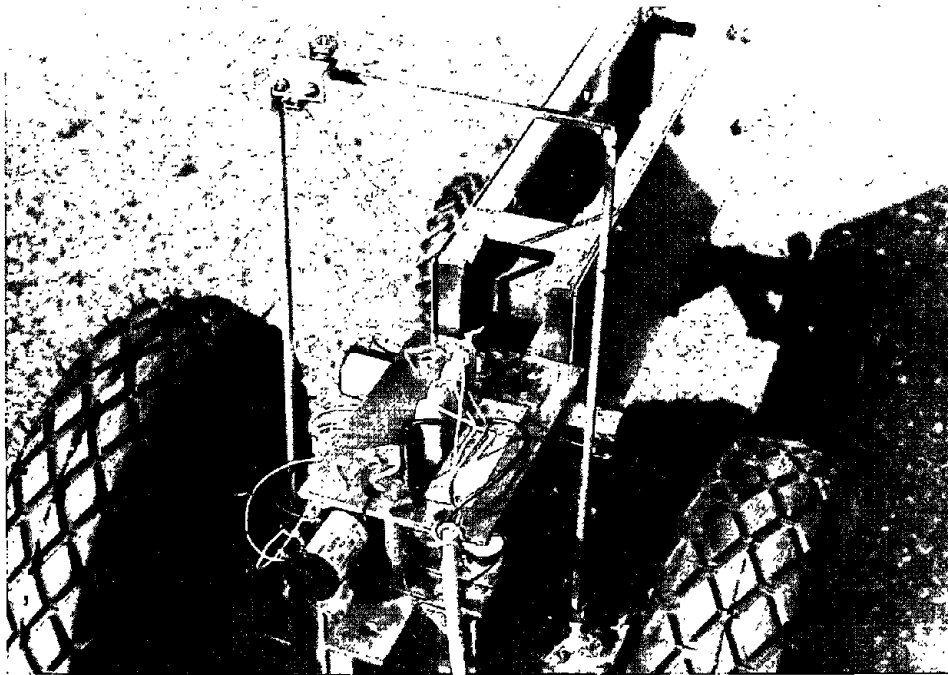


Figure 47. The three-dimensional deflection measuring transducer.

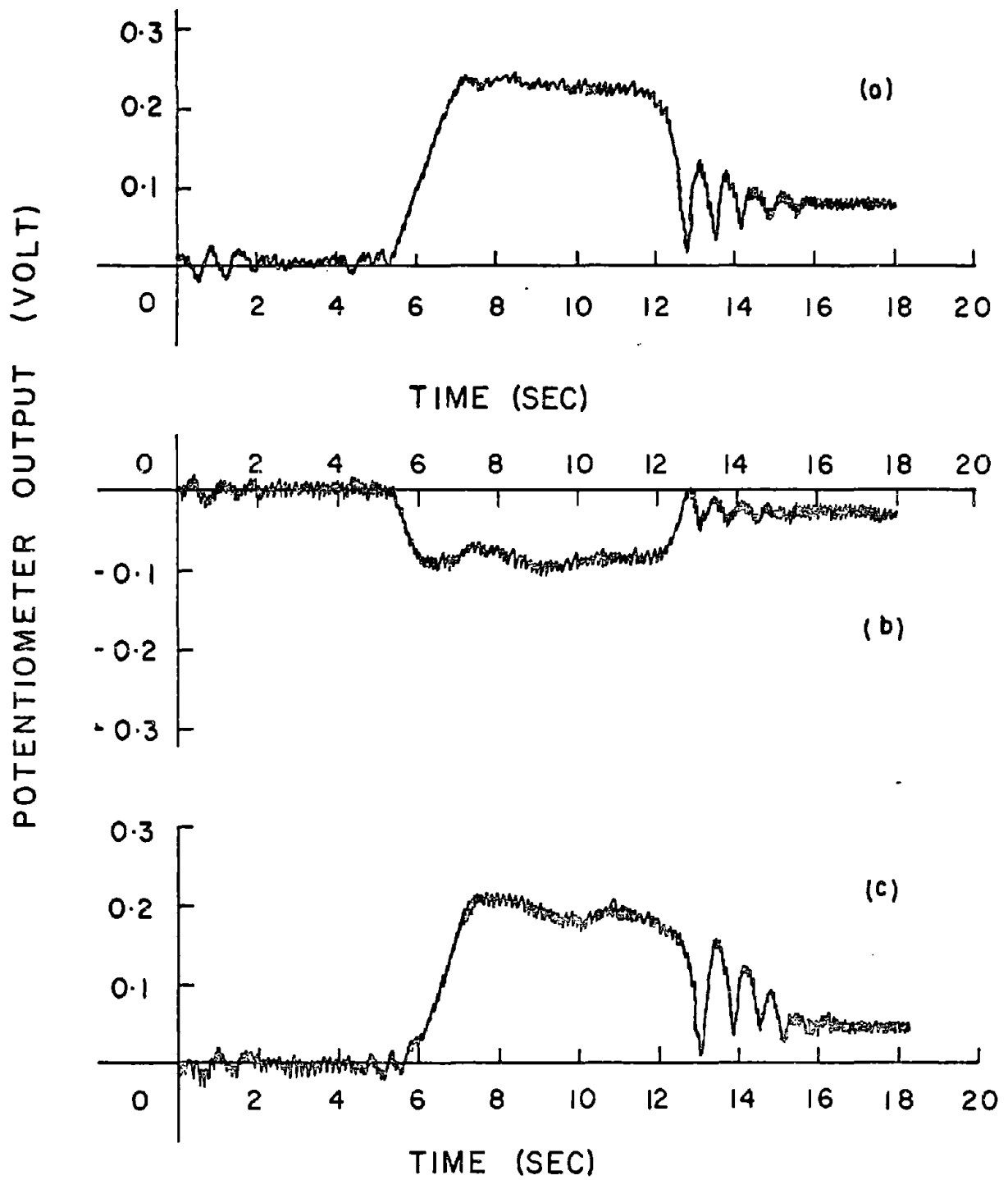


Figure 48. An example of the recorded output of the three-dimensional displacement transducer during an overturn test.

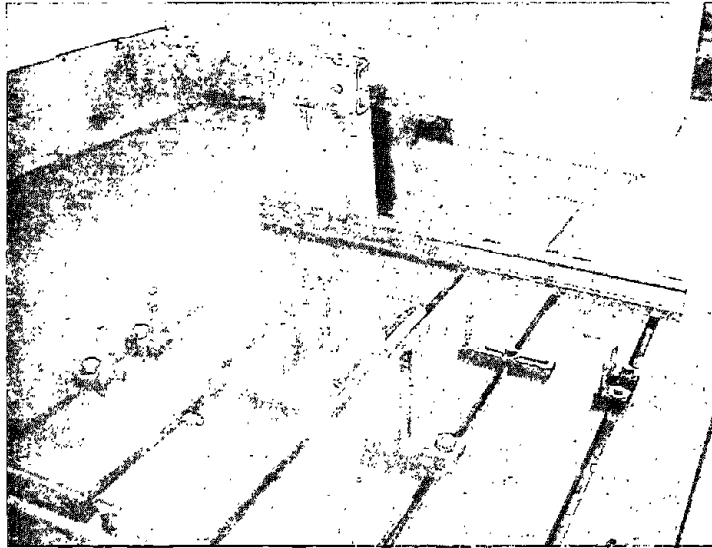


Figure 49a. Fixtures to measure final frame deformation after an overturn test.

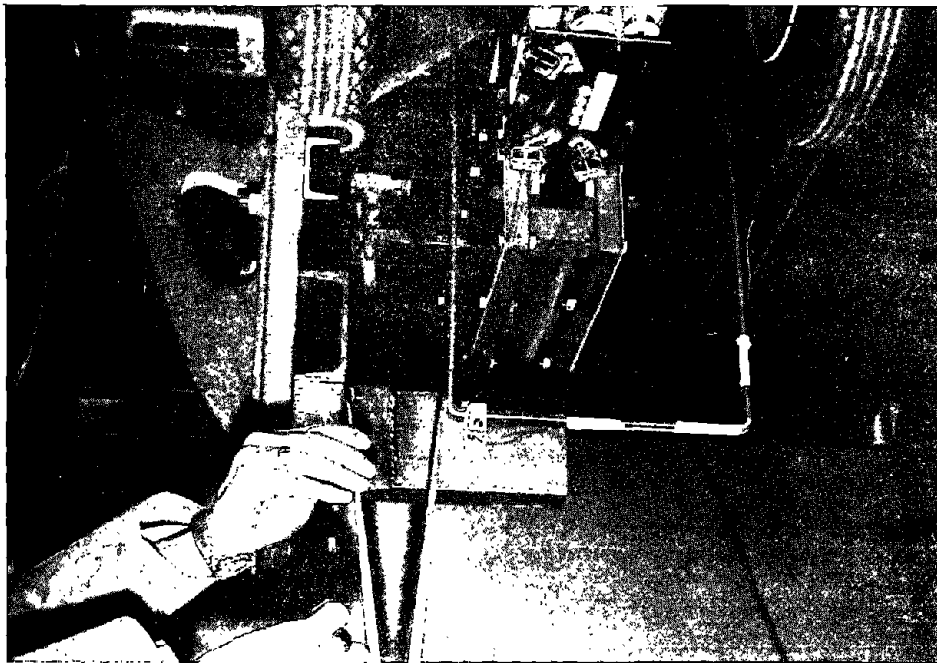


Figure 49b. Measurement is being made to determine frame deformation of the 1/4-scale frame.

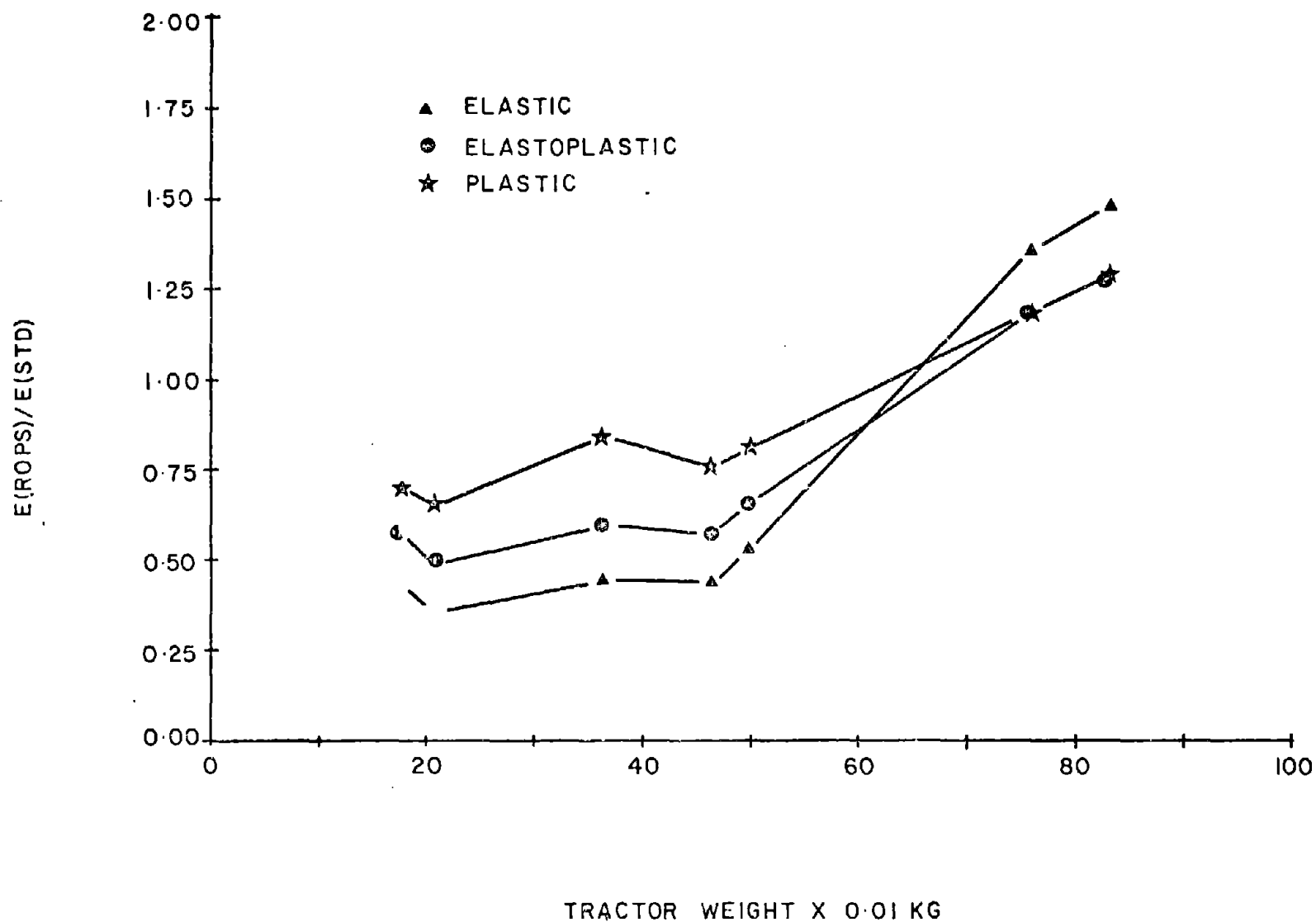


Figure 1. Effect of tractor weight and the type of frame deformation on the energy absorption with respect to the standard requirements.

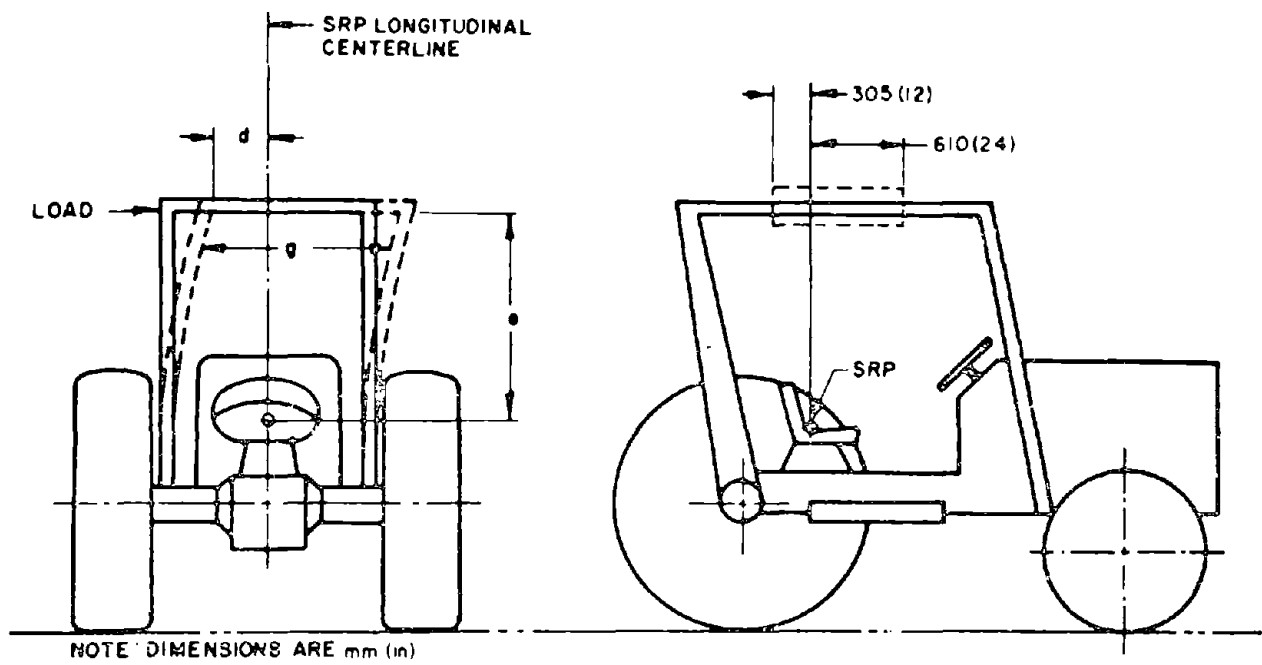


Figure 2. Typical side load application (Ref. ASAE Standard S383).

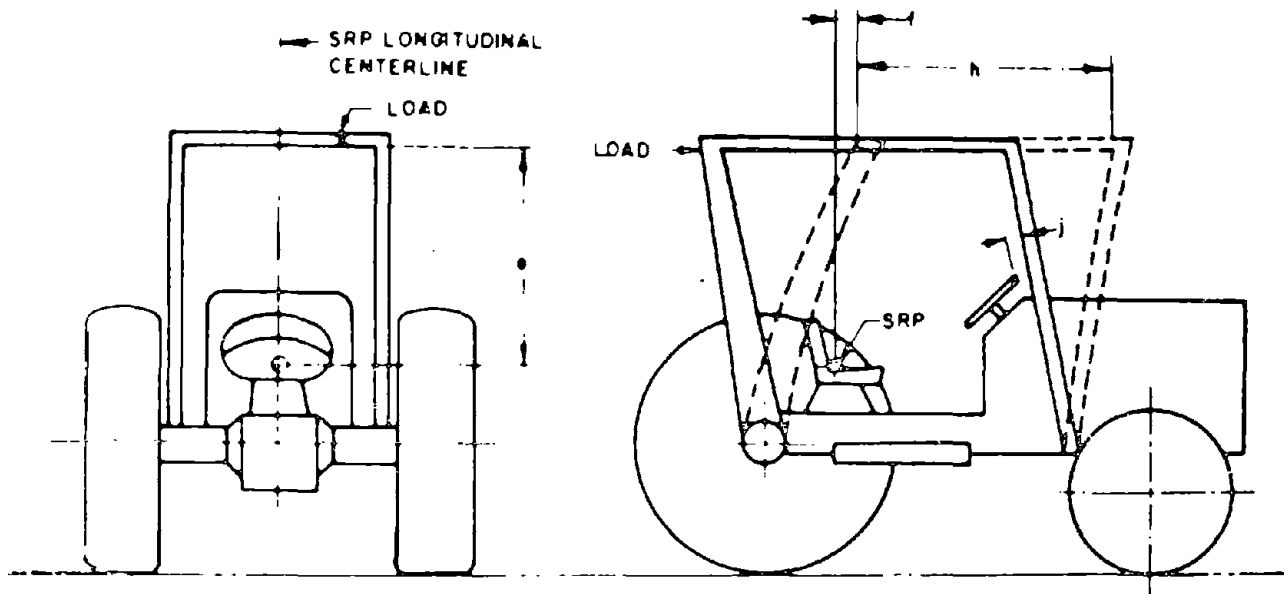


Figure 3. Typical rear load application (Ref. ASAE Standard S383).

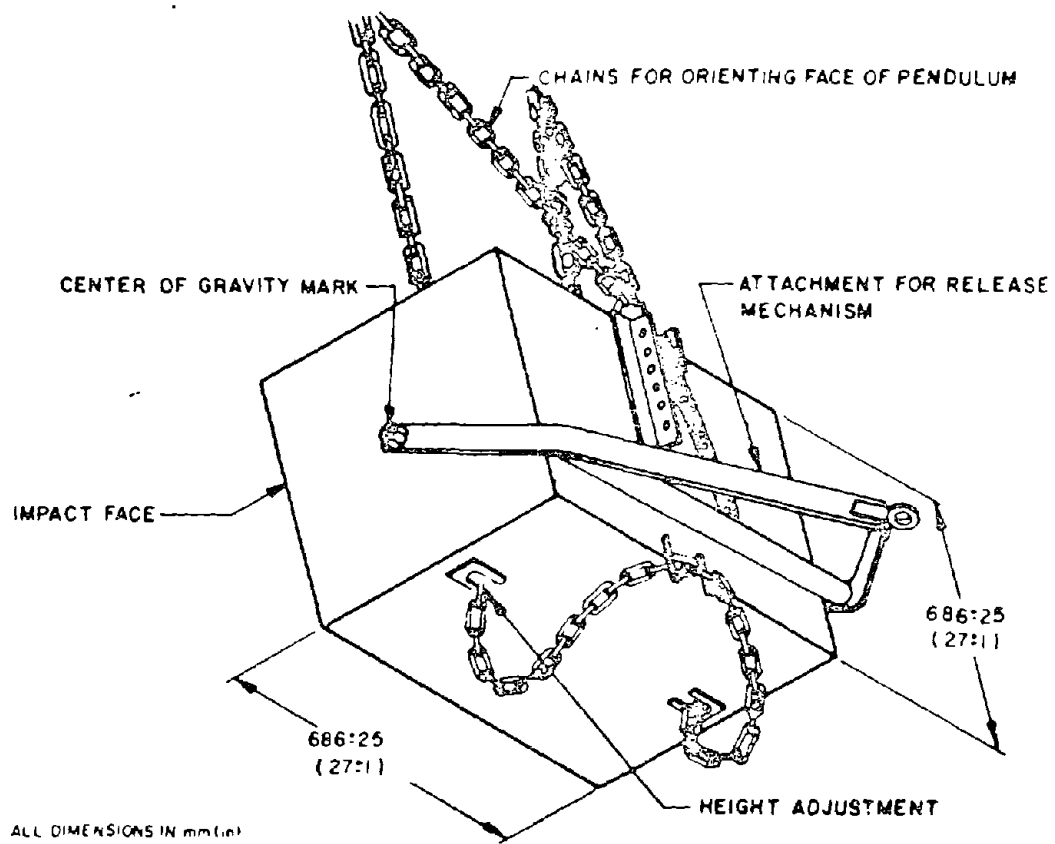


Figure 4. Standard impact pendulum (Ref. ASAE Standard S383).

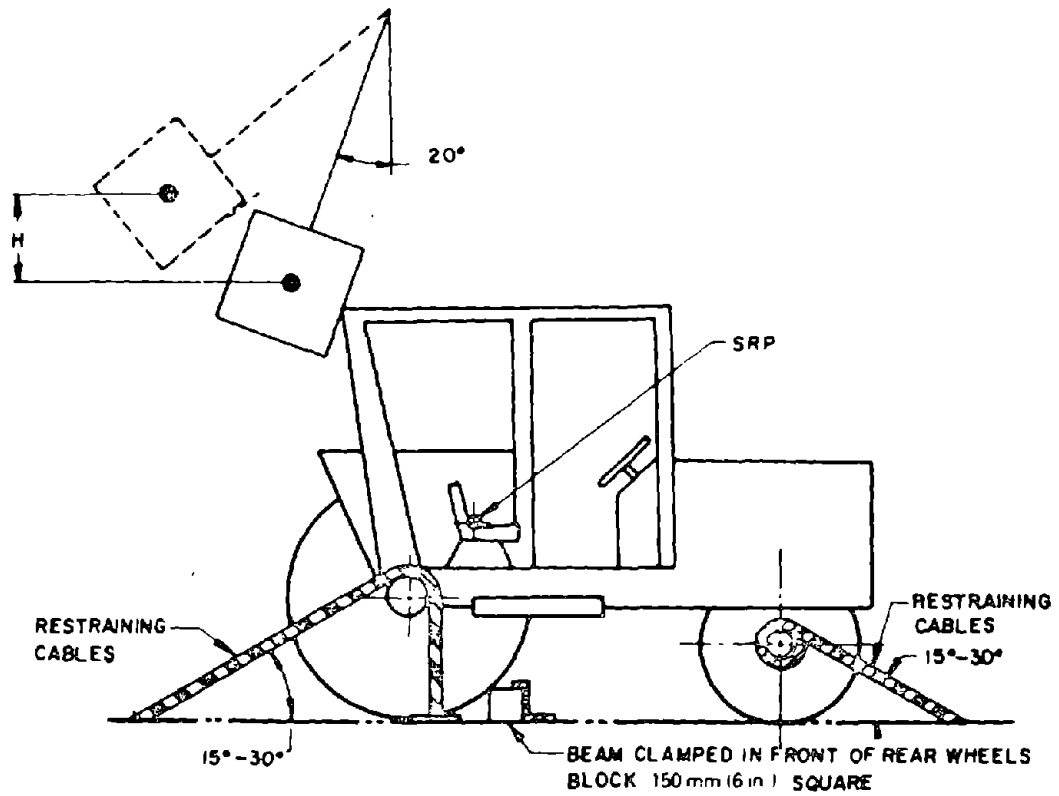


Figure 5. Typical rear impact application (Ref. ASAE Standard S383).

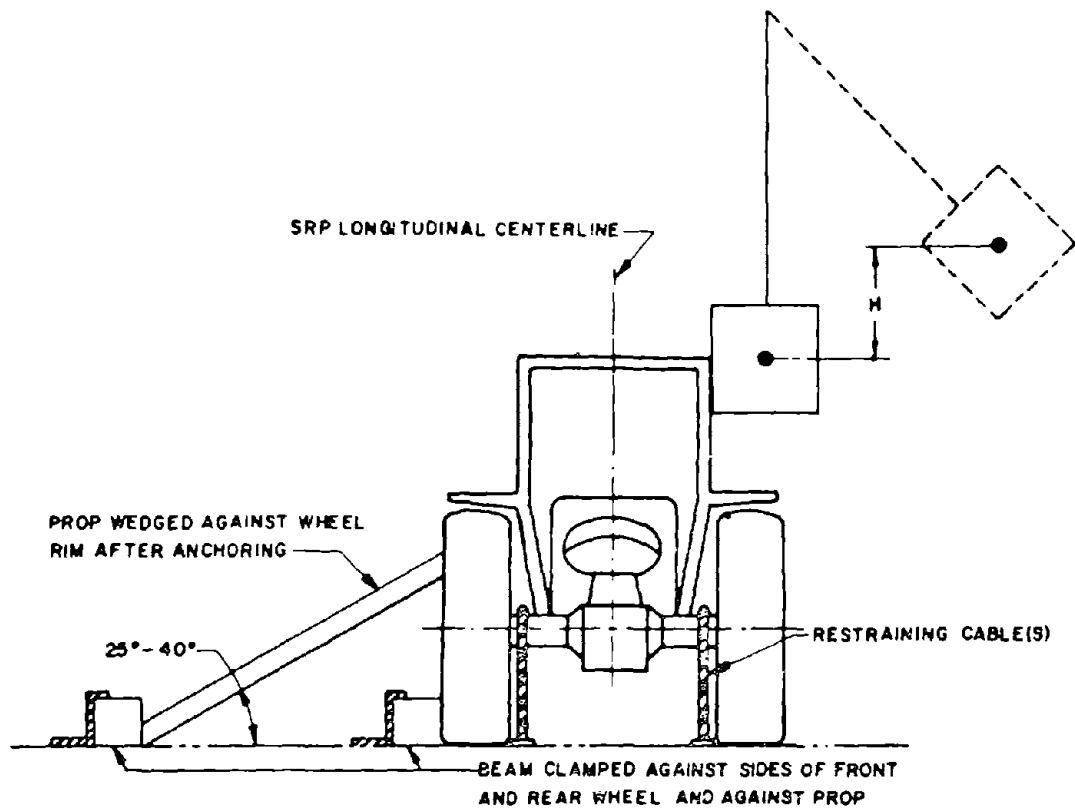


Figure 6. Typical side impact application (Ref. ASAE Standard S383).

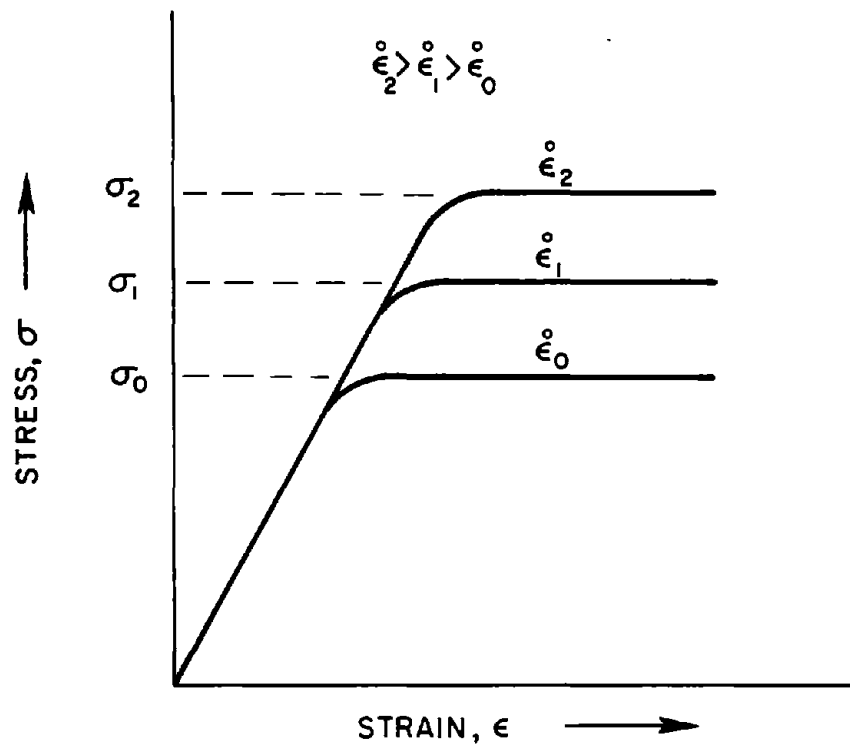


Figure 7. An idealized stress-strain diagram showing the effects of rate of strain on yield stress.

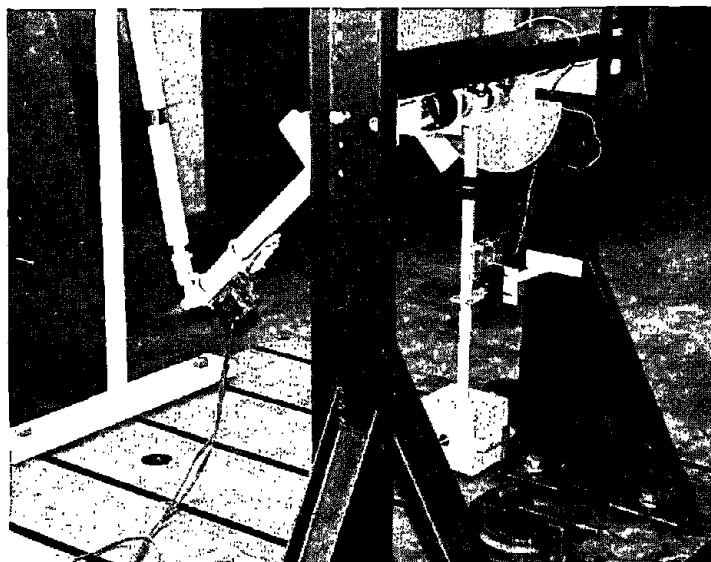


Figure 8. A general view of the experimental equipment showing the impact pendulum, the release mechanism, an angular displacement transducer and a cantilever beam.

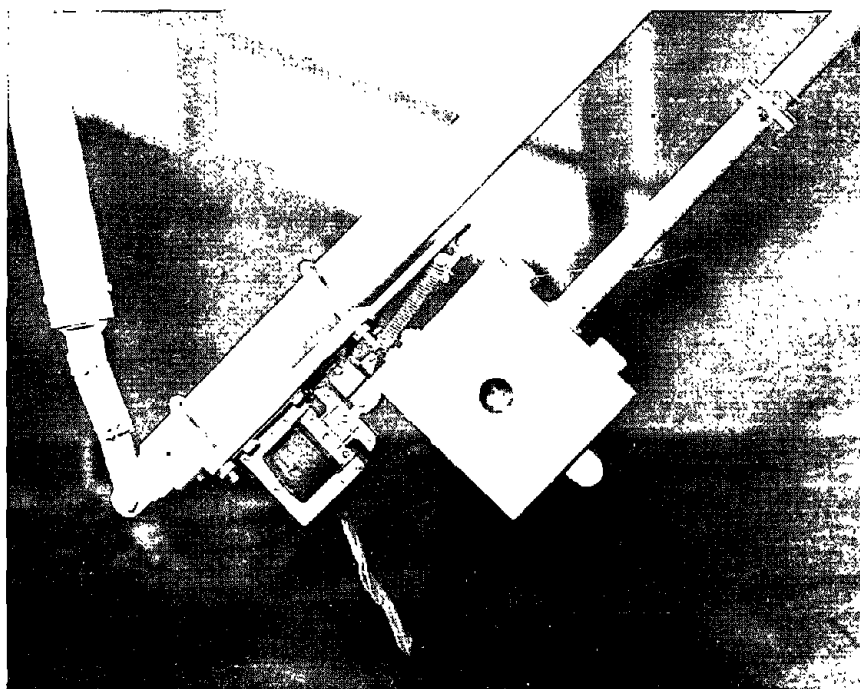


Figure 9. Pendulum release mechanism.

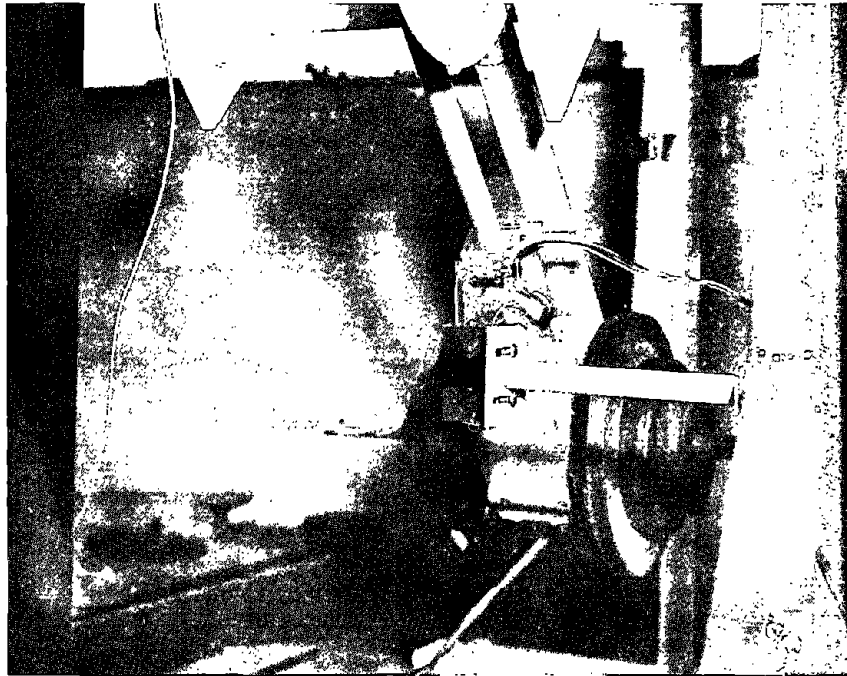


Figure 10. Method of adding weights to the pendulum.

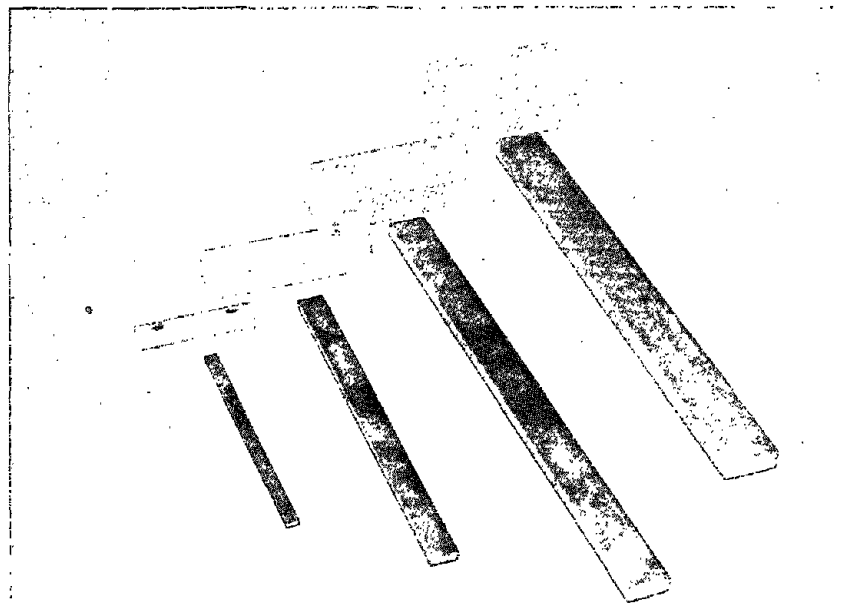


Figure 11. Different size beams and the impact pieces.

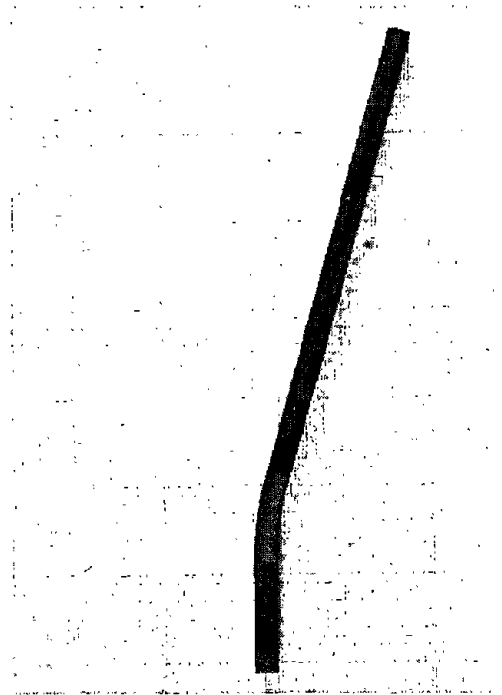


Figure 12. A typically deformed beam after impact by the pendulum. Figure also illustrates method of measuring permanent deflection.

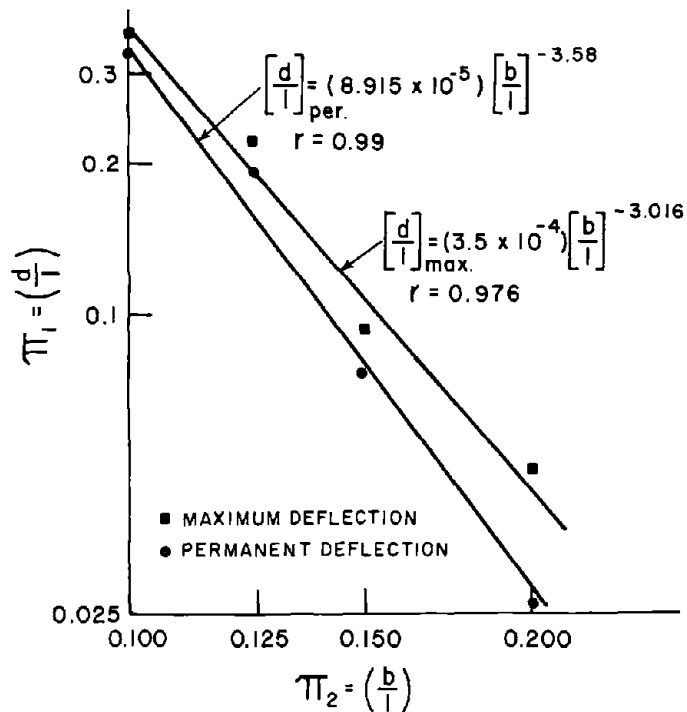


Figure 13. Effects of changing b/l on beam deflection. Other π -terms were held constant ($\pi_3 = 0.4$; $\pi_4 = 0.448$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

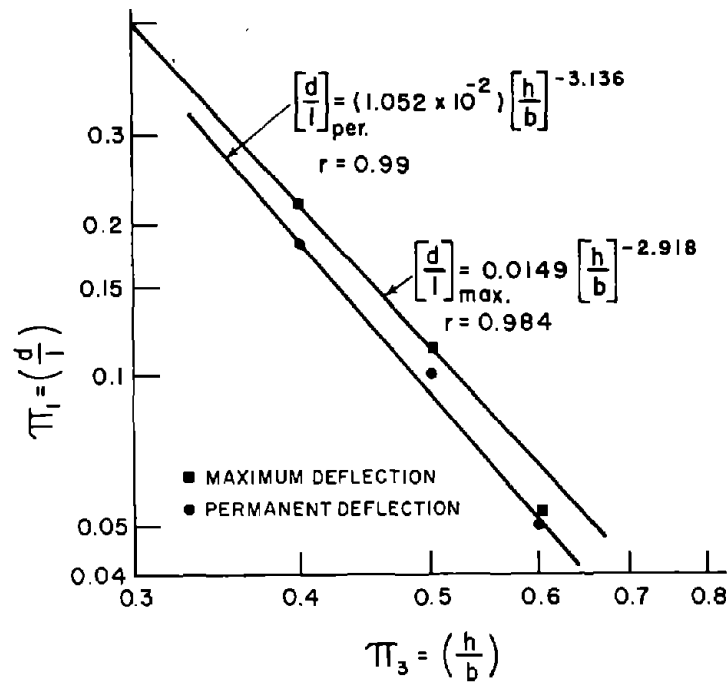


Figure 14. Effects of changing h/b on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_4 = 0.448$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

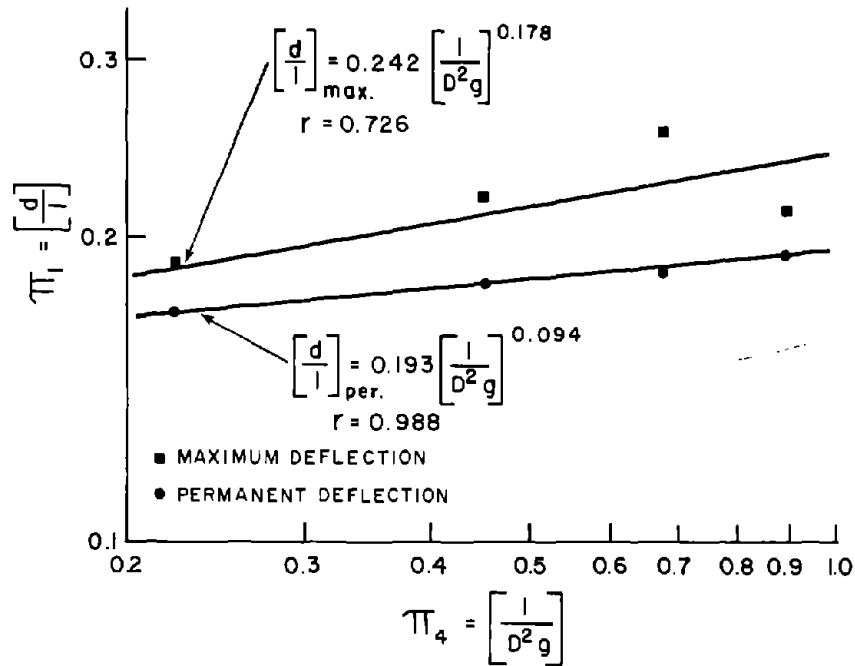


Figure 15. Effects of changing l/D^2g on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_5 = 16.34$; $\pi_6 = 0.96$).

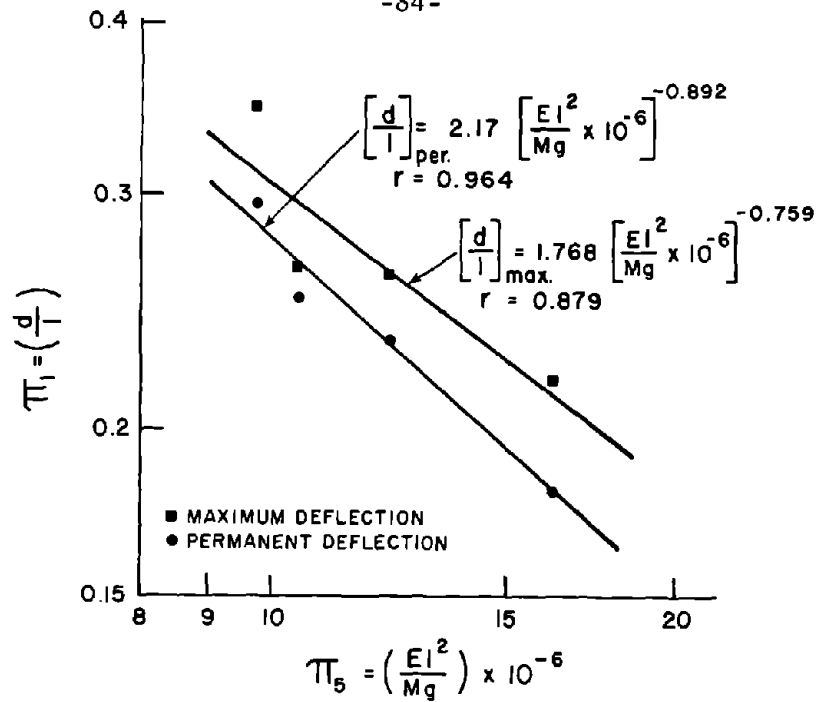


Figure 16. Effects of changing El^2/Mg on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_4 = 0.448$; $\pi_6 = 0.96$).

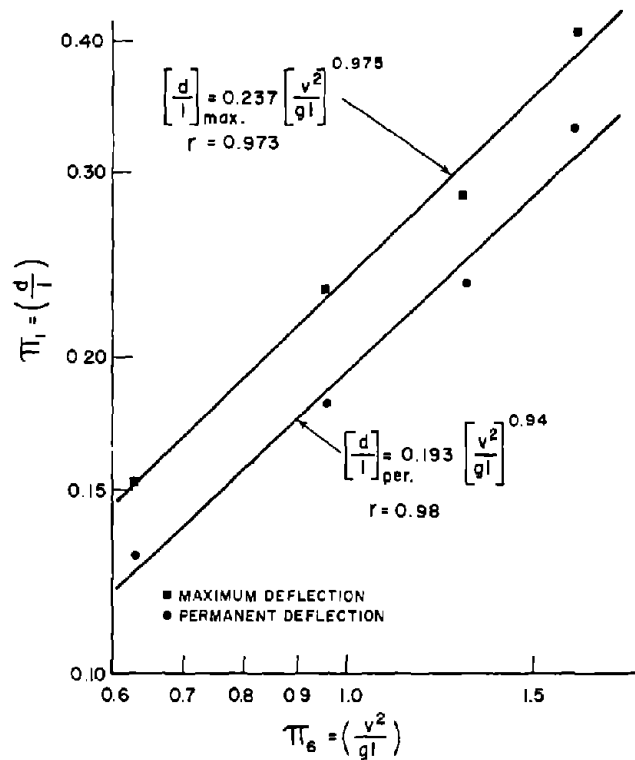


Figure 17. Effects of changing v^2/gl on beam deflection. Other π -terms were held constant ($\pi_2 = 0.125$; $\pi_3 = 0.4$; $\pi_4 = 0.448$, $\pi_5 = 16.34$).

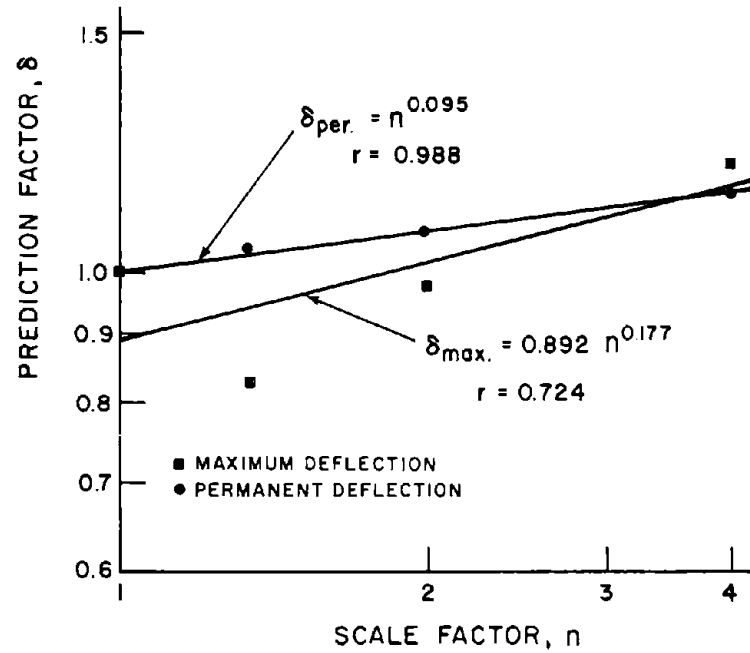


Figure 18. Effects of distortion in the strain rate design condition on prediction factors for dimensionless maximum and permanent deflections.

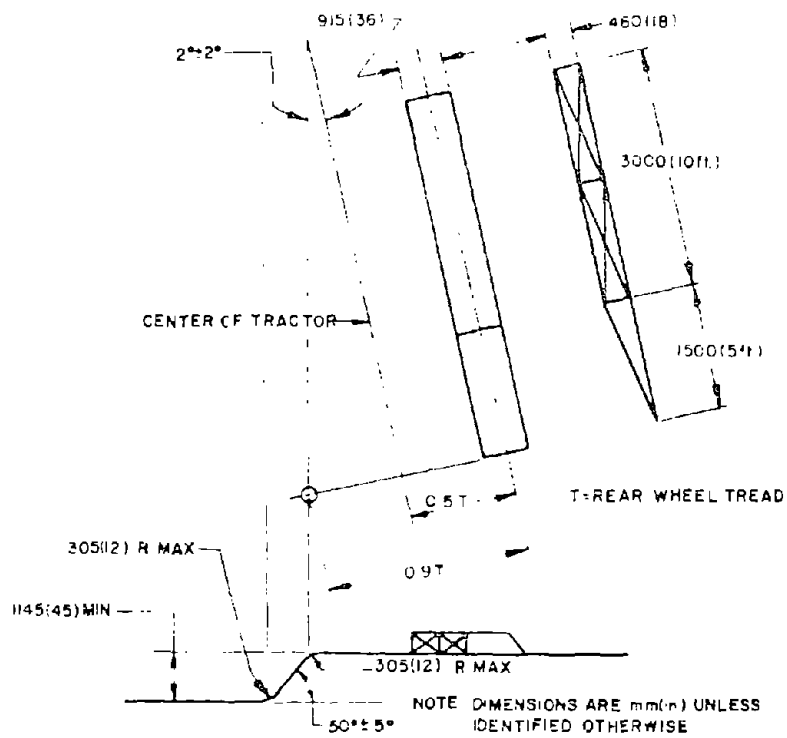


Figure 19. Typical overturn test bank and ramp (ASAE Standard S383).

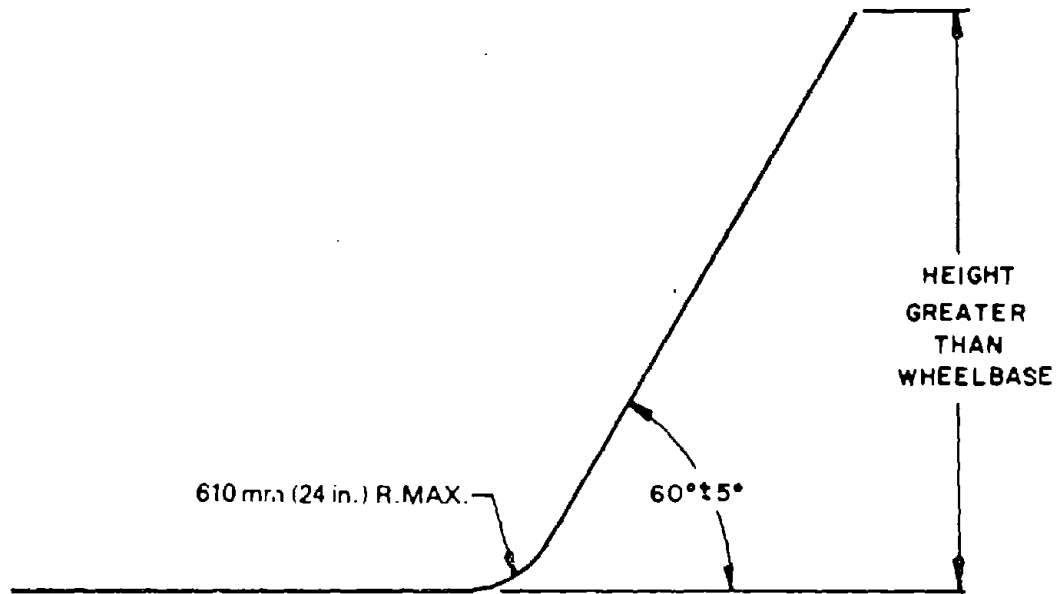


Figure 20. Rear overturn bank (ASAE Standard S383).

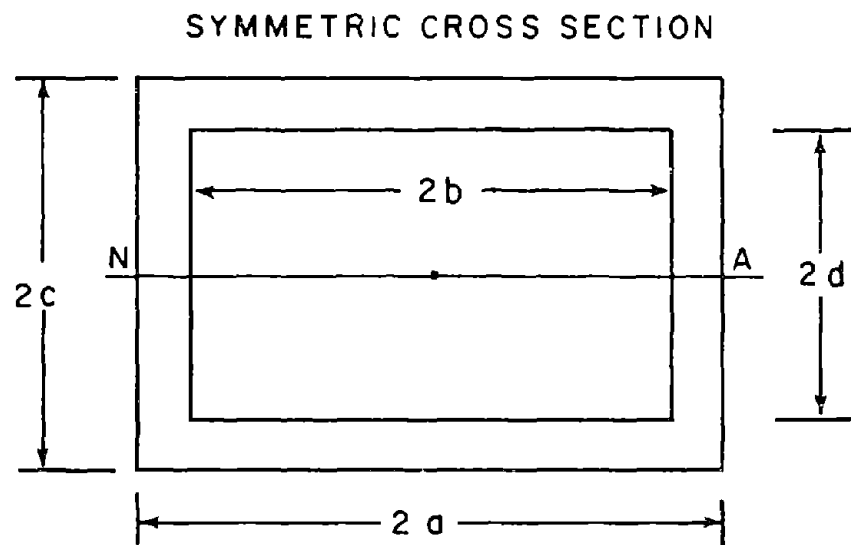


Figure 21. Symmetric rectangular hollow cross-section for calculation of shape factor.

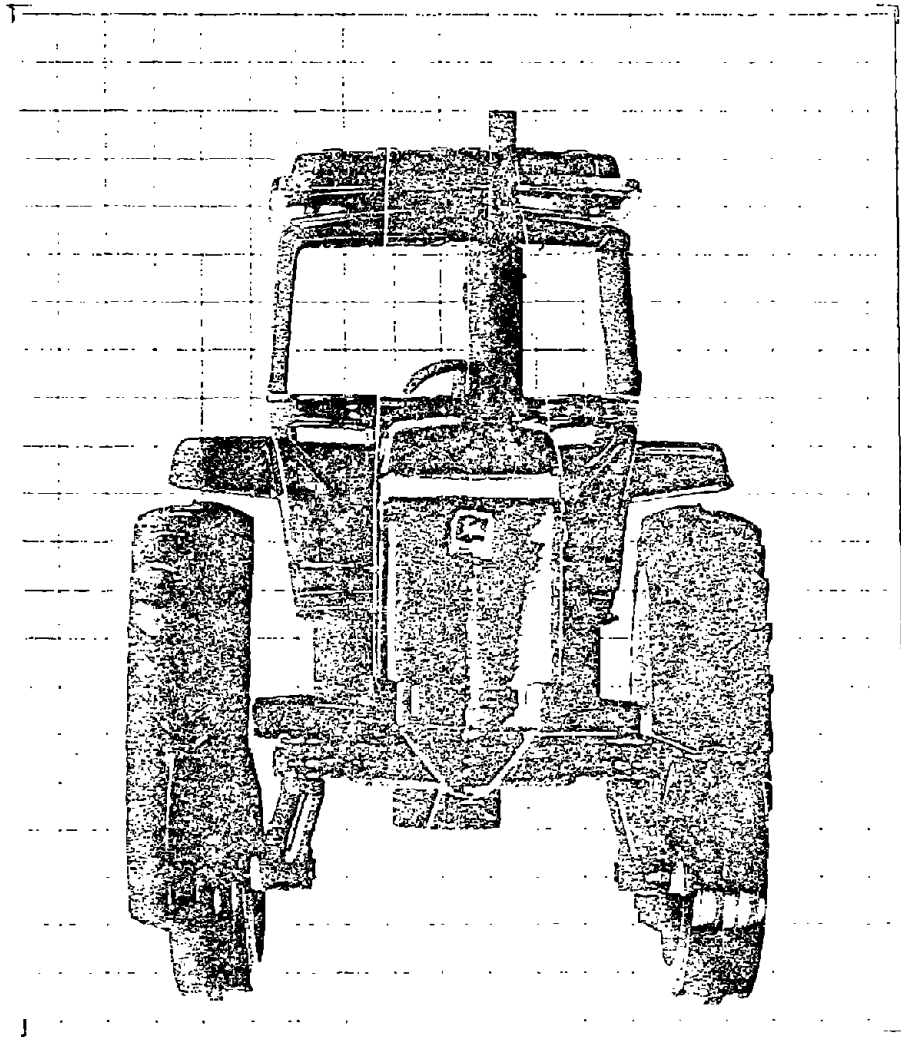


Figure 22. Front view of a 1/16-scale model toy tractor.

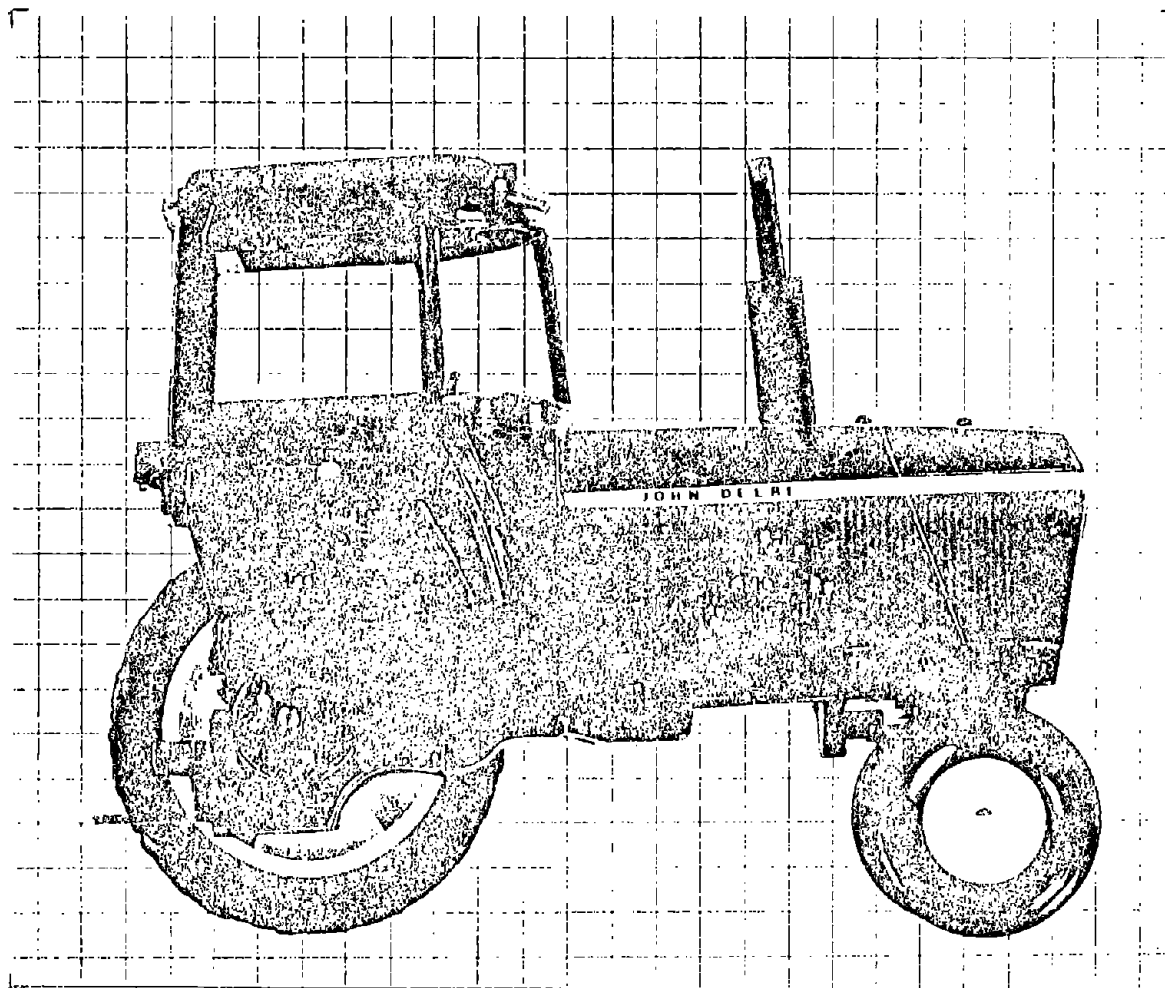


Figure 23. Side view of a 1/16-scale model toy tractor.

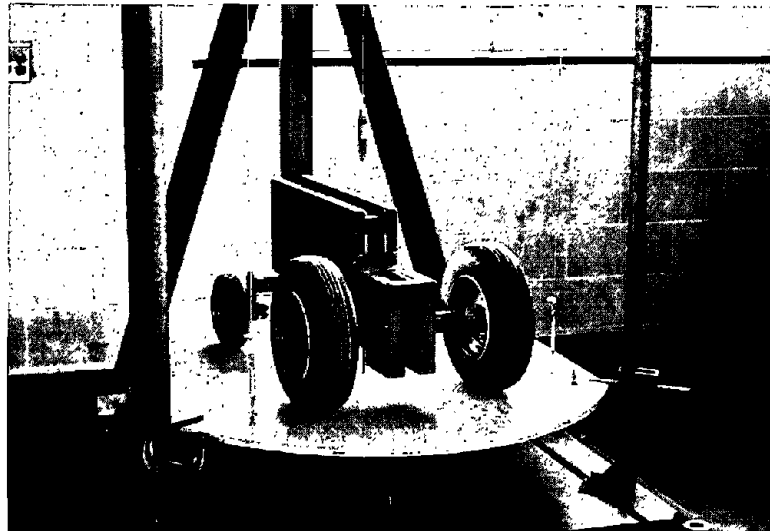


Figure 24. Determination of mass moments of inertia for model tractors using trifilar pendulum.

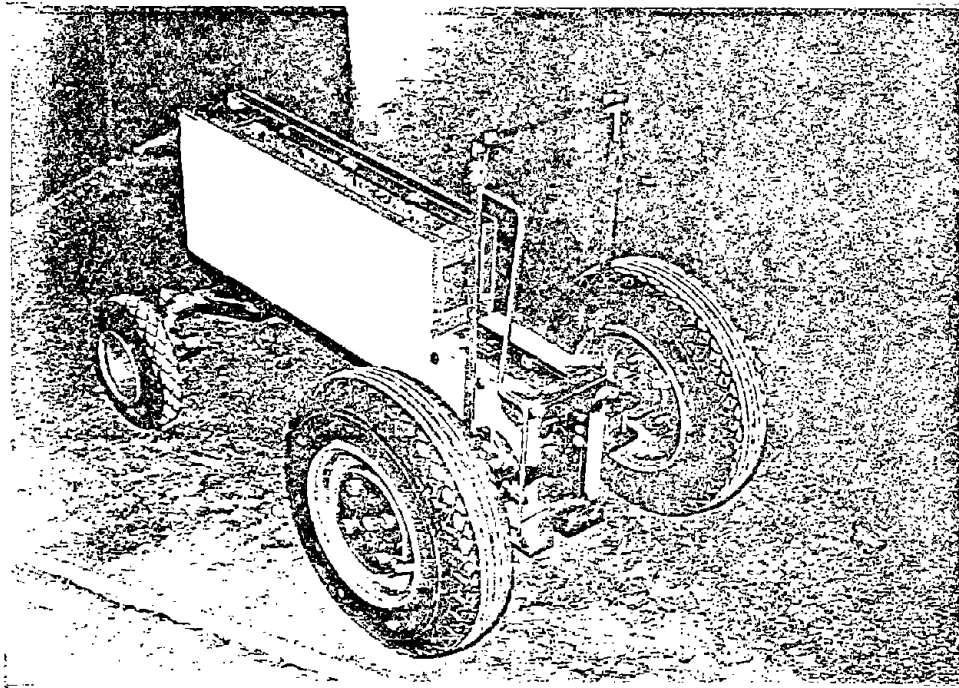


Figure 25. The quarter scale model tractor and ROPS.

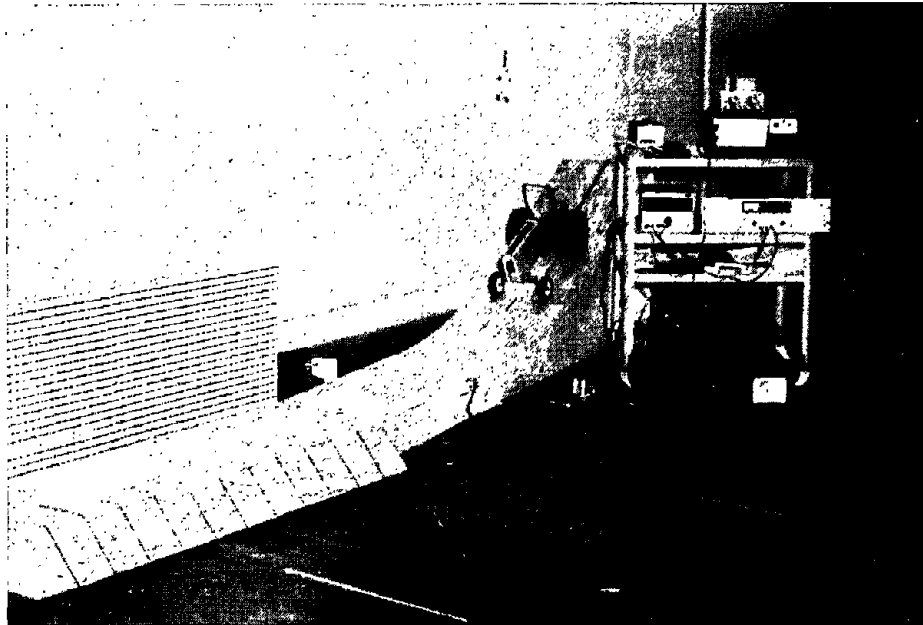


Figure 26. Experimental setup for model overturn test. Shown are the 1/8-scale tractor model on the starting ramp in the release position, the overturn ramp, the grid board for the motion analysis and the deflection measuring instrumentation.

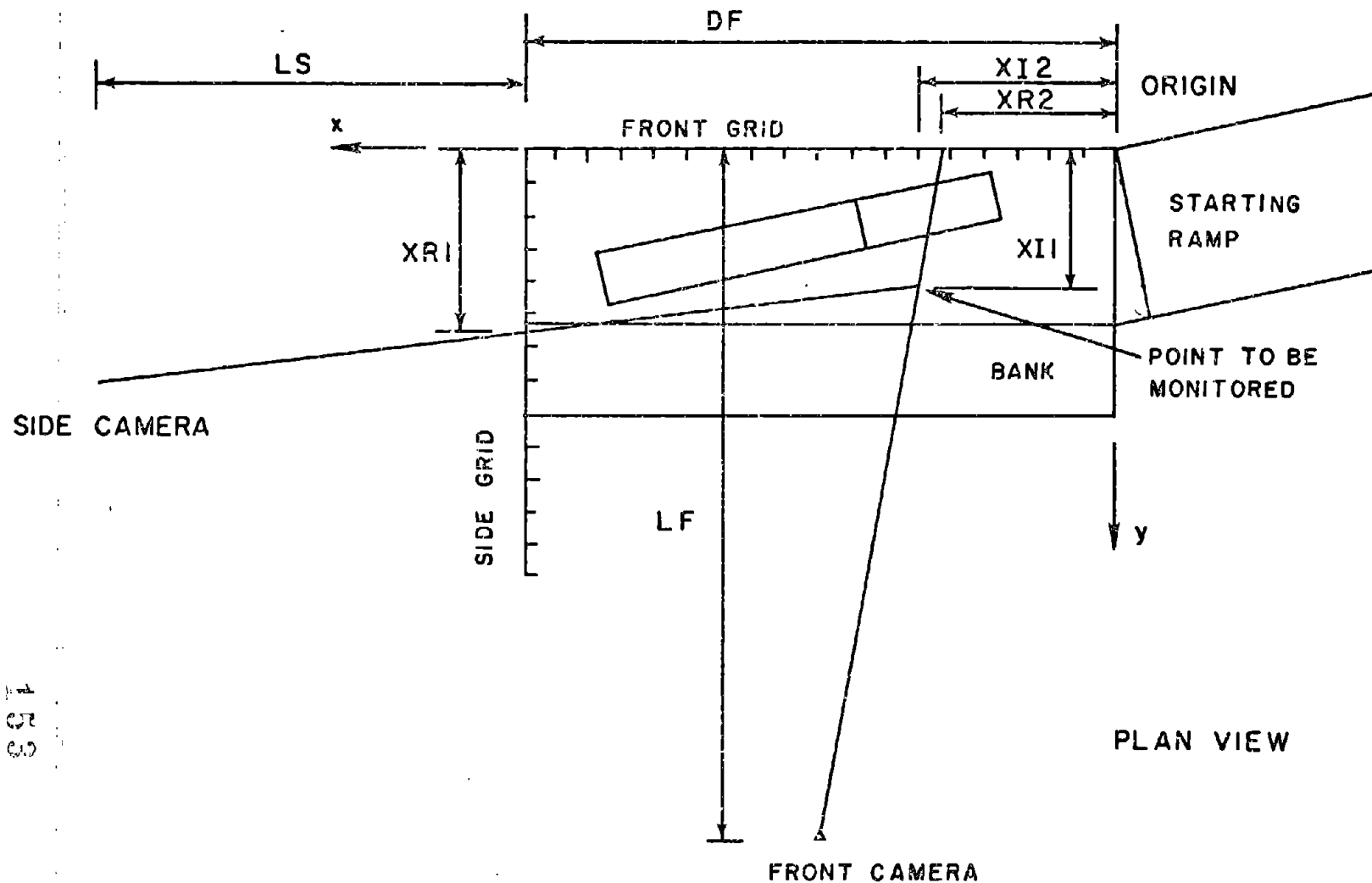


Figure 27. Plan view of the layout for high speed motion picture analysis.

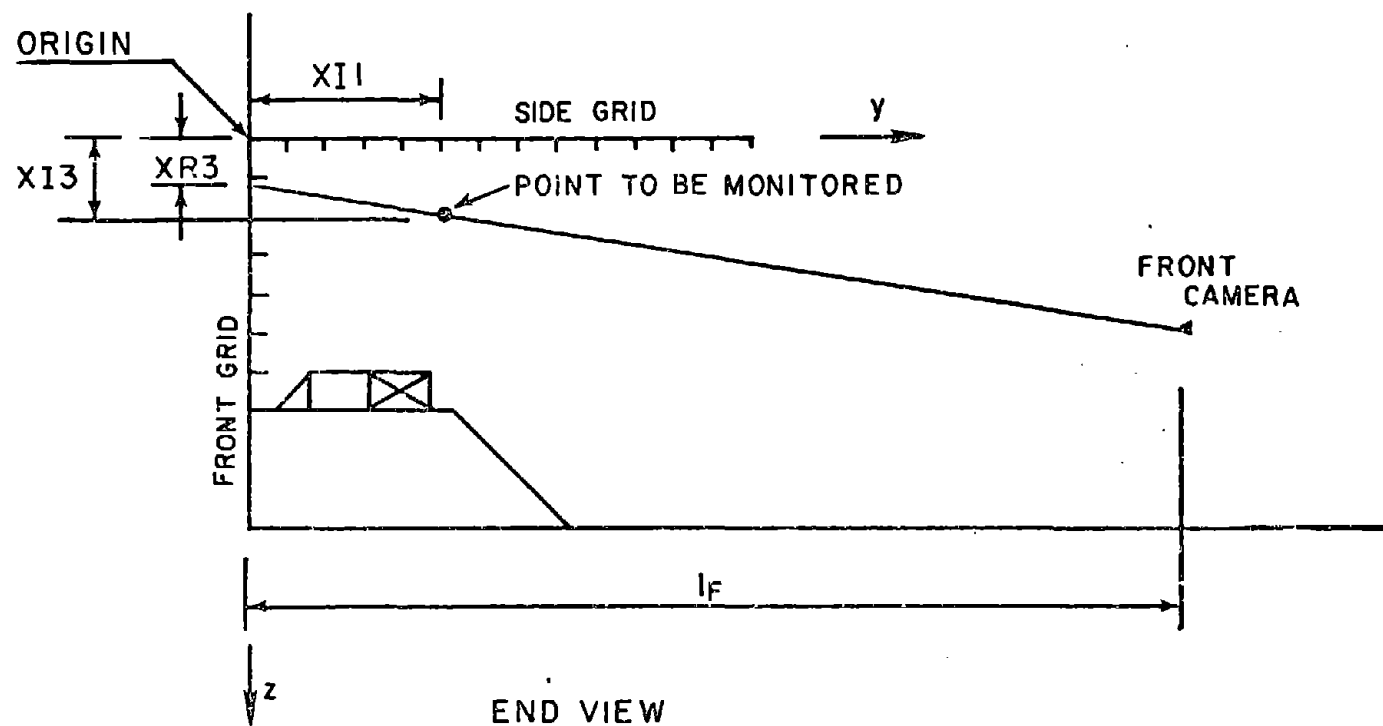


Figure 28. End view of the layout for high speed motion picture analysis.

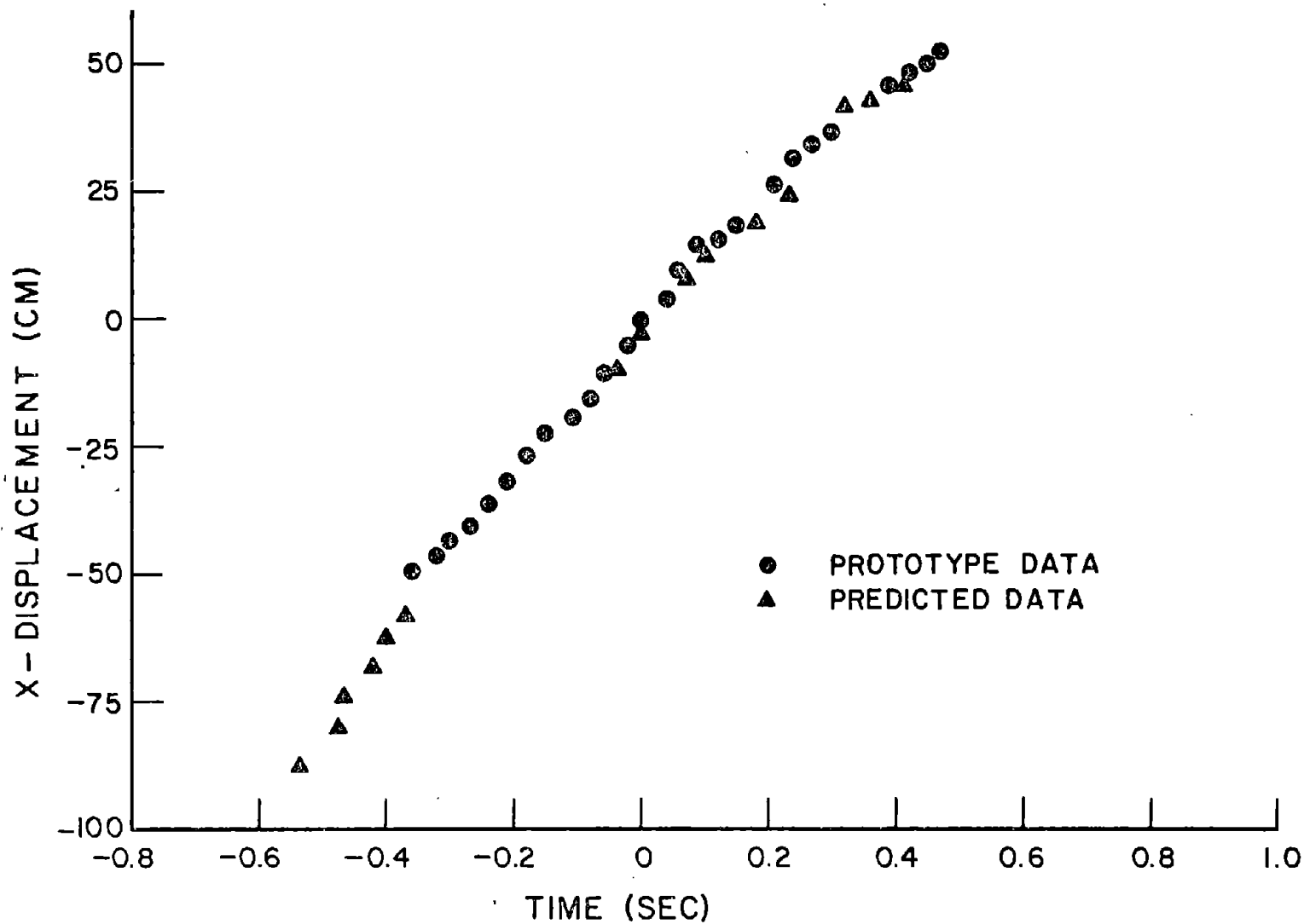


Figure 29. Results of high speed film analysis showing the displacement of left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

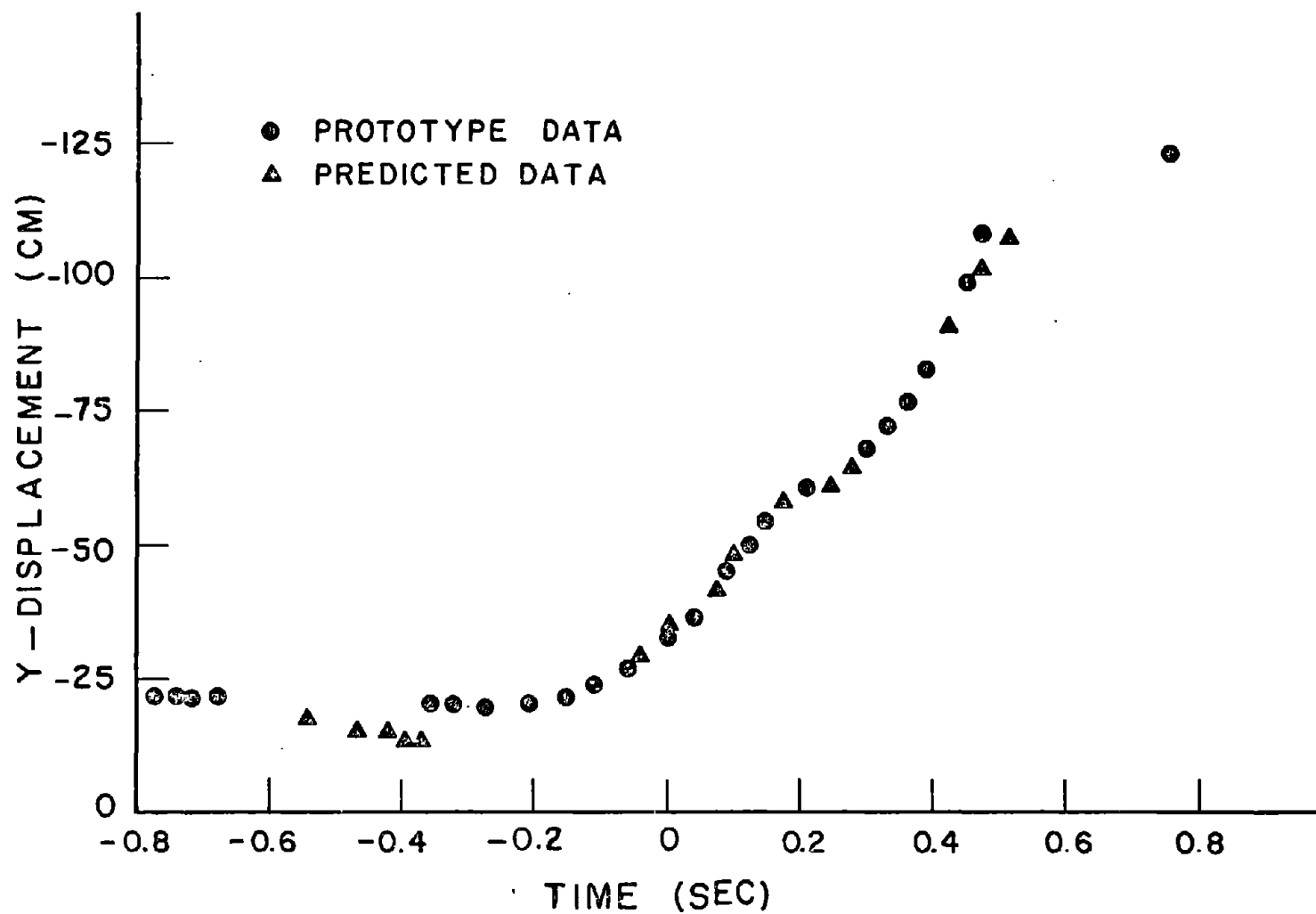


Figure 30. Results of high speed film analysis showing the displacement of left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

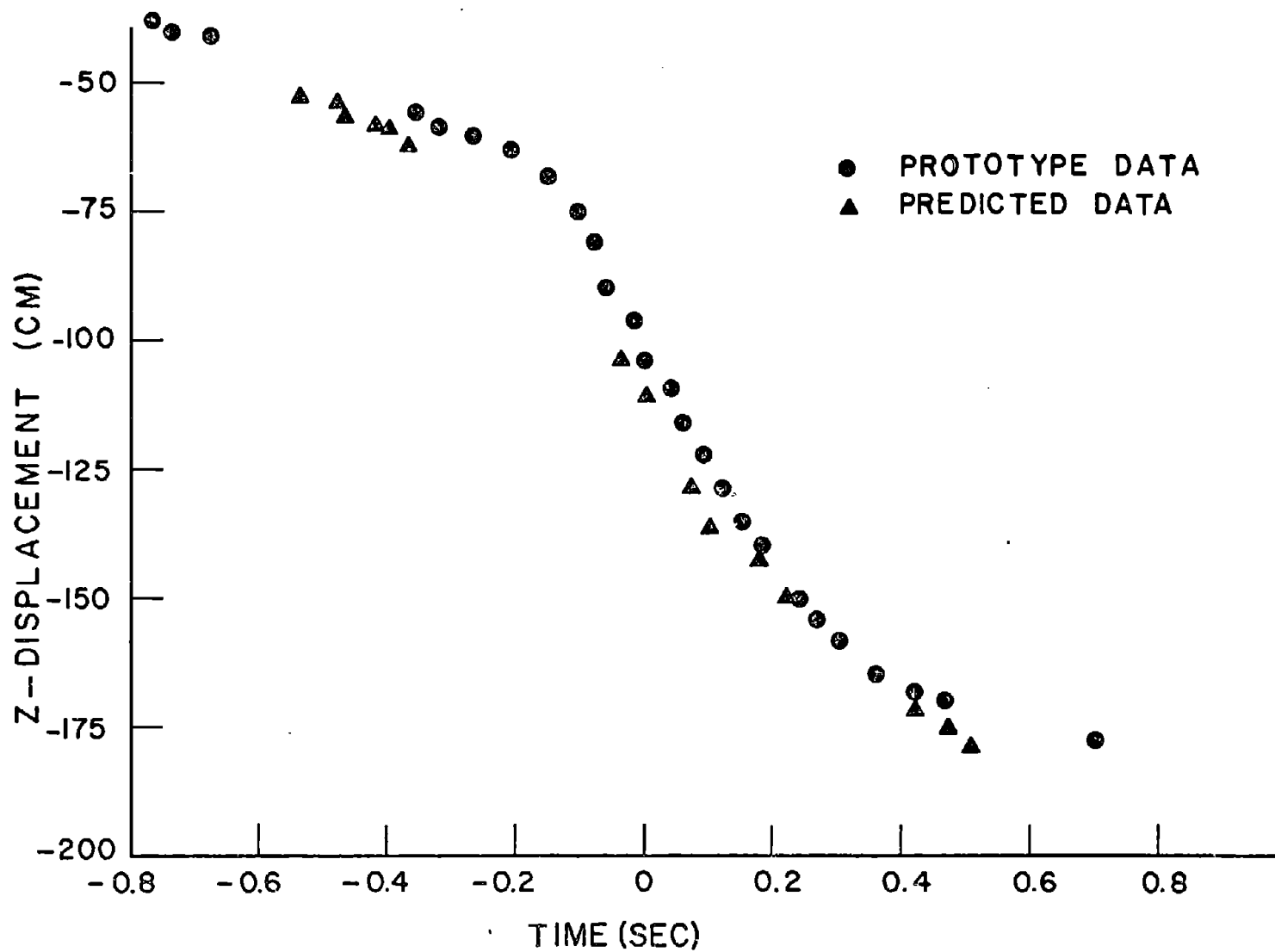
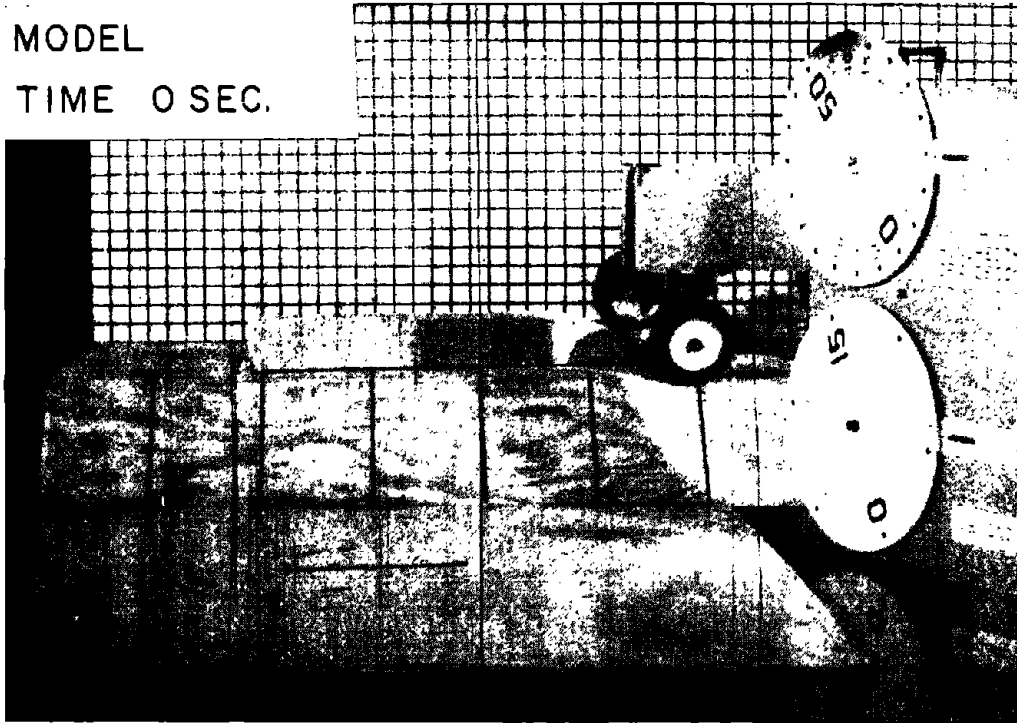


Figure 31. Results of high speed film analysis showing the displacement of the left hand upper ROPS corner for the 1/4-scale tractor model and the predicted motion using the 1/8-scale tractor model.

MODEL
TIME 0 SEC.



PROTOTYPE
TIME 0 SEC

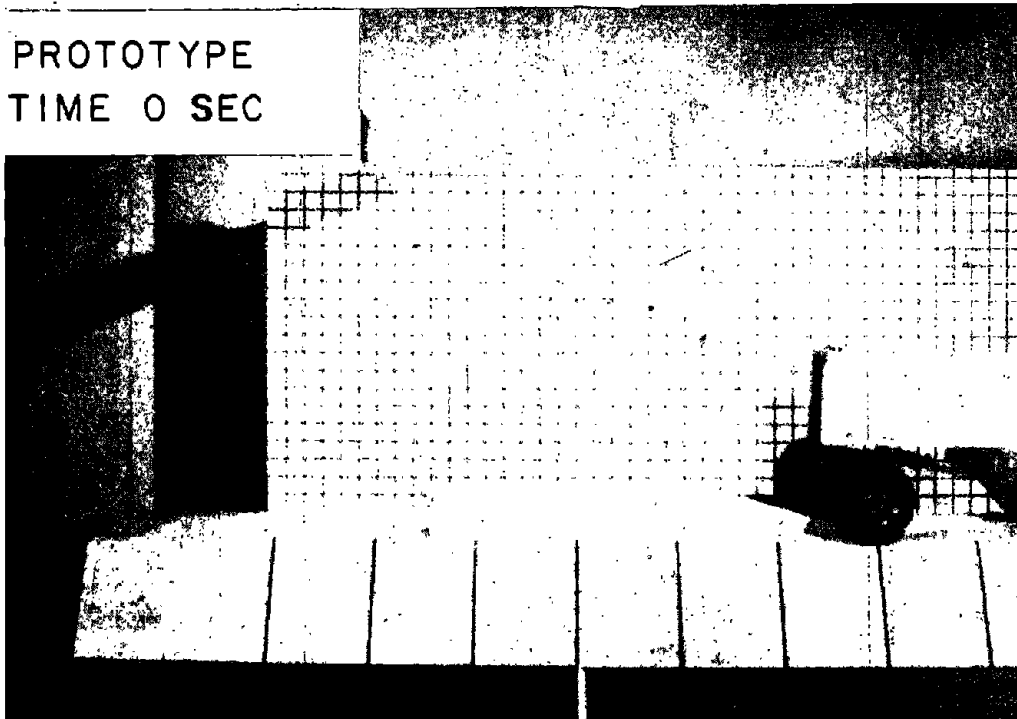
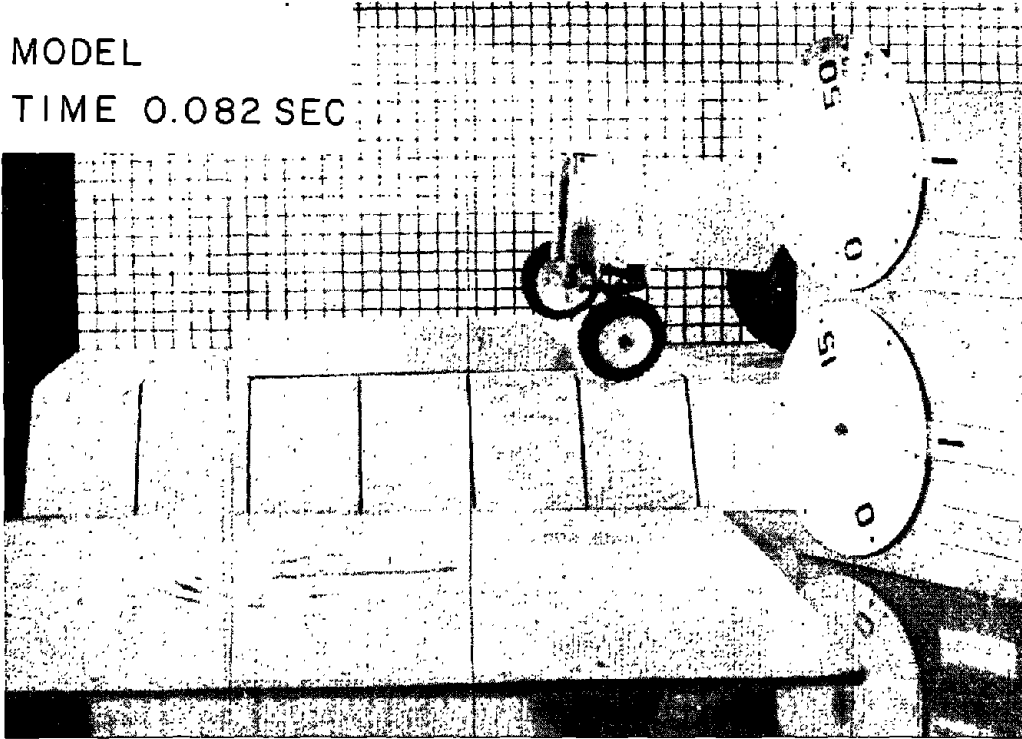


Figure 32. Sequence of photographs made from the high speed motion film to show similarity in the overturning behavior of the 1/4-scale and the 1/8-scale tractor models. The time interval of these photographs corresponds to the time scale factor, $t = \sqrt{n} \text{ tm}$.

MODEL
TIME 0.082 SEC



PROTOTYPE
TIME 0.115 SEC

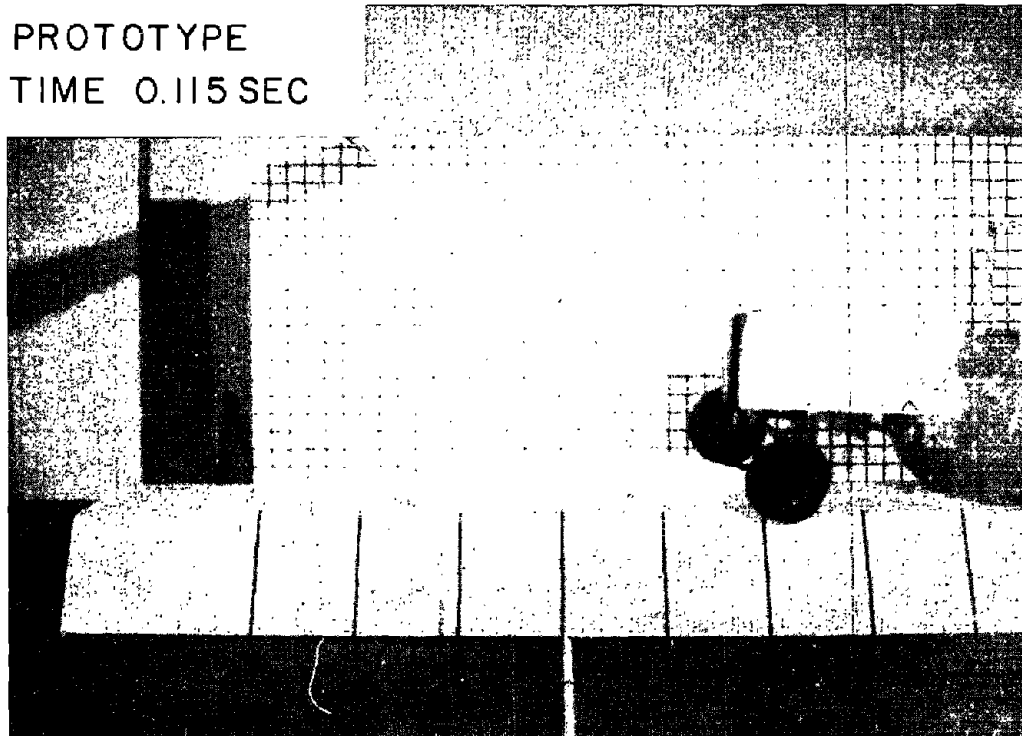
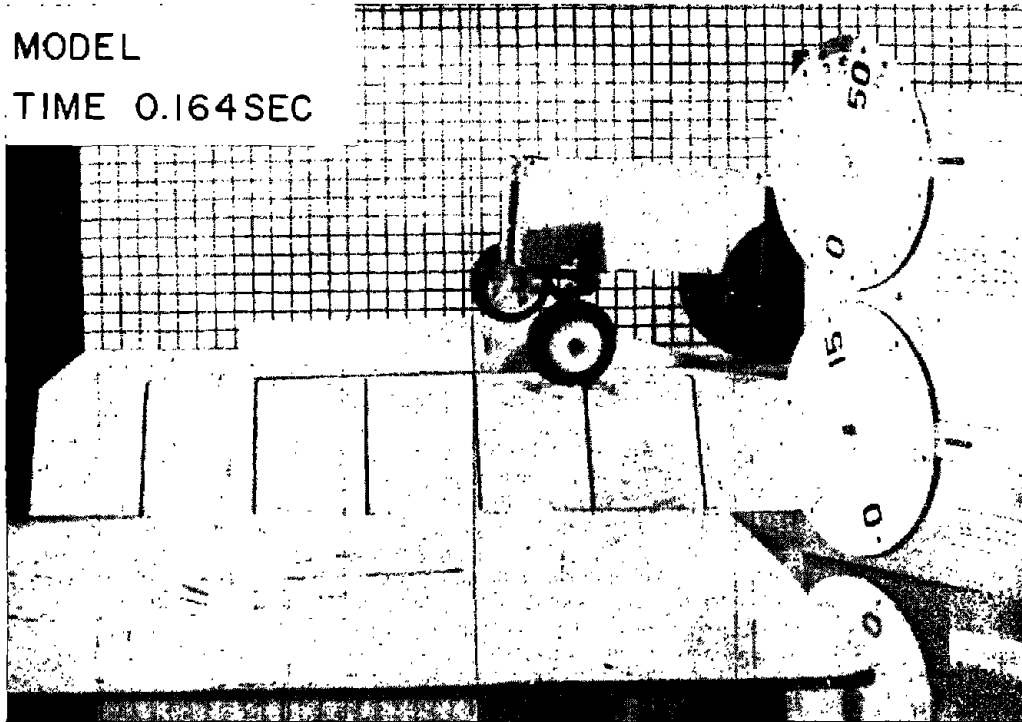


Figure 32. cont'd.

MODEL
TIME 0.164 SEC



PROTOTYPE
TIME 0.230 SEC

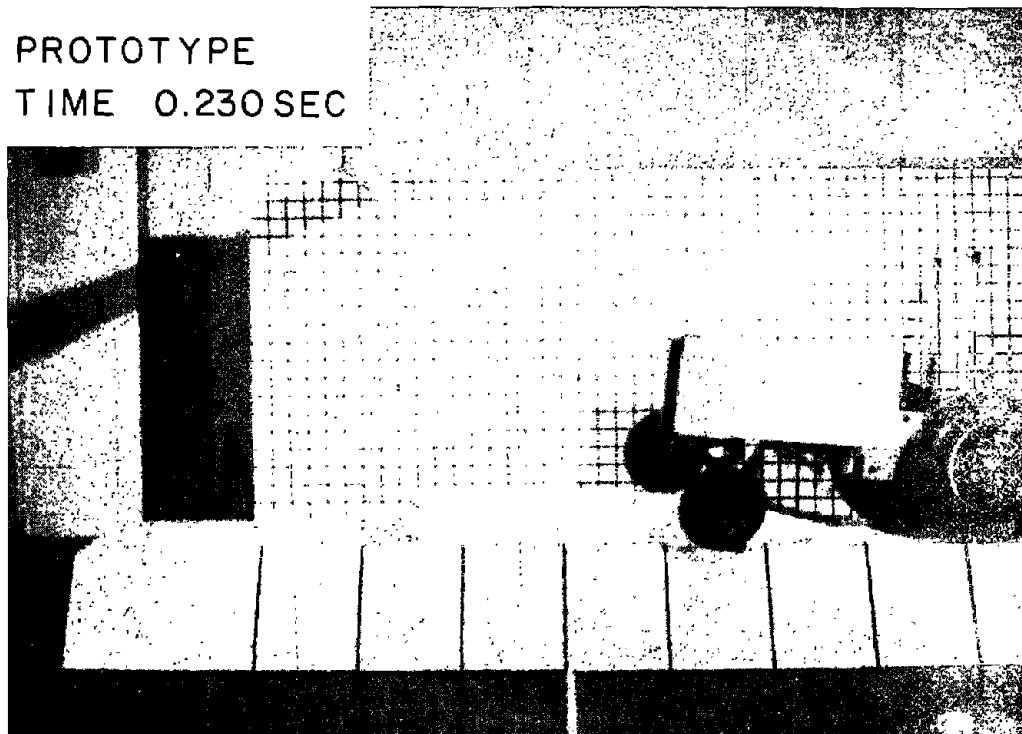
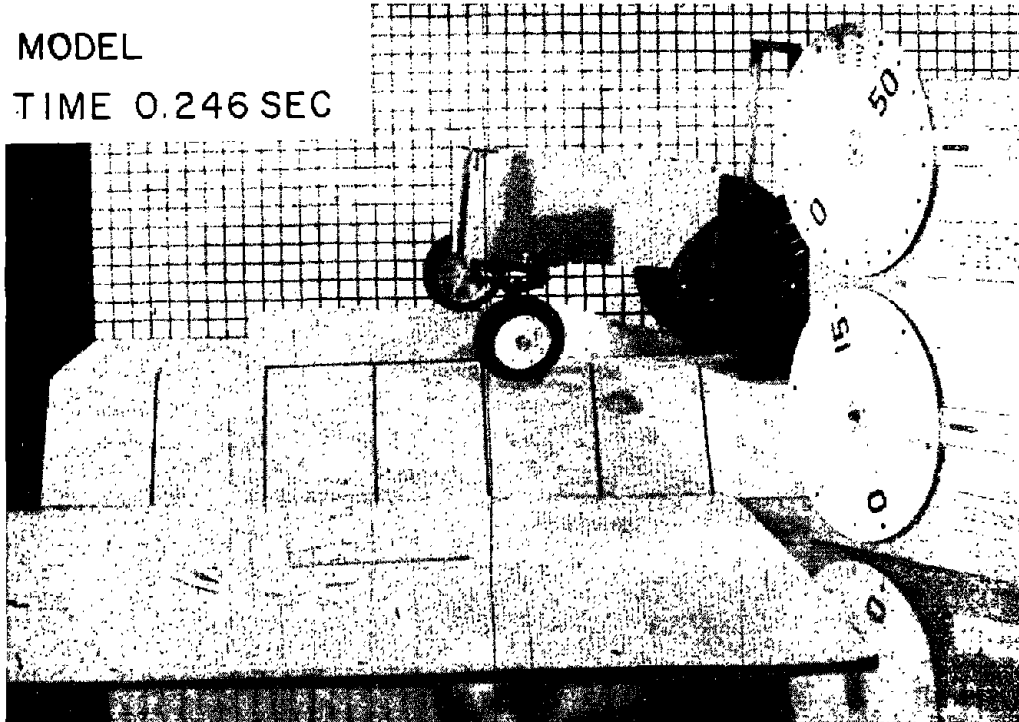


Figure 32. cont'd.

MODEL
TIME 0.246 SEC



PROTOTYPE
TIME 0.344 SEC

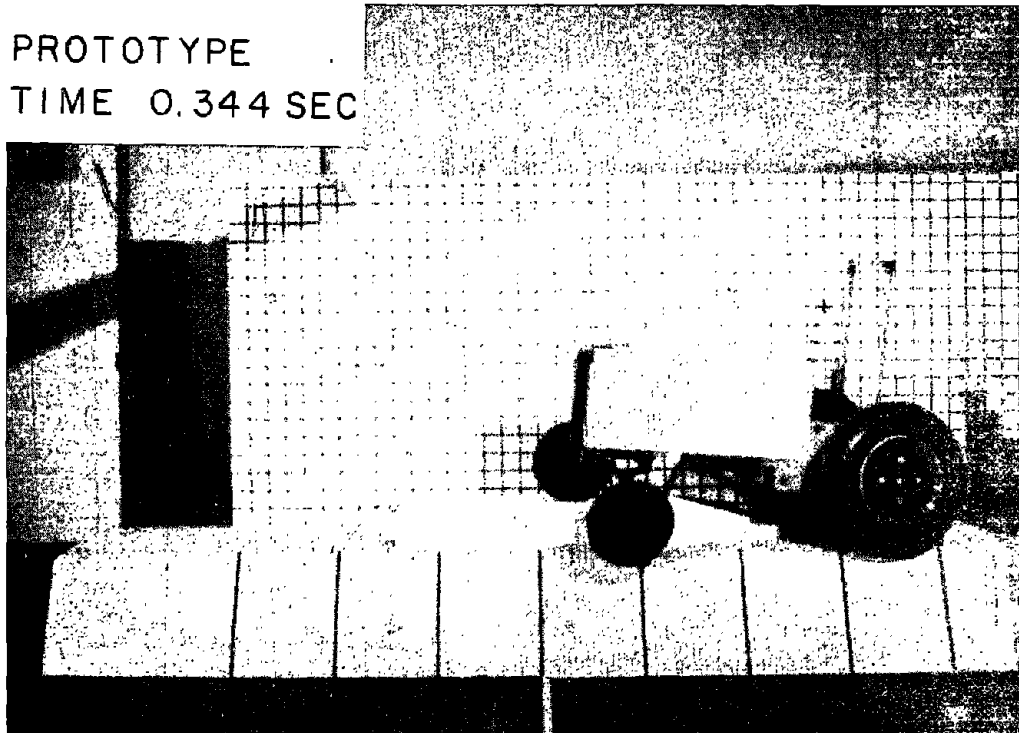
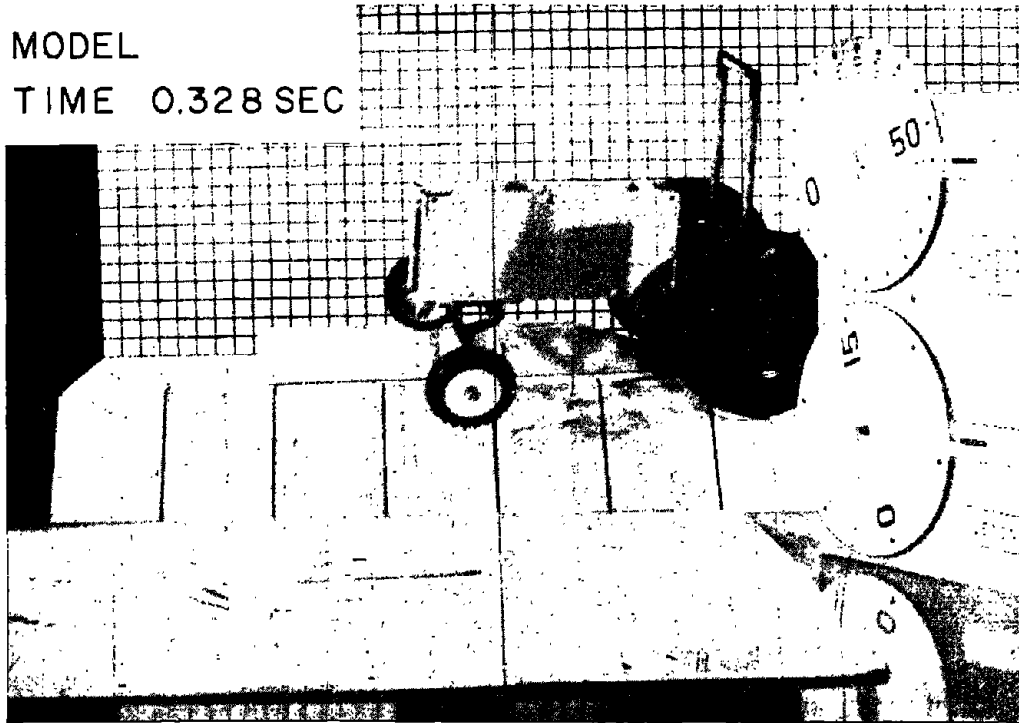


Figure 32. cont'd.

MODEL

TIME 0.328 SEC



PROTOTYPE

TIME 0.459 SEC

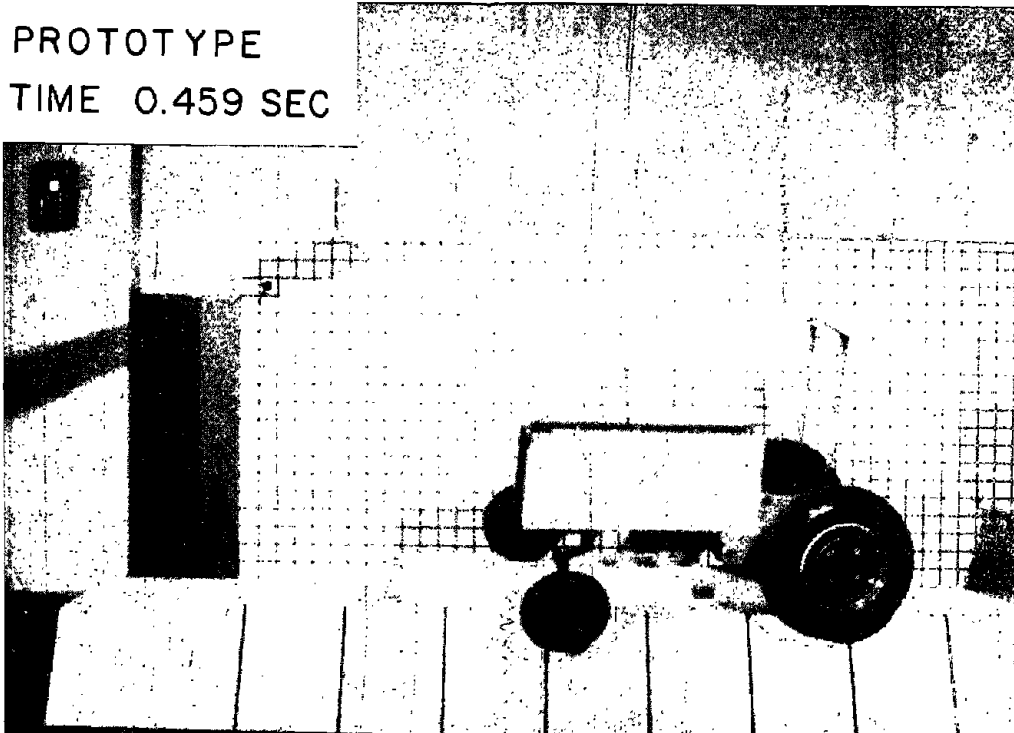
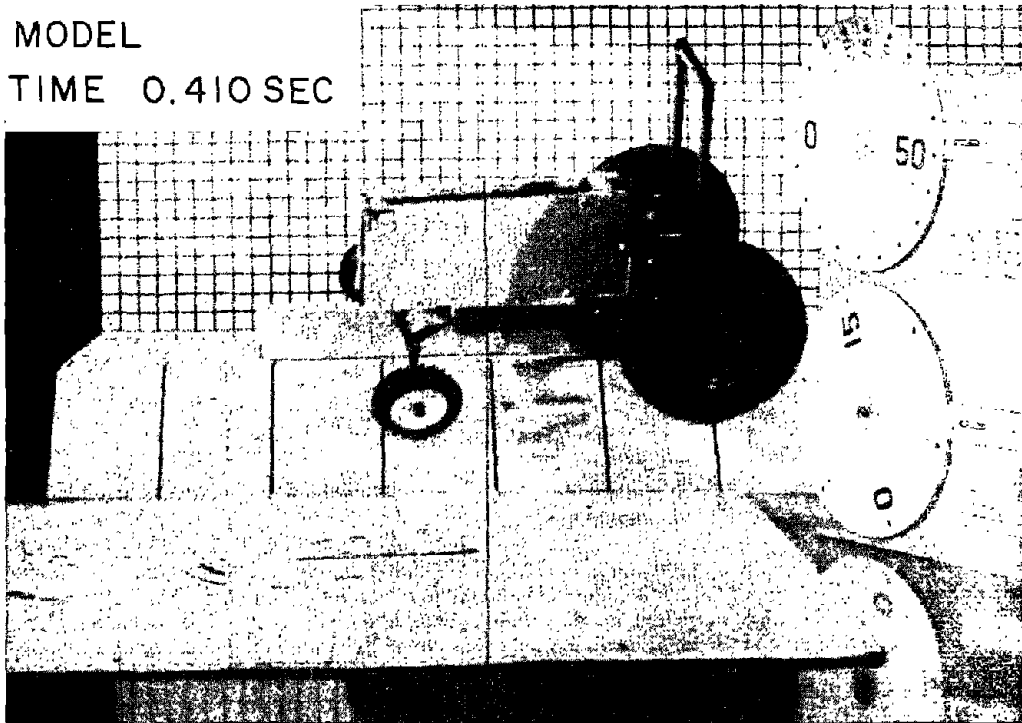


Figure 32. cont'd.

MODEL
TIME 0.410 SEC



PROTOTYPE
TIME 0.574 SEC

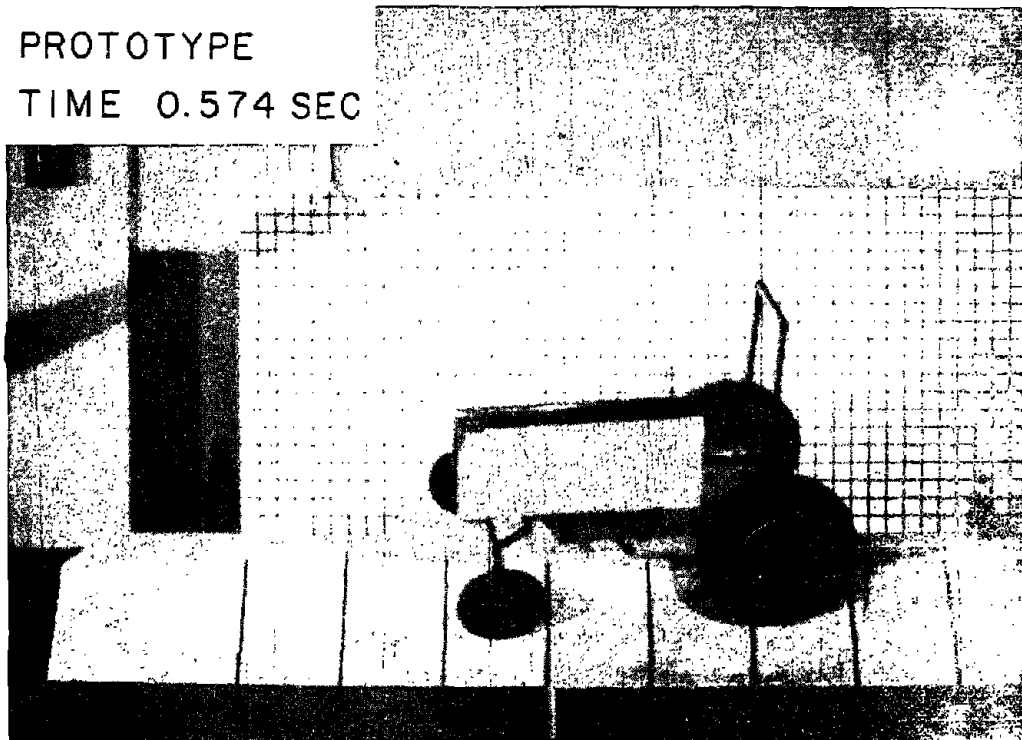
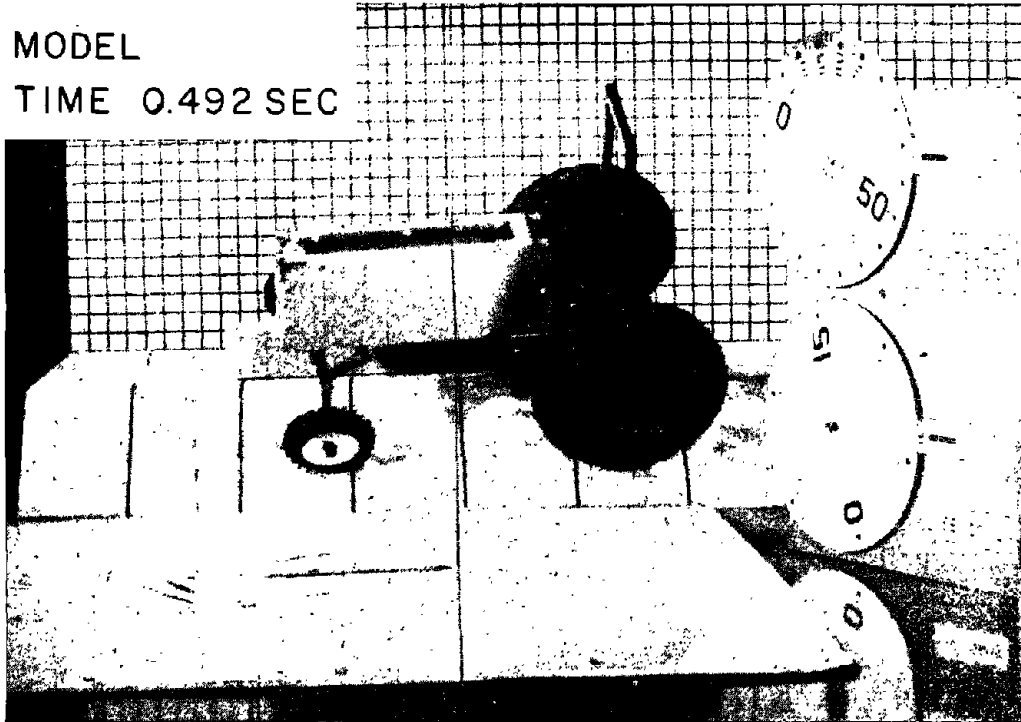


Figure 32. cont'd.

MODEL
TIME 0.492 SEC



PROTOTYPE
TIME 0.689 SEC

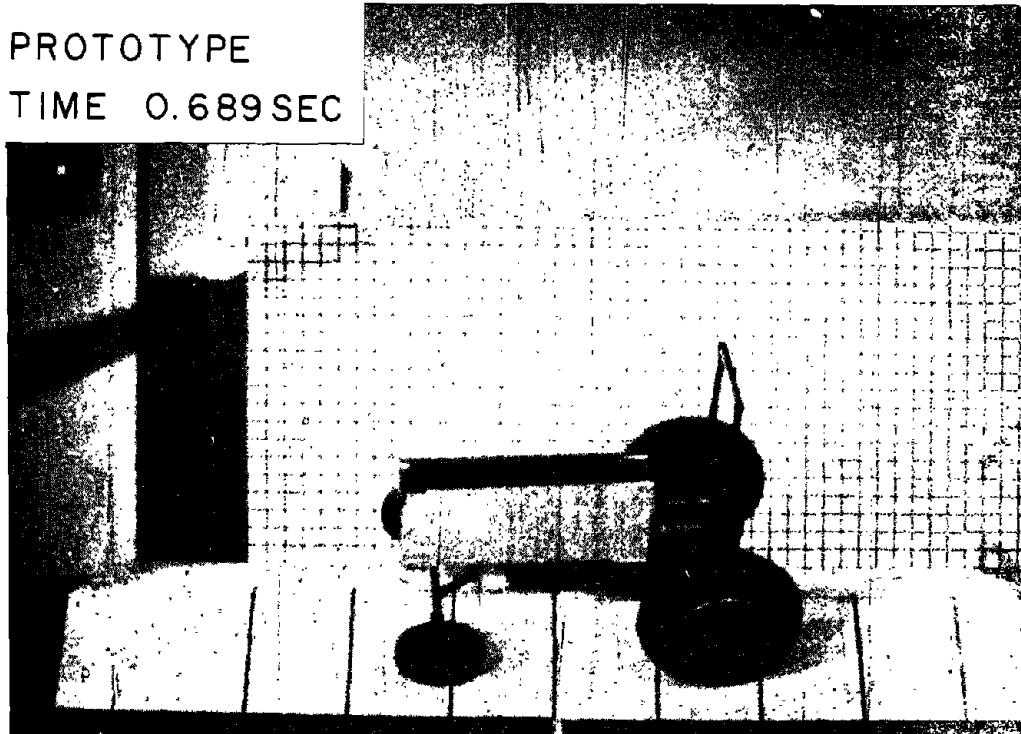
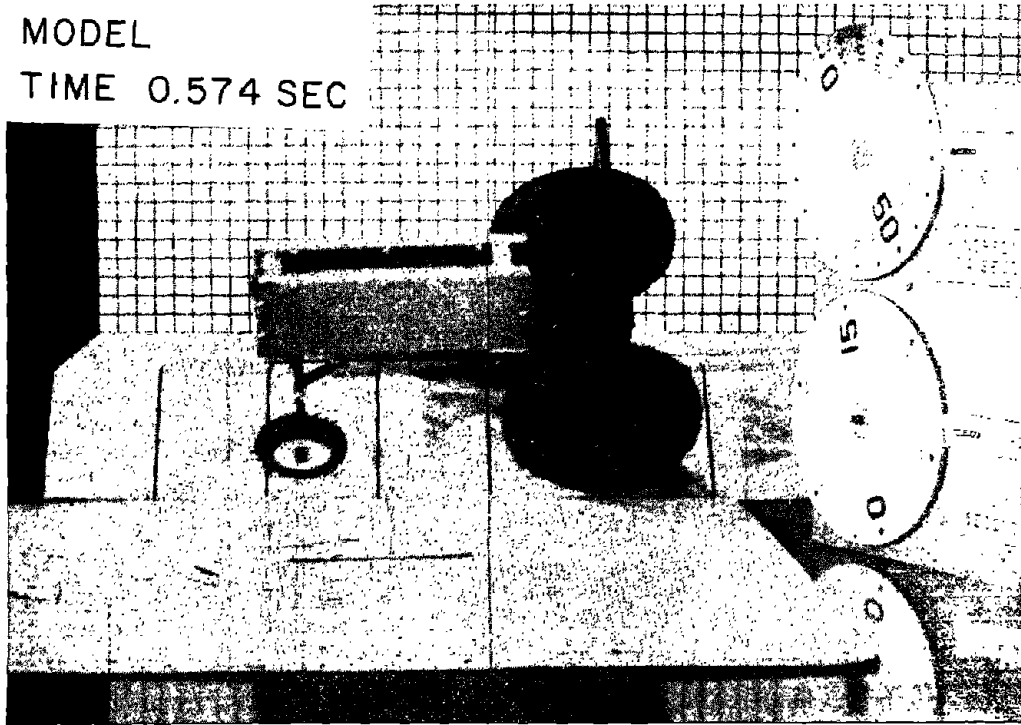


Figure 32. cont'd.

MODEL
TIME 0.574 SEC



PROTOTYPE
TIME 0.803 SEC

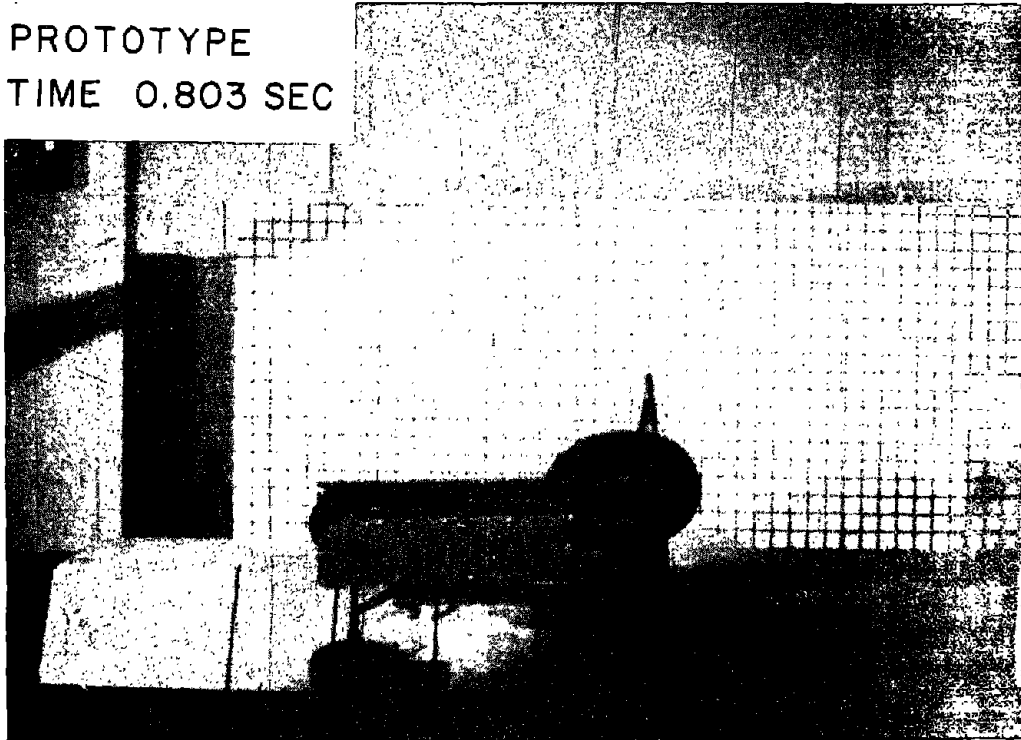
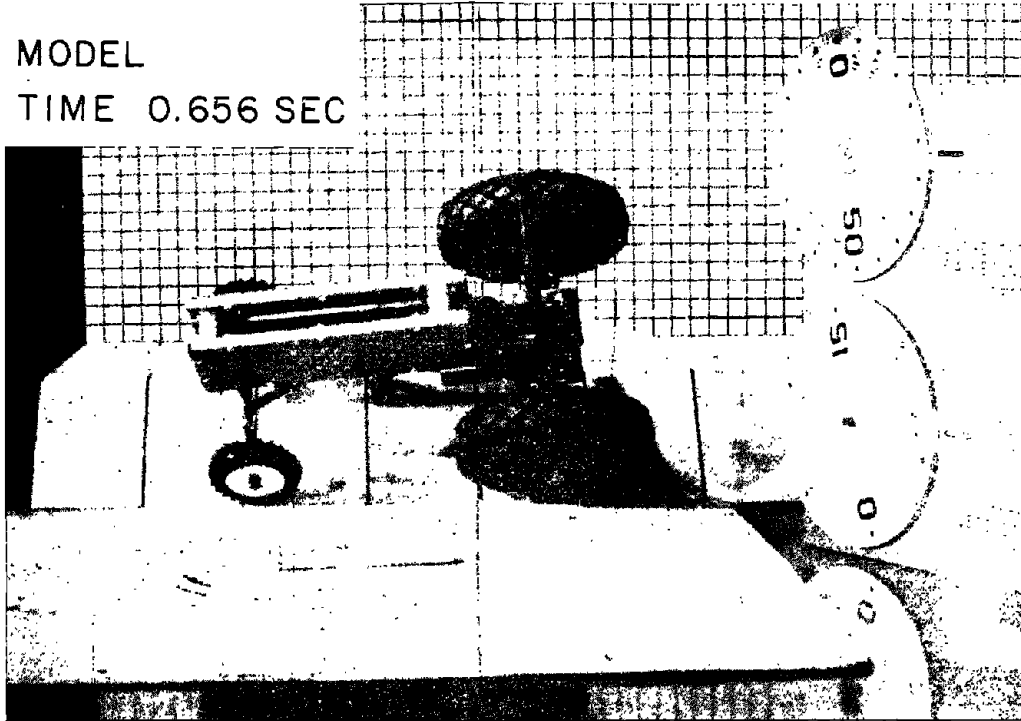


Figure 32. cont'd.

MODEL
TIME 0.656 SEC



PROTOTYPE
TIME 0.918 SEC

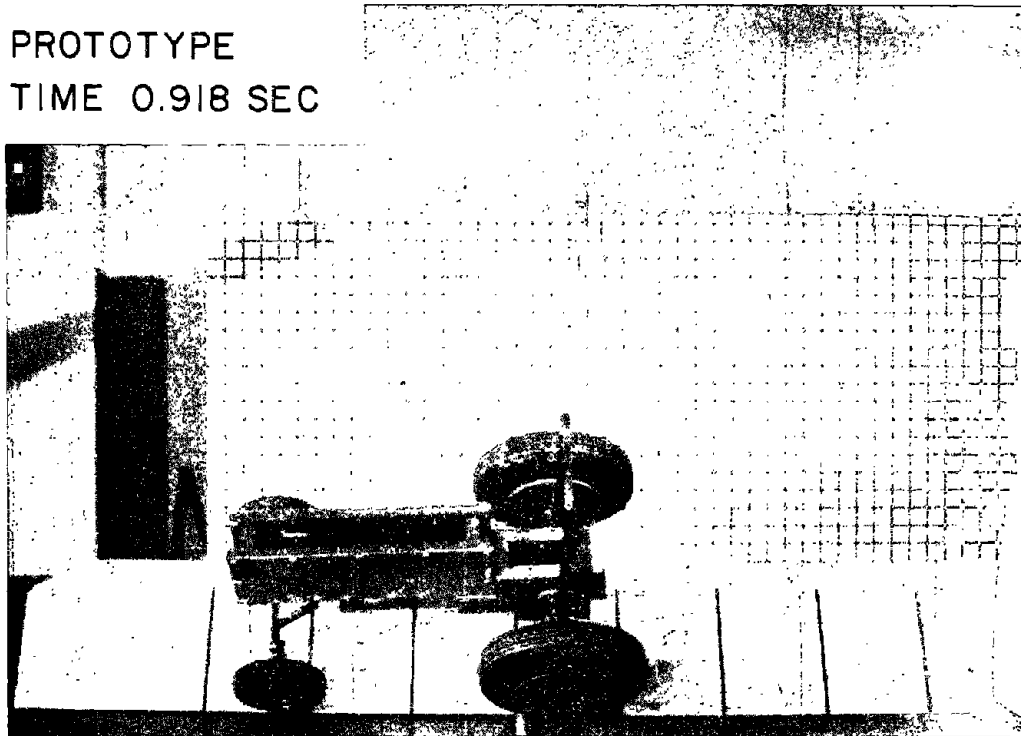
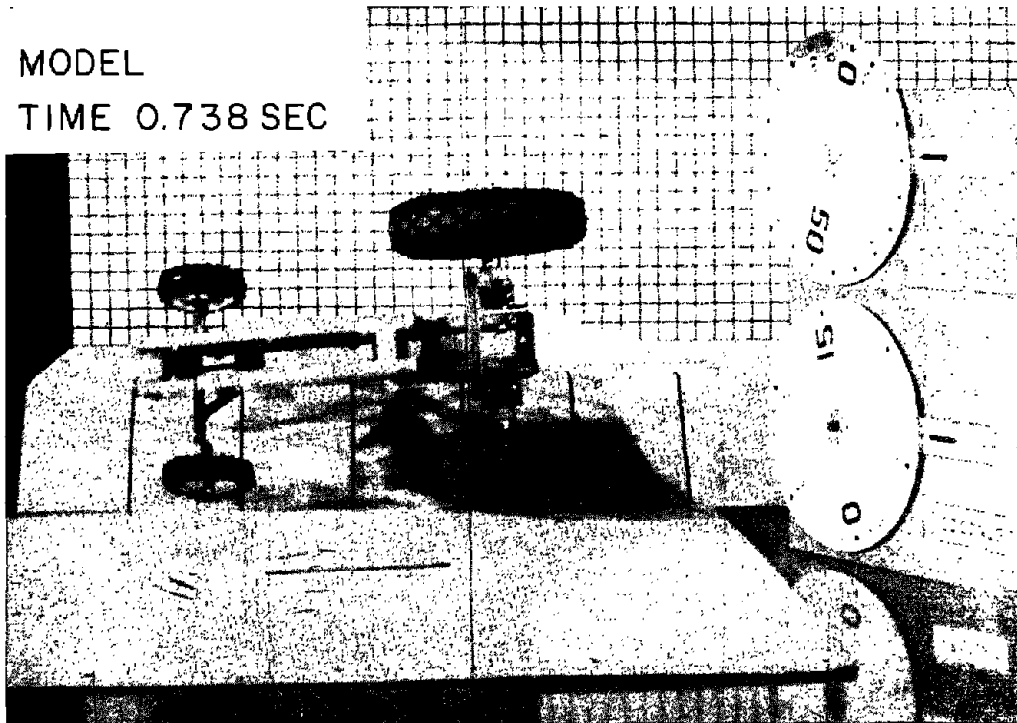


Figure 32. cont'd.

MODEL

TIME 0.738 SEC



PROTOTYPE

TIME 1.033 SEC

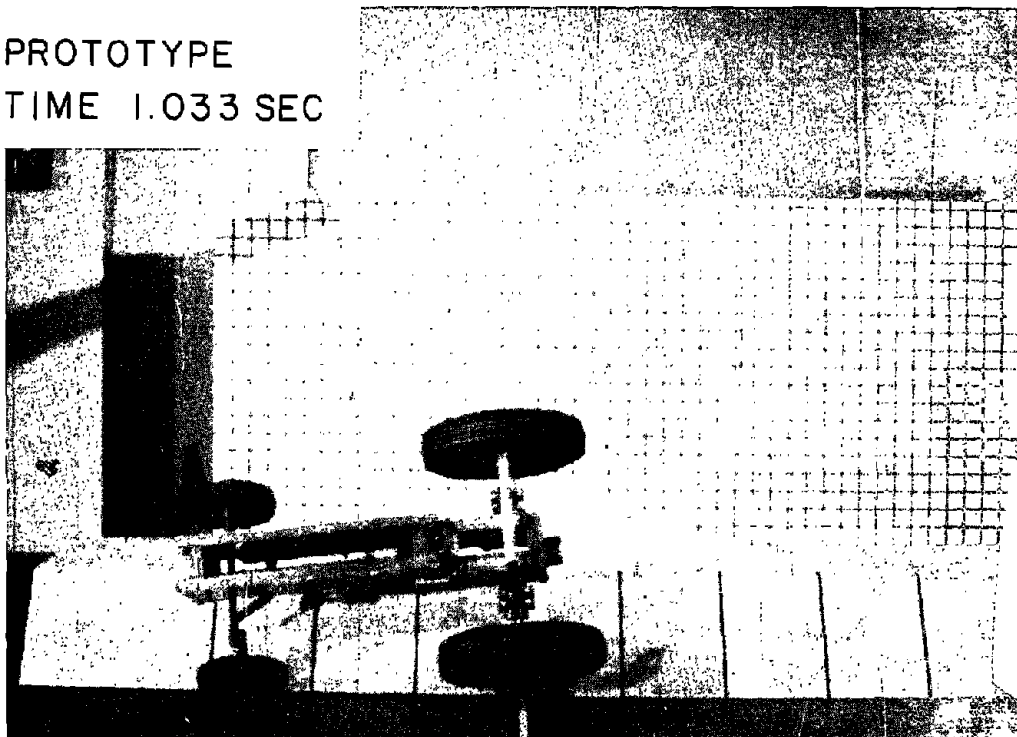
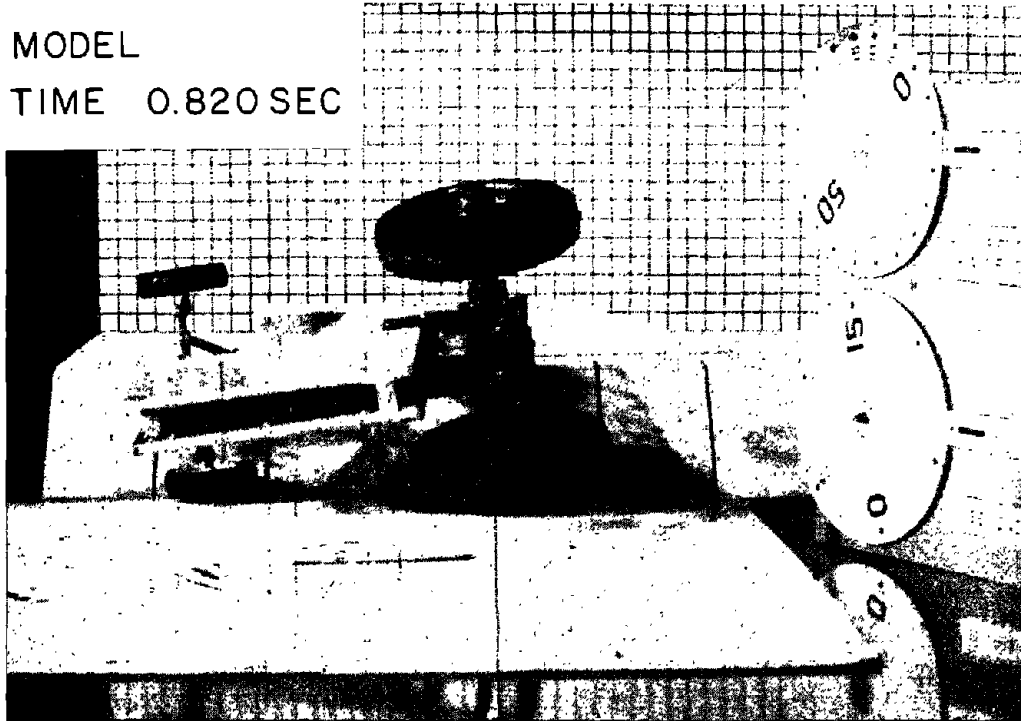


Figure 32. cont'd.

MODEL
TIME 0.820 SEC



PROTOTYPE
TIME 1.148 SEC

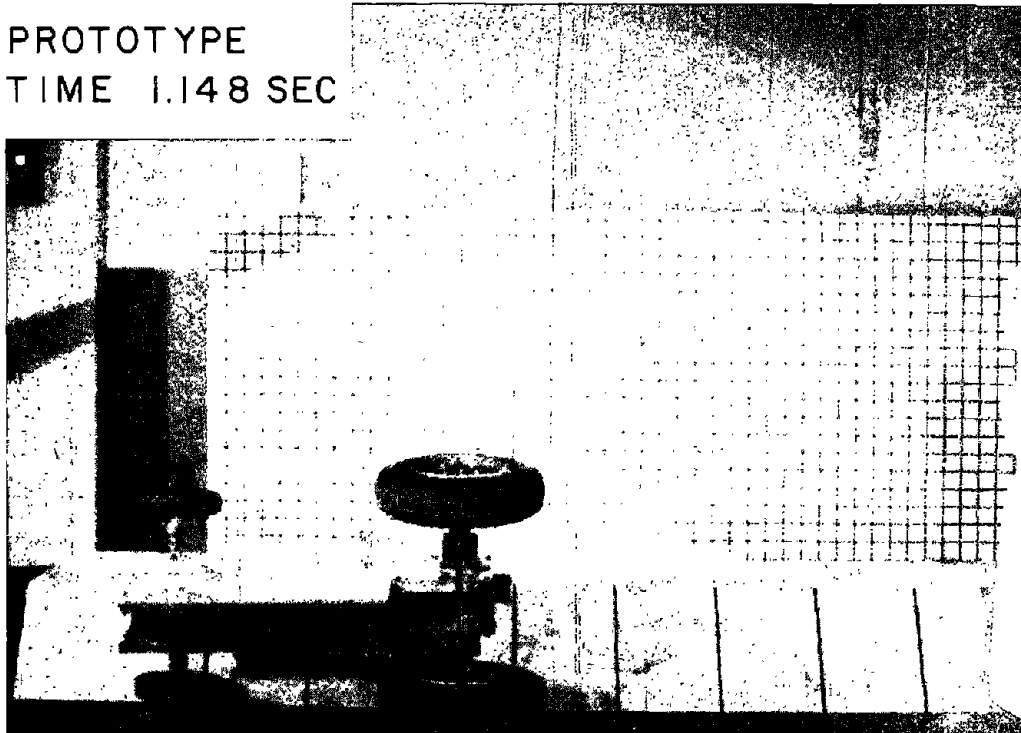
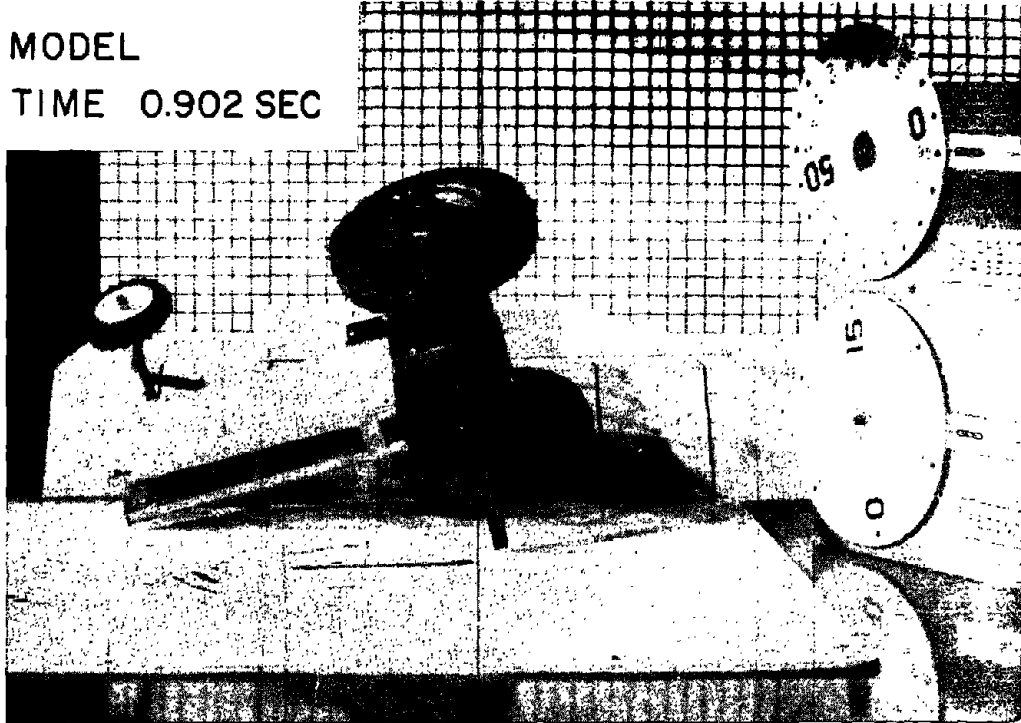


Figure 32. cont'd.

MODEL
TIME 0.902 SEC



PROTOTYPE
TIME 1.262 SEC

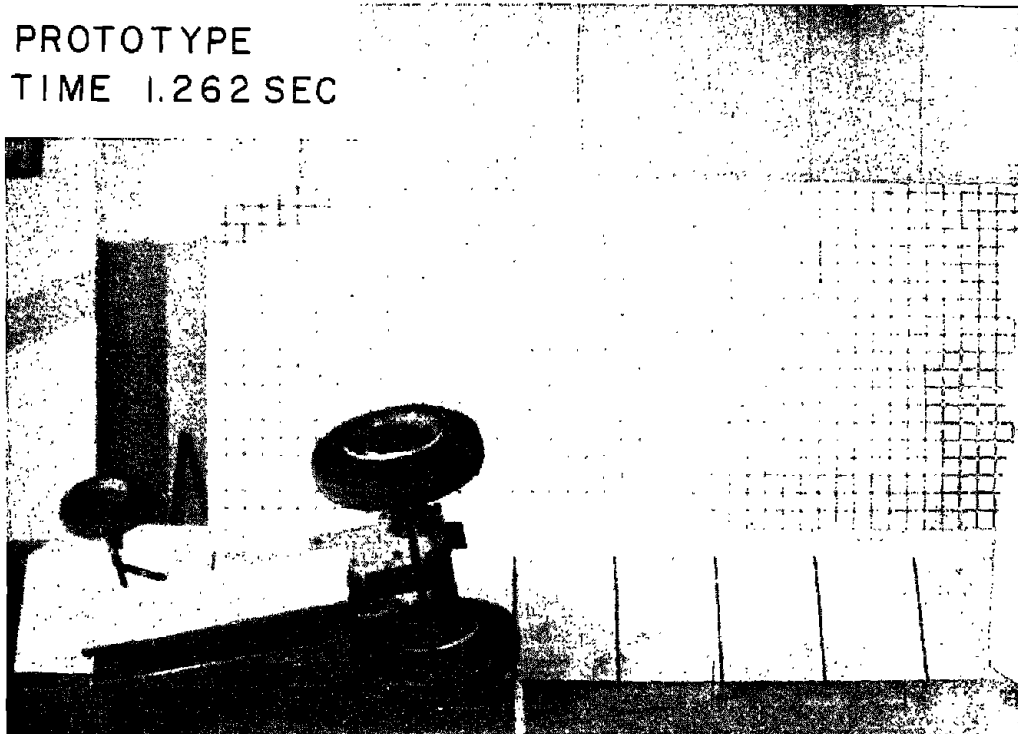


Figure 32. cont'd.

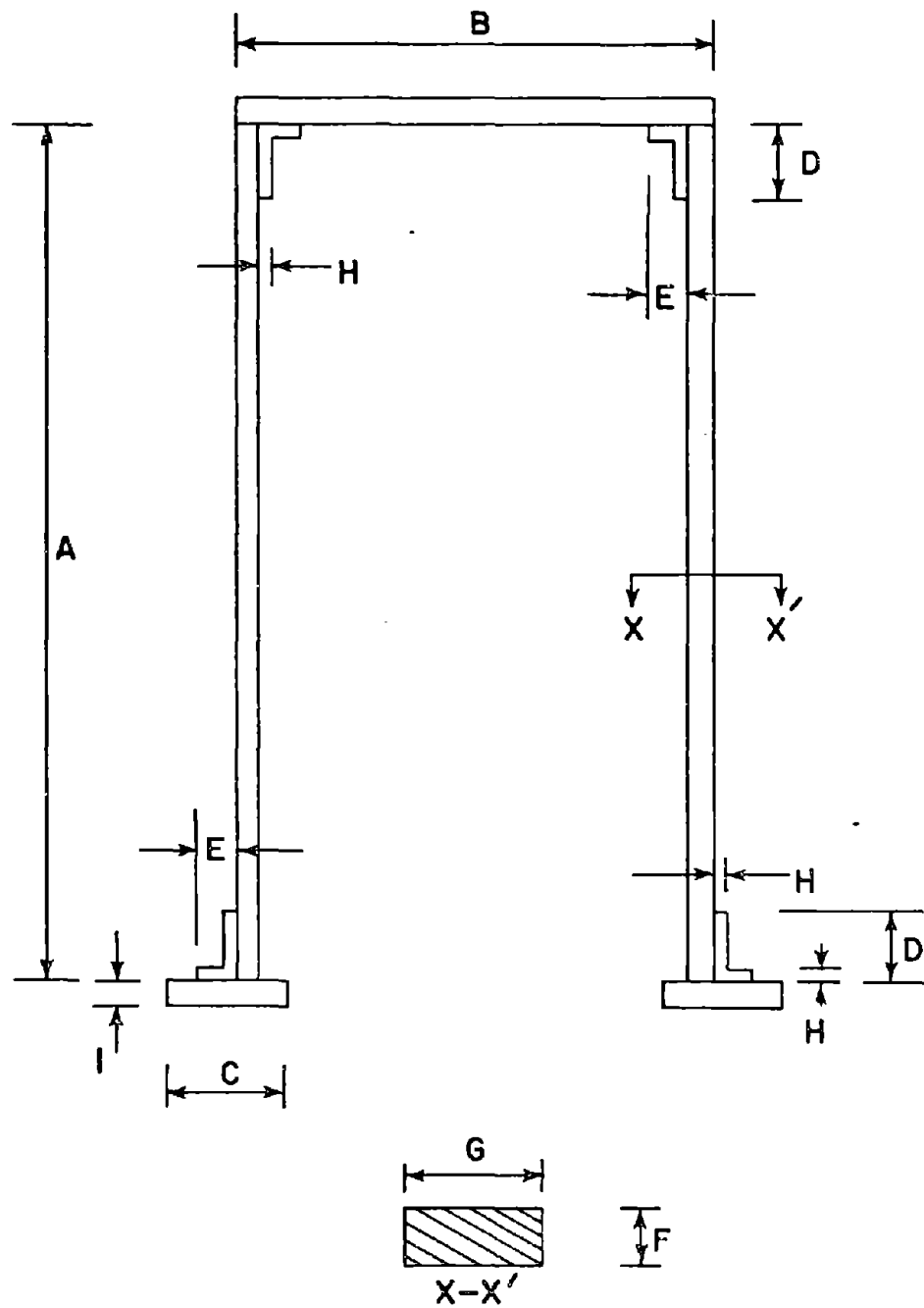


Figure 33. Generalized diagram of two-post type frame used in the model studies.

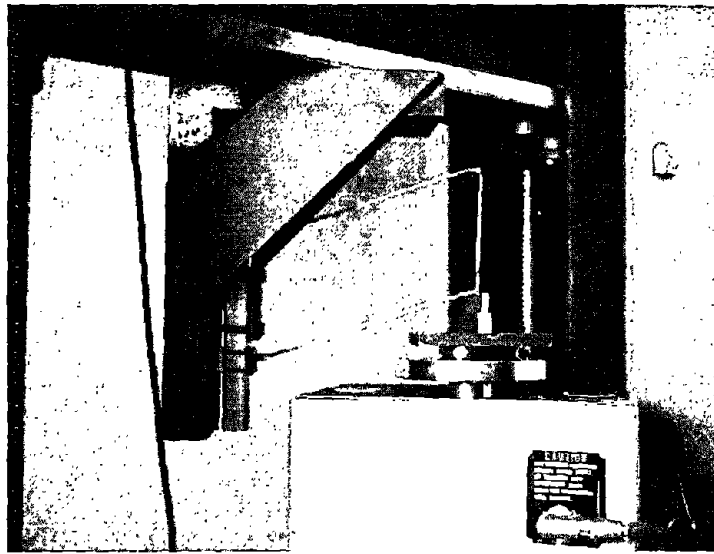


Figure 34. A two-post frame being tested in the Instron testing machine.

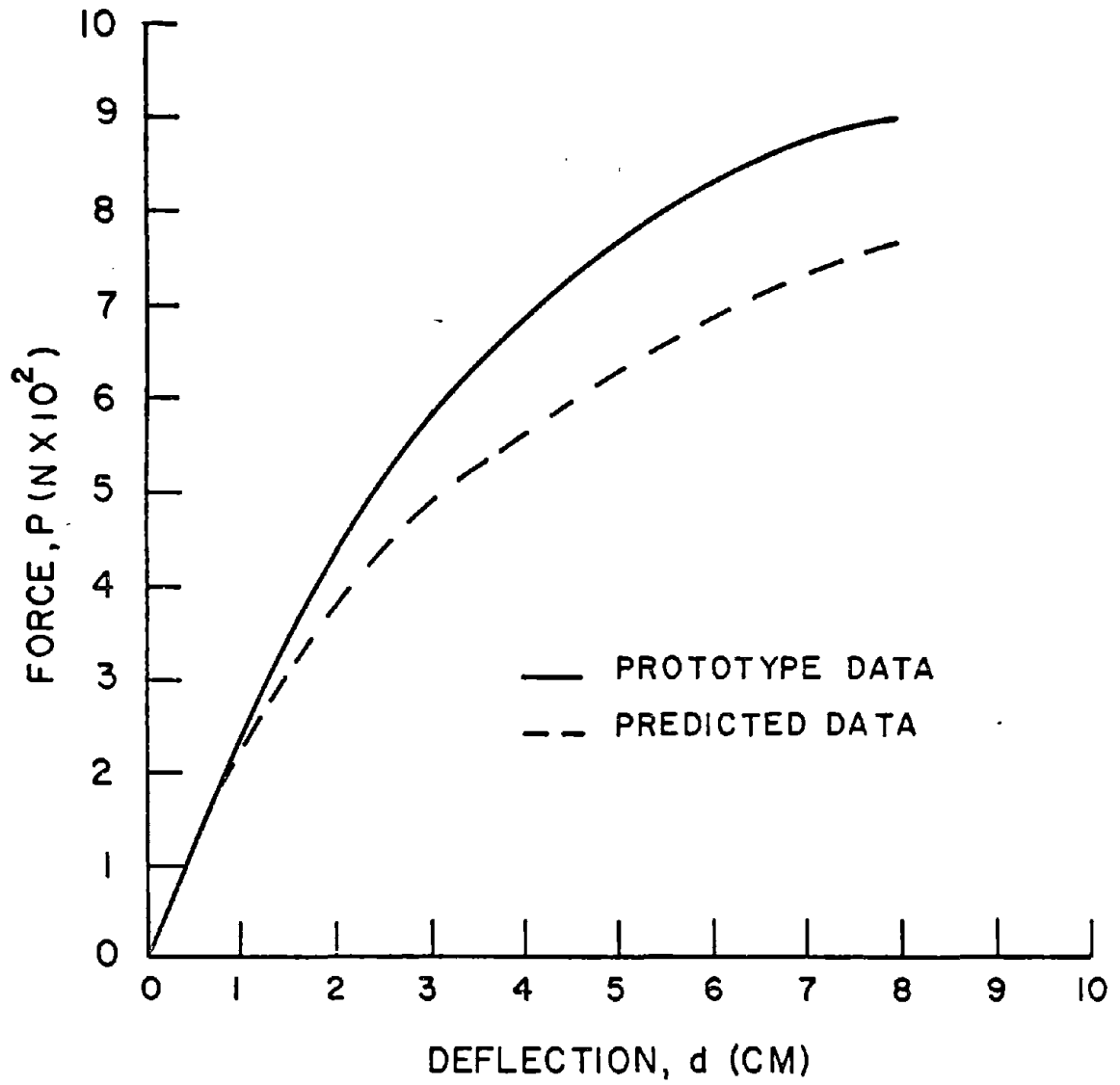


Figure 35. Static test results of the two-post type bolted frame.

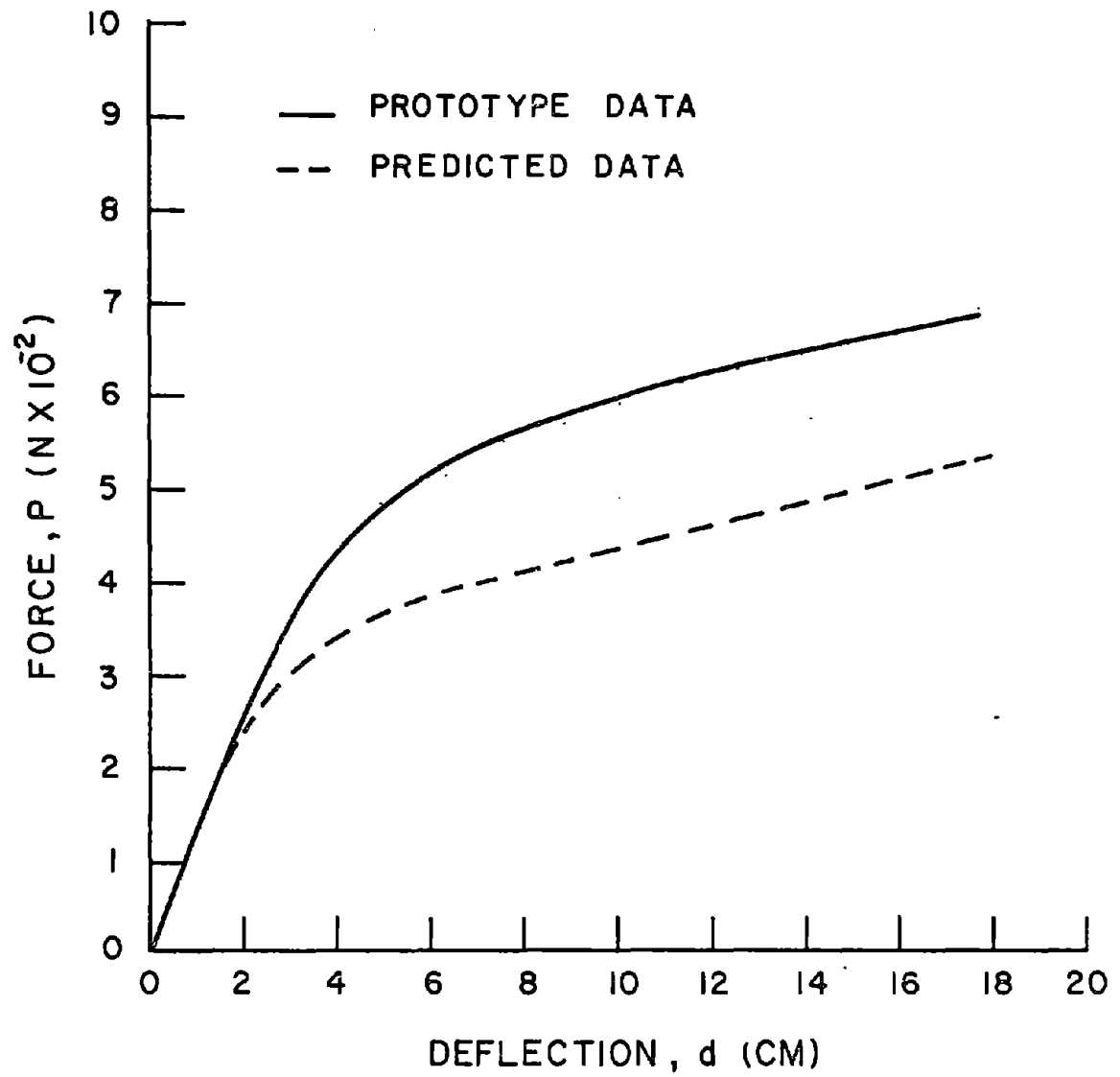


Figure 36. Static test results of the two-post type welded frames.

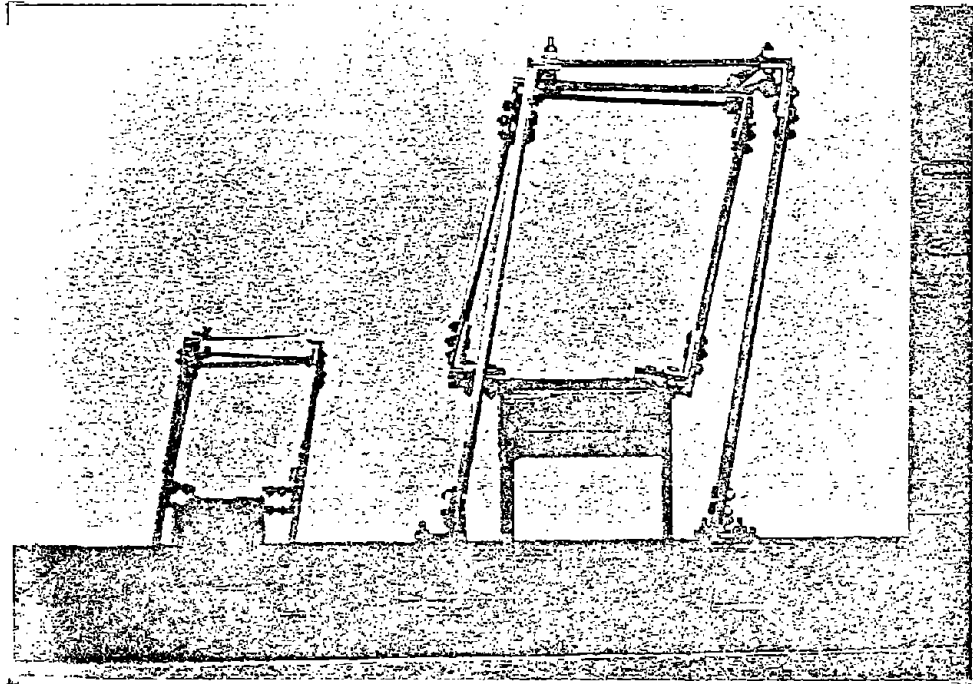


Figure 37. Typically bent 1/4-scale and 1/8-scale four-post type model ROPS after static testing.

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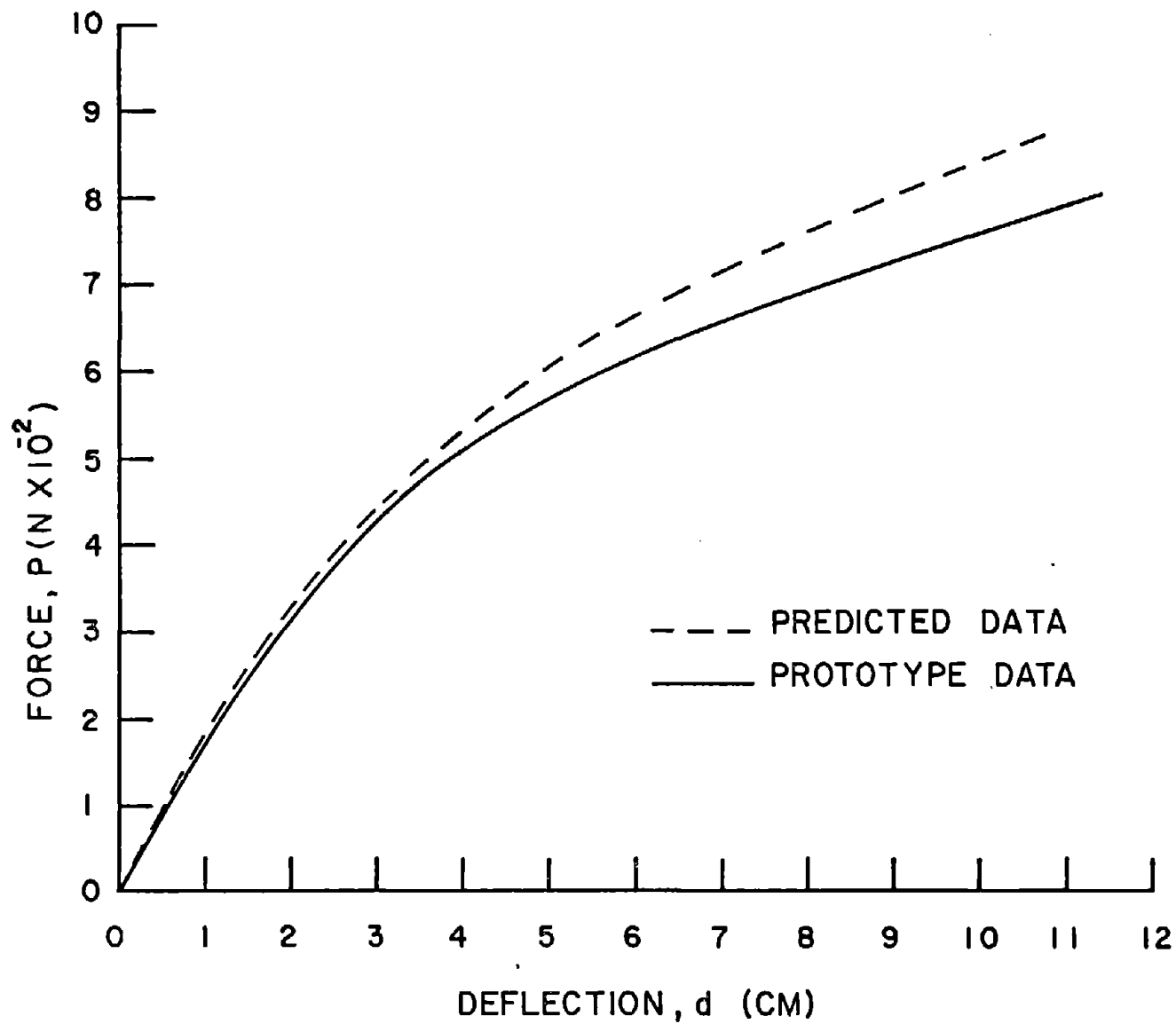


Figure 38. Static test results of four-post type model ROPS.

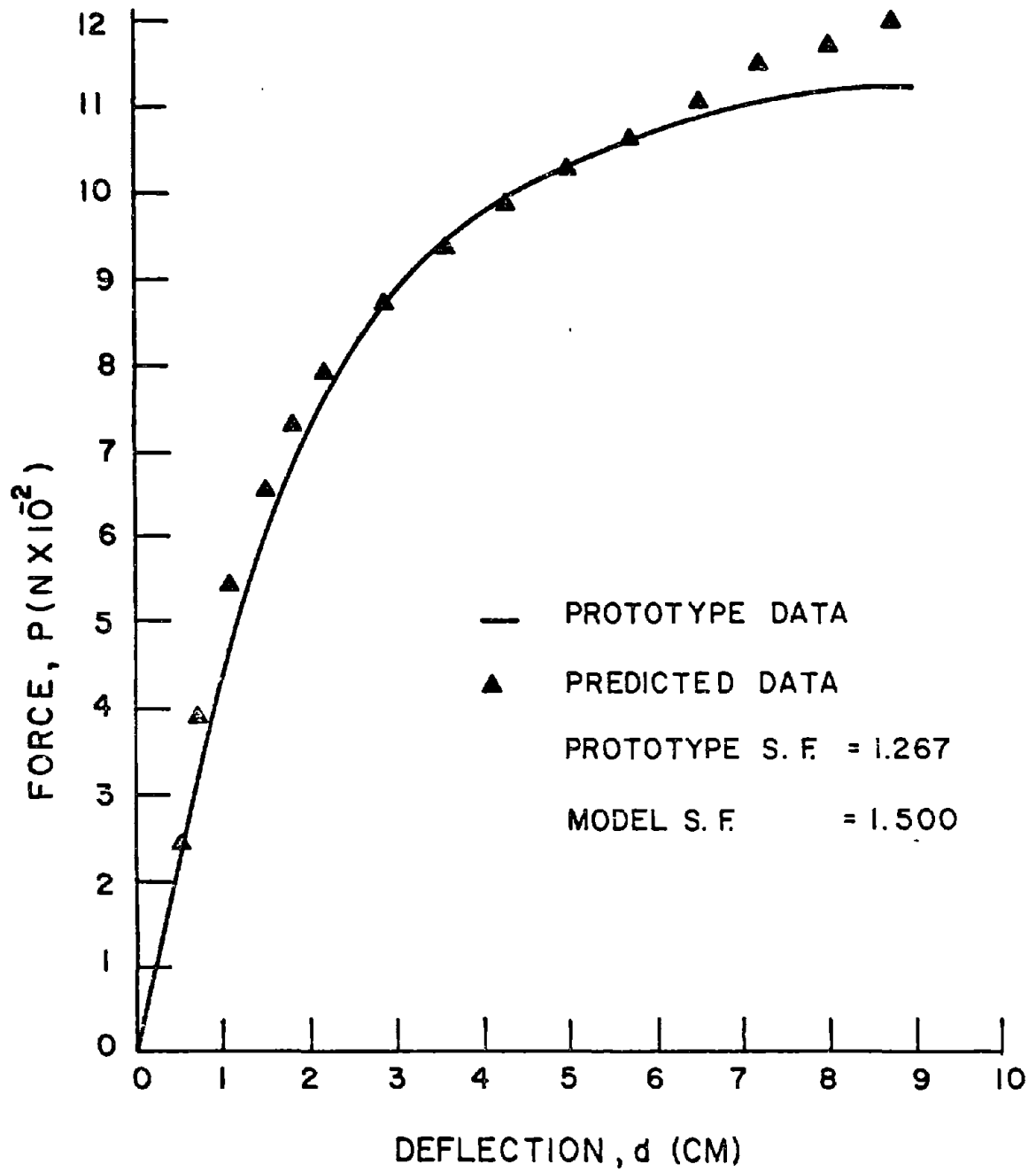


Figure 39. Static test results of dissimilar cross-section model frames.

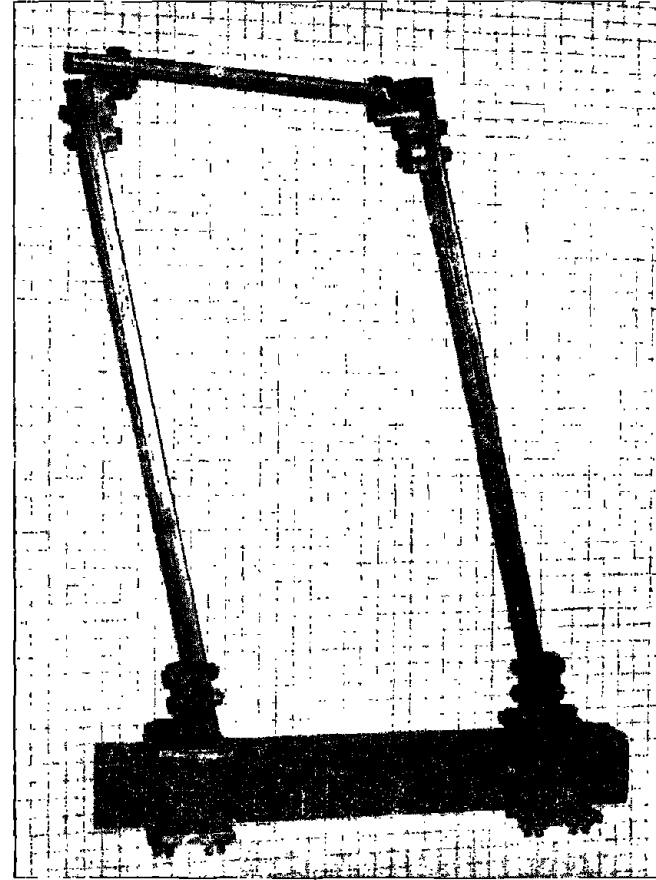
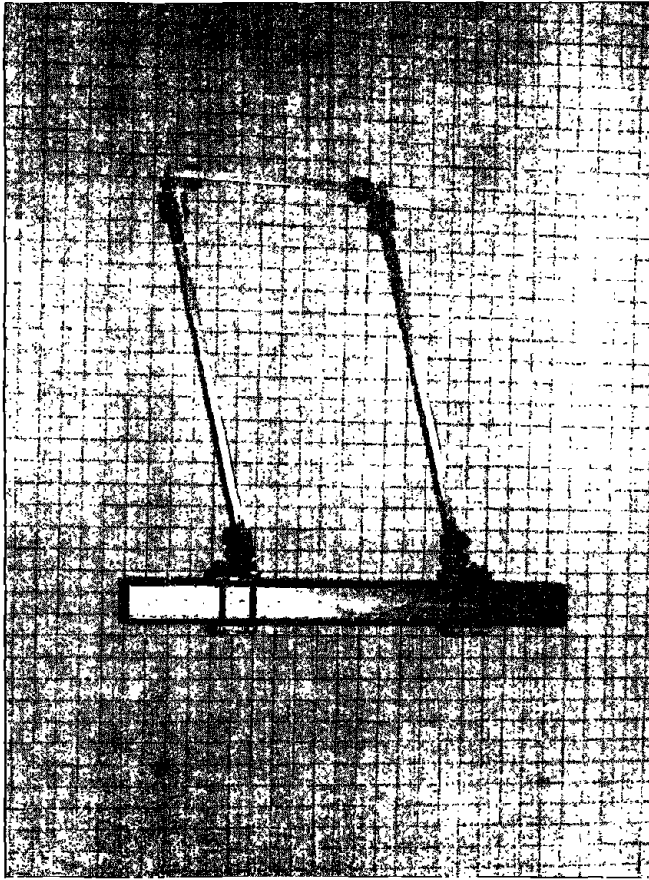


Figure 40. Photograph showing the similarity in deformation between the solid-section model '(a)' and the hollow-section prototype '(b)' frames.



Figure 41. Commercially manufactured two-post full scale ROPS after static test (courtesy International Harvester Company).

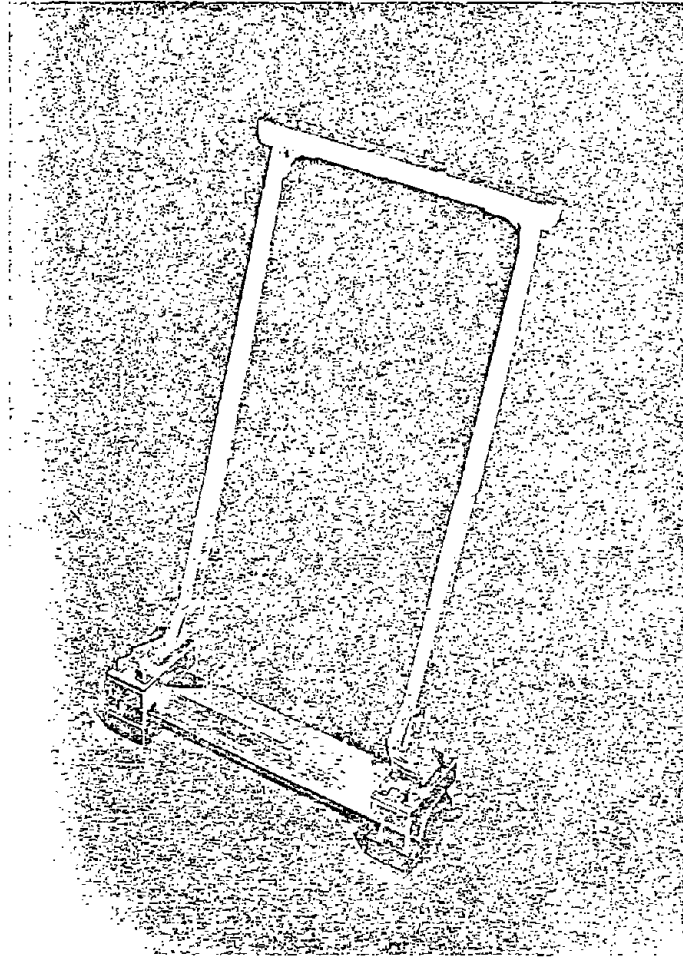


Figure 42. Quarter scale model of the IH-frame.

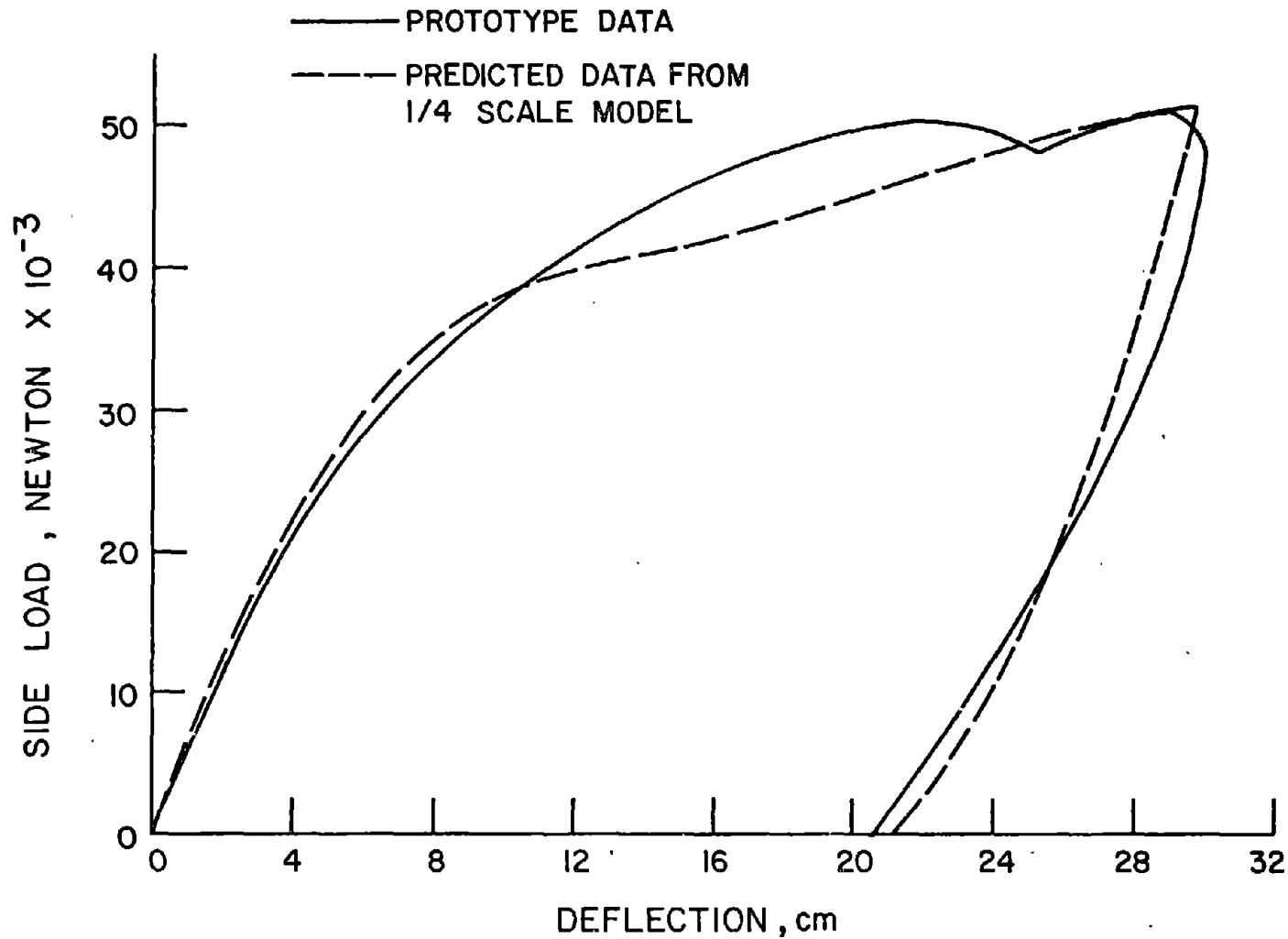


Figure 43. Similarity in the predicted load-deflection data using a 1/4-scale model frame and the static test data of a commercially manufactured full size ROPS provided by the International Harvester Company.

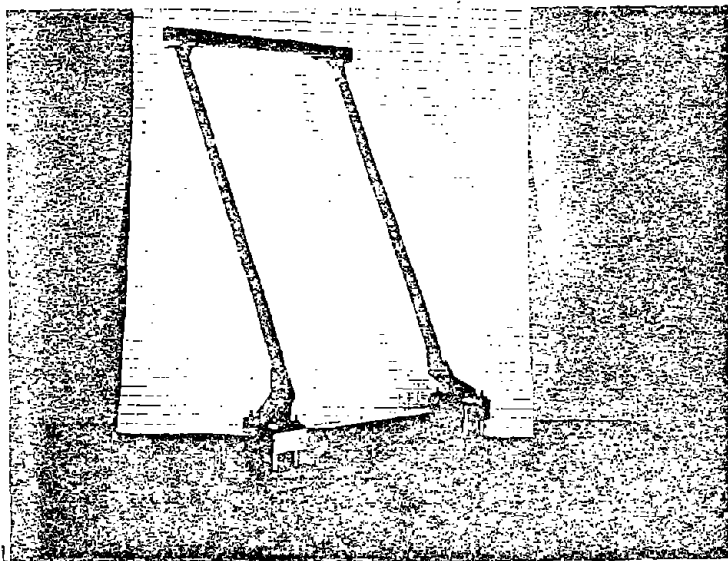
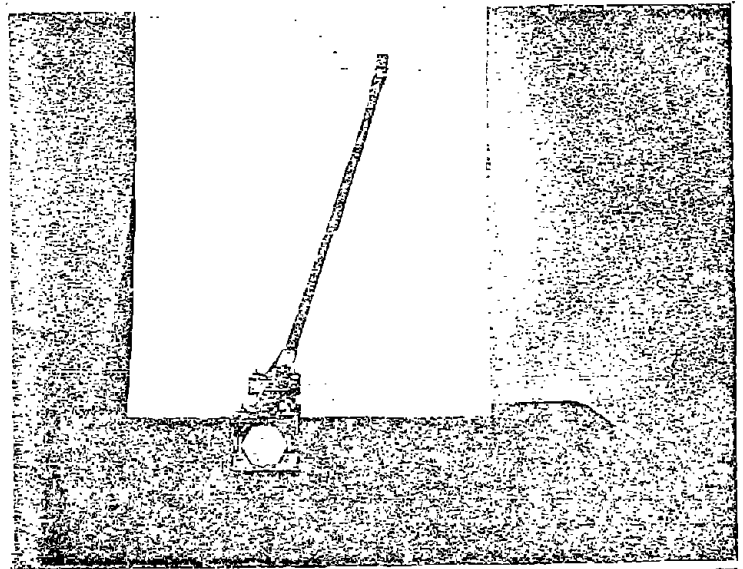
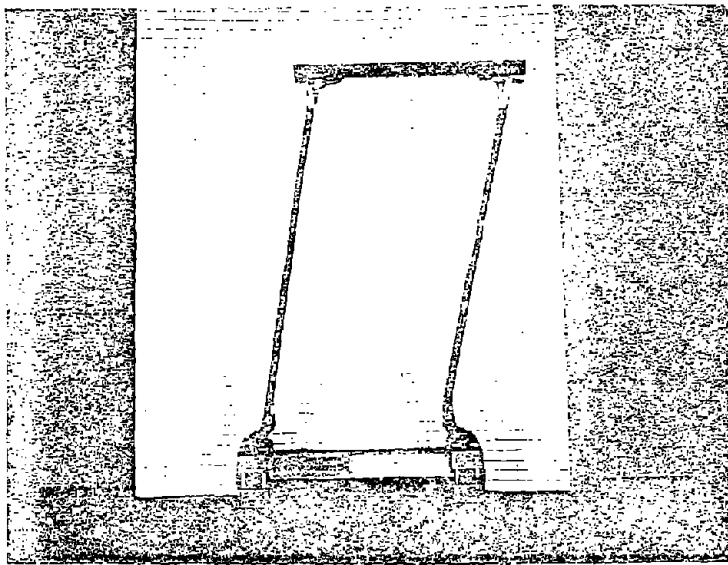


Figure Figure 44. The quarter scale frame after static testing.

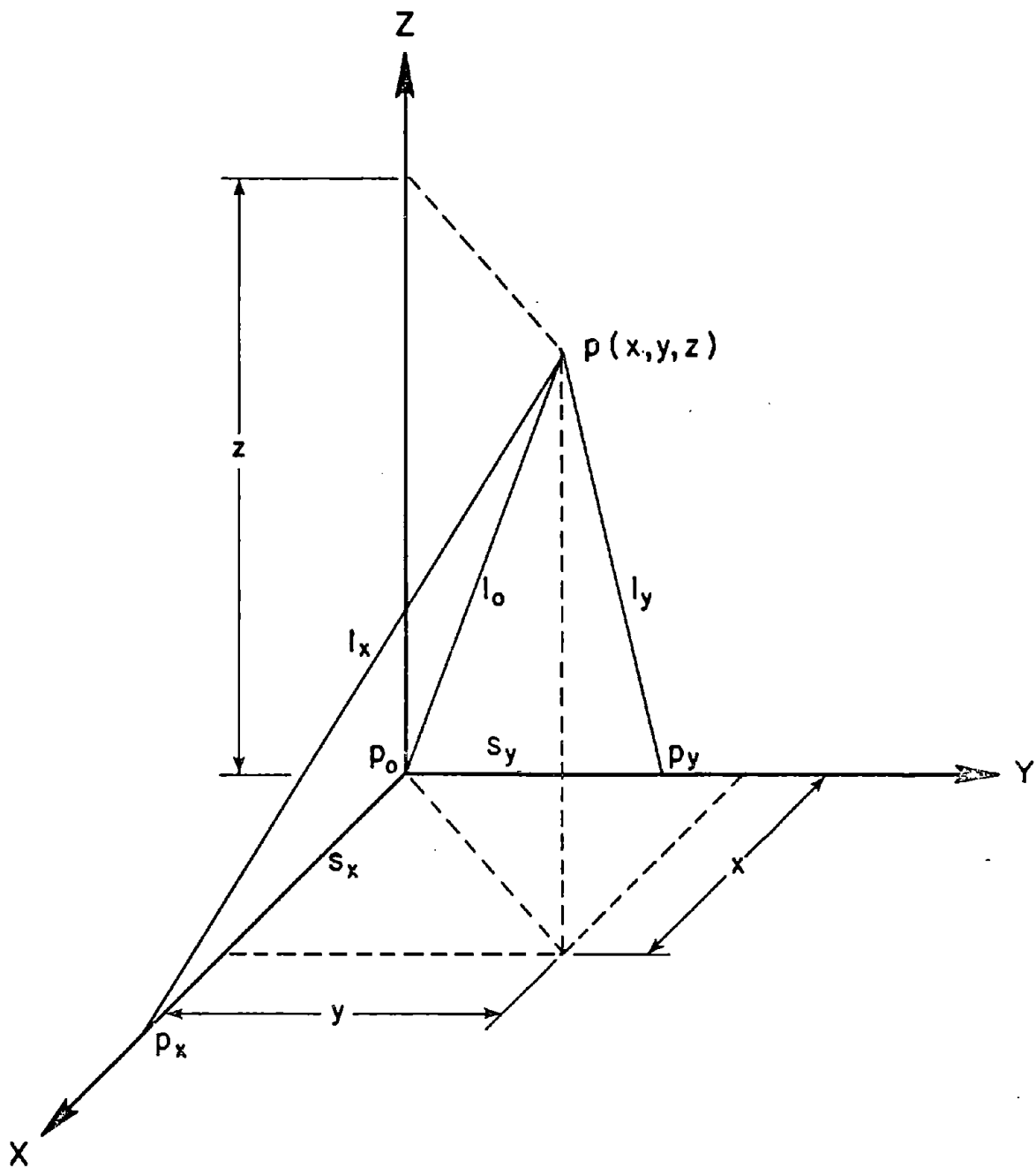


Figure 45. Coordinate system used to illustrate the principle of the 3-dimensional frame deflection measuring transducer.

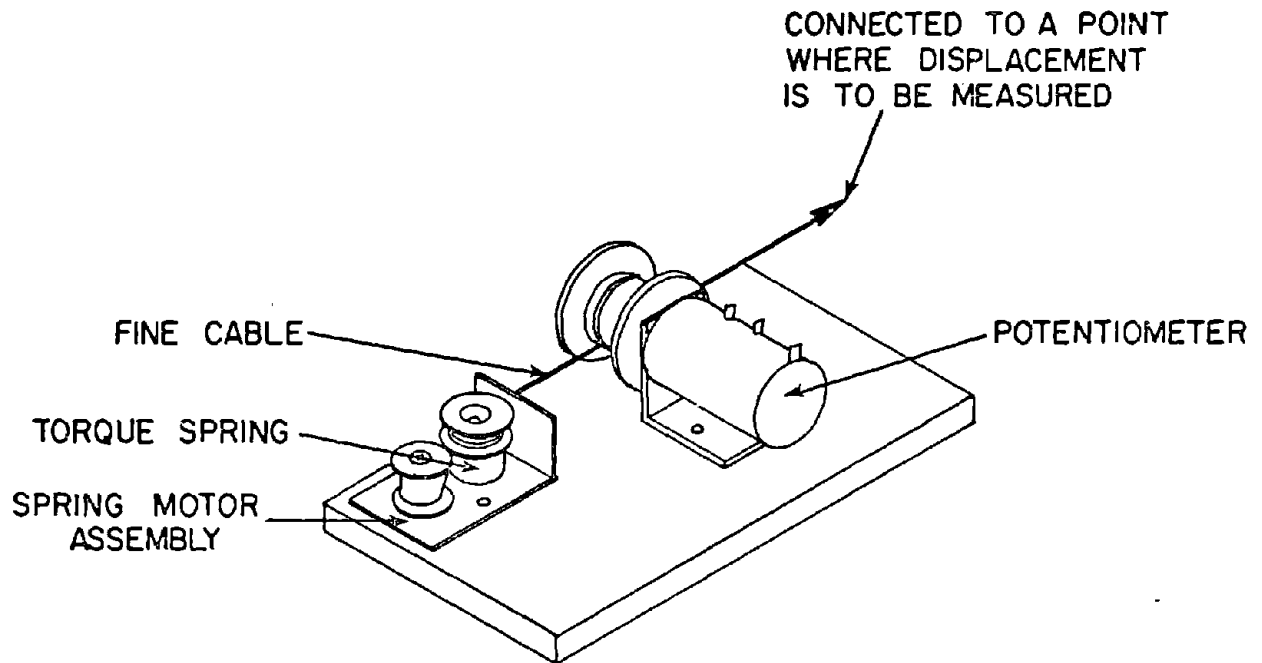


Figure 46. Constant torque motor and potentiometer unit as used to measure change in each line segment.

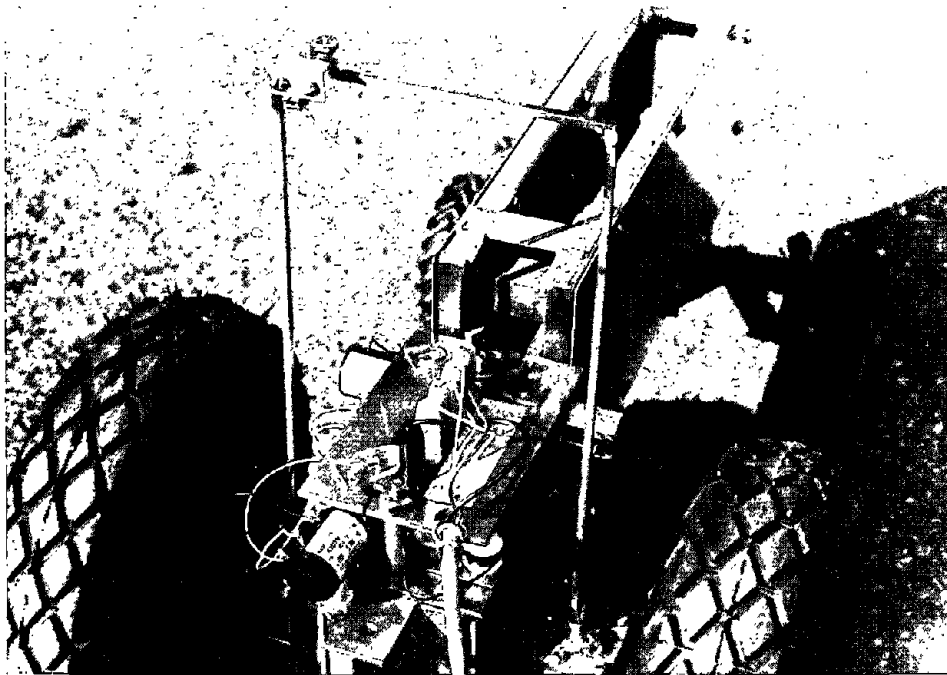


Figure 47. The three-dimensional deflection measuring transducer.

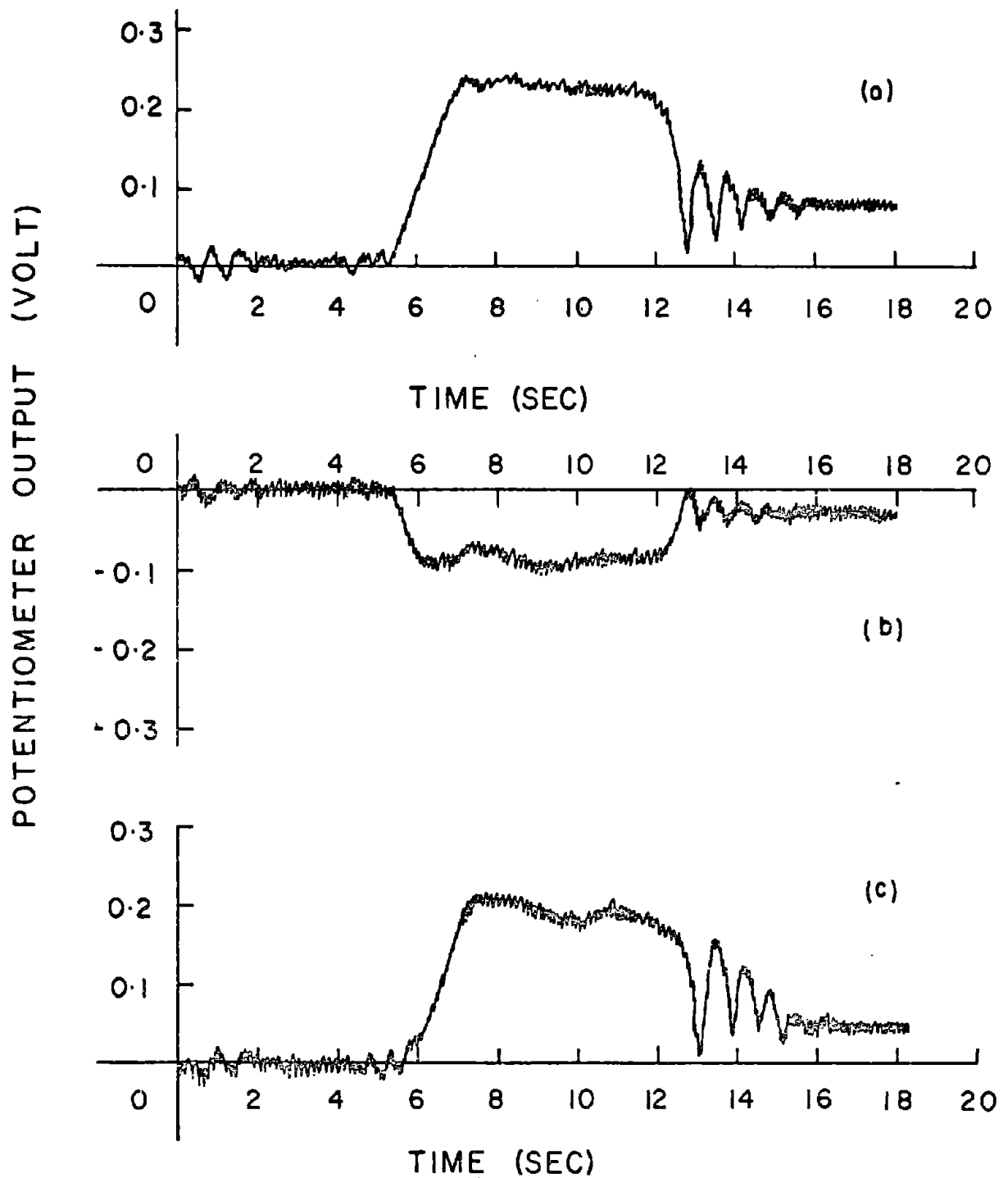


Figure 48. An example of the recorded output of the three-dimensional displacement transducer during an overturn test.

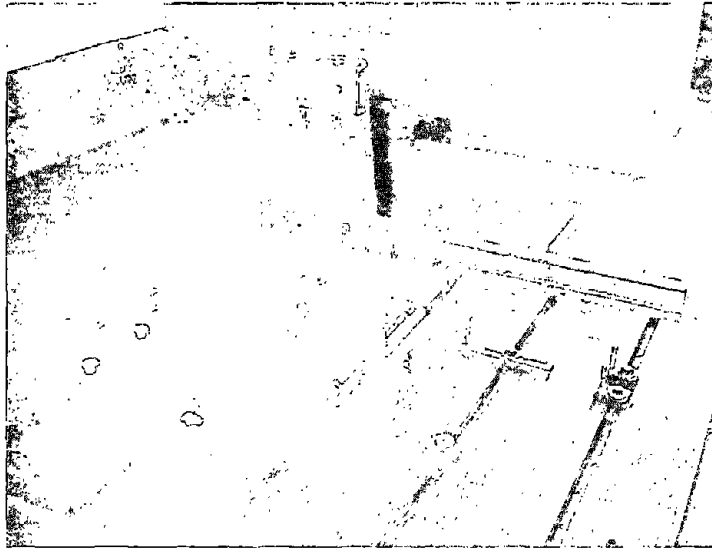


Figure 49a. Fixtures to measure final frame deformation after an overturn test.

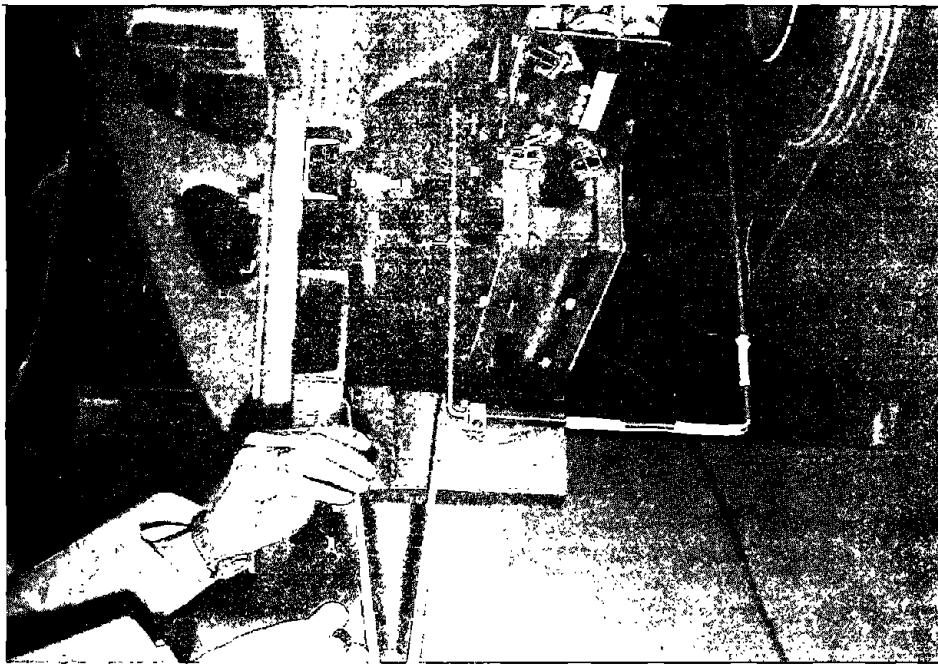


Figure 49b. Measurement is being made to determine frame deformation of the 1/4-scale frame.

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