To the Graduate Council:

I am submitting herewith a thesis written by Robert Stephen Comer entitled "Evaluation of Engineering Plastic for Rollover Protective Structure Mounting." I have examined the final paper copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Biosystems Engineering.

Paul D. Ayers, Major Professor

We have read this thesis and recommend its acceptance:

Acceptance for the Council:

Vice Chancellor and Dean of Graduate Studies

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EVALUATION OF ENGINEERING PLASTIC FOR ROLLOVER PROTECTIVE STRUCTURE MOUNTING

A Thesis
Presented For the
Master of Science
Degree
The University of Tennessee, Knoxville

Robert Stephen Comer
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DEDICATION

This thesis is dedicated to my parents, Robert A. Comer and Theresa Comer, and family for their support and belief in me, and to my fiancée, Melissa Shelton, for her daily inspiration and encouragement as I tried to achieve my goals throughout these past years.

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ABSTRACT

The agricultural industry has one of the highest fatality rates of any industry in the U. S. This death rate is largely due to deaths caused by roll over accidents while on agricultural vehicles. The addition of a Rollover Protective Structure (ROPS) has aided in lowering these high fatality rates on full-size tractors. Studies have shown that death rates have been reduced by as much as 98% with the addition of rollover protective structures. However, a large number of older tractors still do not utilize a ROPS. This is due to the difficulty of designing and creating a mounting structure. To aid in elimination of this difficulty, an evaluation engineering plastics to aid mounting structure fastening on older vehicles was conducted. The use of engineering plastics around axle housings may create a uniform mounting style, as well as, lower costs of aftermarket rollover protective structures for these vehicles.

Various strengths of plastics were examined through shear testing, scale model testing, compressive strength testing, and cost examinations. Materials examined were a steel-filled epoxy, a fast curing steel-filled polyurethane, and an iron oxide-filled epoxy. The iron oxide-filled epoxy was chosen based upon strength and cost. This material showed the highest shear strength, 28.6 MPa, and compressive stress resistance. Compression testing of this material showed that it had an ASTM D695 unconfined compressive strength of 82.1 MPa and a compressive index of 157.1 MPa/mm. This material was also the least expensive at $0.06 per cubic centimeter of epoxy.

After material selection, testing of the plastic's strength was conducted through axle housing torsional loading, static ROPS loading tests, and field ROPS loading upset testing. To fulfill these tests an axle housing and plastic filled box was needed. Allis Chalmers D-17 axle housings were chosen due to this tractor still having a high number of tractors in use, estimated to be over 41,000 in 1993. In addition, there currently are not any commercial rollover protective structures available for this tractor. And, previous studies have been conducted that show D-17 axles could withstand ROPS loading.

Plastic filled boxes were designed to be a two part box that fit around the axle housing and sealed shut. Box dimensions were 25.4 cm long by 15.2 cm wide by 21.6 cm tall. Tops of all boxes produced had two 2.54 cm holes that allowed for plastic materials to be poured into the boxes.

Torsional axle housing and plastic filled box tests revealed that the torsional strength of the ROPS mount and axle housing combination had higher strengths with less twisting than the axle housing alone. Peak compressive stress on the axle housing with box was 62 MPa at 12.3 degrees of angular rotation. Previous researchers testing of a D-17 without a plastic filled box showed a peak compressive stress of 57.8 MPa at a angular rotation 18.3 degrees.

Next, a mounting structure was tested through static ROPS longitudinal and lateral tests done in accordance with SAE Standard J2194. From these tests, it was found that mounting structure and ROPS withstood energy requirements at ROPS deflections of

22.3 cm longitudinally and 16.3 cm laterally at the ROPS top. These values are well below the 41.5 cm longitudinal and 40.0 cm lateral calculated maximum allowable deflections needed to keep the ROPS from intruding into the operator's zone or exposing the operator's zone to the ground.

The final test conducted was a longitudinal field upset test. This test placed a plastic mounting structure on to a remotely operated D-17 tractor ballasted to normal operating weight, 2,118 kg. This tractor was then tested in accordance to SAE J2194 field upset test requirements. Results from this test show a passing of the standards requirements since ROPS deflection was 8.2 cm. Again well below the maximum allow longitudinal deflection of 41.5 cm.

An engineering plastic ROPS mounting system did withstand the forces applied during the static longitudinal and lateral ROPS tests. Field upset testing revealed the mounting system could withstand impact loads seen during actual upsets without a failure of the mounting system. During both static testing and field upset testing no permanent twisting of the mounting system was found. Since the mounting structure using engineering plastic showed no failures or permanent deflections during any testing and fulfilled all standard testing requirements, this system could be a viable option for a universal mounting structure for older tractors lacking an aftermarket ROPS. However, this study did not evaluate any weathering or degradation of the plastic material. Studies on how engineering plastic materials change after being introduced into weathering influences such as temperature extremes and cycling, UV light exposure, vibrations, and agricultural chemicals is needed.

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Chapter 1. Introduction

The agricultural industry has one of the highest fatality rates of any industry in the United States. A 1999 study gave a death rate of 22.5 workers for every 100,000 workers per year. This was six times higher than the national industrial average rate and second highest only to mining (Rautianen and Reynolds, 2002). Each year the United States has averaged between 250 and 350 fatalities due to tractor related accidents (NIOSH, 2004). An estimated 50% of these deaths were due to an overturn of tractors (National Safety Council, 1997).

Many steps have been taken to help in reducing the number of deaths caused by an overturning vehicle. A very successful step in lowering the number of rollover deaths was the addition of a rollover protective structure (ROPS) to vehicles (NIOSH, 2004). In Sweden, the government began requiring rollover protective structures on all new tractors in 1959. By 1990, 93% of tractors in Sweden had a ROPS or safety cab structure (Springfeldt et al., 1998). The death rate at that time was less than one death per 100,000 workers; 24 times lower than the United States (Thelin, 1998).

In 1993, there were around 4.8 million tractors in use on U.S. Farms. Roughly 1.8 million of these tractors had a ROPS in place (Myers and Snyder, 1995). This left more than 62% of tractors without rollover protection. It was estimated at the current pace of replacement and retirement, if no further retrofitting occurred, it would take 24 years for 74% of tractors in use to have rollover protective structures (Myers and Snyder, 1995).

There were many models of tractors that did not have protective structures available for them. This was partially because cost and complexity of manufacturing a mount made it unfeasible to produce aftermarket rollover protective structures (Li, 1997). The difficulty in designing ROPS mounting brackets came from various shaped axle housings. Since many vehicle models did not have square shaped axle housings, they could not have a traditional compression fitted mounting bracket placed on them. An example of this was the Allis Chalmers' D-17 (Ayers, 2003). This model was the fifth most popular pre-ROPS tractor. In 1993, there was over 41,000 D-17's still being used and the most used tractor without some type of retrofit ROPS available (Myers and Snyder, 1995). A pre-ROPS tractor was classified as a tractor that was manufactured without ROPS mounting locations designed into the axle housing or chassis. Many of these pre-ROPS tractor axle housings do not have flat surfaces to utilize a compression fitted ROPS mounts.

Engineering plastics have proven to be useful in aerospace and automotive industries due to their high strengths and low weights. These plastics could be used to provide an added structure onto pre-ROPS axle housings. This would give pre-ROPS axle housings a flat structure that would allow for compression fitted ROPS.

Chapter 2. Literature Review

2.1 Tractor Overturn and Vehicle Instability

From the early 20th century, tractors have evolved from heavy stationary power machines to lighter, road-ready vehicles (Jackson, 1985). However, little emphasis was given to safety and overturns at that time due to the large number of threshing accidents. As vehicles continued to become lighter, tractor upsets became increasingly common (Arndt, 1971; Klose, 1969).

The National Safety Council estimated that tractor overturns alone cause over 50% of the agricultural industry's annual deaths. The Council also estimated that there were over 350 deaths in the agricultural industry with an average of 4.3 deaths for every 100,000 tractors in use (National Safety Council, 2000). A 1999 study estimated a death rate of 22.5 deaths per 100,000 workers occur each year. This death rate was six times higher than the U. S. industry average (Rautiainen and Reynolds, 2002).

Throughout the United States there have been many studies conducted to examine the agricultural industry's high rate of fatalities. From 1988 to 1990, Lehtola and Marley (1992) conducted a study of tractor related accidents in the state of Iowa. During this study, there were 90 overturns reported, 56.7% of which resulted in a fatality. Of reported tractor fatalities, overturns were the leading cause of fatalities at 58.6%. The number of deaths annually due to overturns shows how vulnerable operators are to a loss of tractor stability.

Kim and Rehkugler (1987) stated that stability of a tractor depended on numerous factors that fell into one of five categories. These categories were static properties, dynamic properties, initial conditions, driver parameters, and terrain. Vehicles became unstable when the kinetic energy was quickly transferred into potential energy or tractor acceleration (Liu and Ayers, 1999). This transfer to potential energy can cause tipping motions that usually happened faster than the operator could react (Arndt, 1971). Once the motions and energies were transferred, the end result typically was a vehicle overturn.

Generally there were two types of overturns, side and rear. The situation of a front overturn could occur, but this overturn type was rare. Side overturns have been the source of around 85% of rolls. Therefore, side rolls have received more attention (NIOSH, 1996). This type of overturn was usually caused by the terrain, sharp turns at high speeds, or driving off of the roadway. The less common rearward overturn was caused by a large torque on immovable drive wheels (Kim and Rehkugler, 1987). While the rear overturn happened less, this type of overturn has proven to be more dangerous to operators. Springfeldt et al. (1998) found that 68% of all rear rolls resulted in a death. Side overturns were found to cause a death during 44% of rolls. The higher fatalities of rear overturns could be due to operators being thrown from tractors or a lower chance of operators escaping.

2.2 Rollover Protective Structures

Tractor overturns have been noted since the 1920's. As these rolls became more common, operator safety became a concern. Many safety methods were examined to retain vehicle stability. One such method was the Dooley safety switch. This safety device, introduced in 1939, was a mechanism designed to shut off engines if operators left their seat. This device was later abandoned because flywheel inertia continued to allow the tractor to roll over (Arndt, 1971).

Swedish researchers looked at many options to stop vehicle rollovers. In the end, they determined a frame was better to absorb energy during a roll than trying to prevent rollovers (Thelin, 1998). This frame was originally called an anti-roll bar. It was later changed to rollover protective structure given that anti-roll bar implied that frames would stop vehicles from rolling over (Schnieder, 1990).

In design and manufacturing of rollover protective structures, tradeoffs and balances were made between cost, appearance, function, and safety. ROPS material needed a good balance of both elastic and plastic properties. If a material had more elastic capacity, it would be over designed and add a large amount of weight to tractors. If there were more plastic capacity in the material, large amounts of permanent deflection would be seen. With large deflections, operators became endangered by being exposed to the ground or ROPS posts during a roll (Klose, 1969). A good balance of both properties gives the material exceptional toughness, which allows the ROPS to withstand overturns by absorbing energy without being over designed or adding risk to

operator. Material toughness also allows for minimal fatigue during normal tractor operation (Ross and DiMartion, 1982). Besides structure, other areas that concerned designers were visibility, simplicity, and extra utilities such as canopies (Lamouria et al., 1964). These areas were all important in selling rollover protective structures to operators. If operators did not experience normal visibility, simple use, or extra function to aid agricultural production or comfort, a ROPS would not be purchased (Jackson, 1985; Etherton et al., 1991).

Sweden led the way in ROPS research in the 1950's. Through many efforts they were able to slowly reduce fatalities caused by tractor overturns. Unlike the United States, Sweden did not rely on operator education to get farmers to place rollover protective structures on their tractors (Thelin, 1998). Instead, government mandates stated that all new tractors would be manufactured with a ROPS and seatbelt in 1959. In 1965, all employees had to be protected by a ROPS regardless of tractor age. By 1990, Sweden had a ROPS on 93% of all tractors in use. At that same time, deaths caused by tractor overturns had dropped from 12 per 100,000 workers in 1961 to 0.2 per 100,000 workers (Springfeldt et al., 1998). This amount was 24 times lower than the number of lives lost per 100,000 in the United States due to tractor overturns (Thelin, 1998).

Studies have been conducted in the United States as well that showed the effectiveness of ROPS. Woodward and Swan (1980) conducted a study to see if rollover protective structures were providing adequate protection as compared to non-ROPS tractors. The results from this study can be seen in Table 1. This study showed that more accidents

Table 1. ROPS Effectiveness Comparison (Woodward and Swan, 1980).

Injuries	With ROPS (%) 102 Accidents	Without ROPS (%) 101 Accidents	Unknown (%) 128 Accidents
No Injury	37.3	13.9	35.7
Minor Injury	25.5	13.9	10.7
Major Injury	18.6	21.8	14.3
Fatalities	14.7*	48.5	21.4
Unknown	3.9	1.9	17.9
Total	100	100	100

*14.7% of 102 accidents result in 15 fatalities not the 16 fatalities as indicated in paper.

ended with no injuries or minor injuries if the tractor had a rollover protection. Also, the number of fatalities observed in ROPS equipped tractors was nearly one third of the number observed in non-ROPS tractors. Of 16 fatalities found with ROPS equipped tractors, 14 operators were not wearing a seatbelt. It was thought by researchers that many if not all of these deaths could have be avoided if seatbelts had been utilized. Many manufactures and researchers felt that the use of seatbelts was vital to operator safety. Altamore et al. (1996) used computer simulation to show that operators had a high likelihood of being caught under the tractor or ROPS if they were not using seatbelts.

Other studies on ROPS effectiveness have been conducted with similar results. Schnieder (1990) showed that overturn death rates in Nebraska had been reduced through ROPS use. From 1969 – 1974 there were on average 22 overturn deaths each year. From 1975 – 1990 average overturn deaths had dropped to 6.9 per year. Lehtola et al. (1994) examined overturn reports for five years in Iowa. This group found that all 76 overturn fatalities occurred on tractors that were not equipped with rollover protective structures. Six injuries were reported where an operator was thrown from the tractor when not using the seatbelt supplied with the ROPS.

Unlike Sweden, the United States has relied on education and incentives to get operators to use a ROPS on their tractors. In the late 1960's, rollover protective structures became available as optional equipment on tractors (Myers and Snyder, 1995). Yet, even with added utilities and increased protections that come from a ROPS,

many people still found reasons not to use them. Operators found that the ROPS became restrictive when traveling around lower lying objects. This has lead many producers to remove their ROPS or just not purchase one. Other problems seen with a ROPS were a lack of benefit to production and the cost (Ross and DiMartion, 1982). It was not until 1985 that tractor manufacturers voluntarily agreed to produce all new tractors with a ROPS as standard equipment (Myers and Snyder, 1995).

2.3 ROPS Testing Standards

The initial work conducted in Sweden served as an example of how rollover protective structures could be tested. Need for a standardized method for ROPS testing was seen, (Arndt, 1971) and by 1966 the Society of Automotive Engineers (SAE) produced the first ROPS standard (Ross and DiMartion, 1982). Most ROPS testing standards used in the United States come from three groups; American Society of Agricultural Engineers (ASAE), International Standards Organization (ISO), and SAE. Two standards for agricultural tractors have been published in the last 30 years by ASAE and SAE. SAE Standard J1194, same as ASAE S383.1, was adopted in 1977 and reconfirmed in 1981 (ASAE, 2004a). A newer SAE J2194, same as ASAE S519, was adopted in 1990 and reaffirmed in 1994, 1997, and 2002 (ASAE, 2004b). This newer standard was written so that all U.S. rollover protective structures were acceptable on a worldwide basis (Fritz et al., 1991). A third standard, SAE J1040, was produced for construction vehicles but does includes a section on tractor ROPS testing (SAE, 2000).

2.3.1 SAE J2194

SAE J2194 (ASAE, 2004b) was written to create a set of test and performance requirements for ROPS designed for wheeled agricultural tractors. This was written as a performance code instead of a design code to give manufacturers design freedoms. The goal of this standard was to minimize frequency and severity of operator injury resulting from tractor overturns. Fulfillment of this standard required the minimum testing to be done as follows:

1) Meet material temperature testing requirements
2) Meet laboratory ROPS testing requirements
3) Finally, a seat belt anchorage must meet standard requirements.

The first issue coved by the standard was the operator's clearance zone. This was a protected region where an operator was to be located. This zone was to allow for room and controlled movement of an operator based upon the 95th percentile male (Fritz et al., 1991). As a load is placed up on the ROPS, it begins to deform. If this deformation enters the clearance zone during testing, the ROPS fail. This was known as an intrusion failure. Also during an overturn, the clearance zone might come in contact with the ground plane if there was too much ROPS deflection. This also caused testing failure and was known as an exposure failure. Ayers et al. (1994) developed a mathematical model which can evaluate and define maximum ROPS deflection amounts before intrusion or exposure failures were seen.

The next major issue of the standard was laboratory ROPS testing. This can be performed through either dynamic or static testing. A dynamic test was produced through a swinging pendulum that struck the ROPS from a calculated height. This method of testing was first used by Swedish researchers (Ross and DiMartion, 1982). Mass and height of the pendulum were both based upon tractor mass (ASAE, 2004b).

There were several disadvantages to this dynamic testing method. A large pendulum mass and arc were required for heavy vehicles which was difficult to obtain (Wetjen, 1976). Also, this test was based upon a linear relationship between tractor mass and energy (Arndt, 1971). For light tractors this test was too extreme, and for larger tractors it was not intense enough. Stienbruegge (1975) found that rollover protective structures failed in field testing after passing pendulum testing due to this problem.

Static testing methods were developed to help avoid these disadvantages. A static test was performed by applying a continual load to the ROPS at a very slow rate. The load was increasingly applied until energy requirements were met (ASAE, 2004b). Energy requirements were based upon many field test studies of various sized tractors. This method of testing proved to be more controllable and allowed for more convenient measurements of ROPS deflections. This method was also less expensive to produce for manufactures (Wetjen, 1976). For these reasons, it was felt that the static test was a more efficient test to perform.

Laboratory longitudinal (rear) loads were applied horizontally and parallel to the longitudinal median of the tractor. This load was applied to the uppermost portion of the ROPS or a point that is likely to strike the ground first. Area of application was inward from the outermost corner at one sixth of the ROPS top width. Load rate was considered static if the rate of loading did not exceed 5 mm/s. Energy requirements were found by a calculation shown in Equation 1 (ASAE, 2004b). The energy calculation was empirical and based upon results from rollover studies (Wenjen, 1976). Energy applied during testing was measured by calculating the area under the force-deflection curve.

$$E_1 = 1.4 * m_t \qquad (1)$$

where

$E_1 = Energy\,(Joules)$

$m_t = tractor\ mass\ (kilograms)$

After applying longitudinal loads, lateral (side) loading was to be conducted. Side loads were applied horizontally and perpendicular to the longitudinal median of the tractor. Load application location for a two-post ROPS system was at the uppermost point of the side opposite of where longitudinal loading had been applied. Figure 1 shows the loading location for both longitudinal and lateral testing. Loading was continued, at a static rate, until energy requirements were met. This test's energy requirements were found by Equation 2 (ASAE, 2004b). This equation was also empirical (Wenjen, 1976). Again energy applied was measured by calculating area under the force-deflection curve.

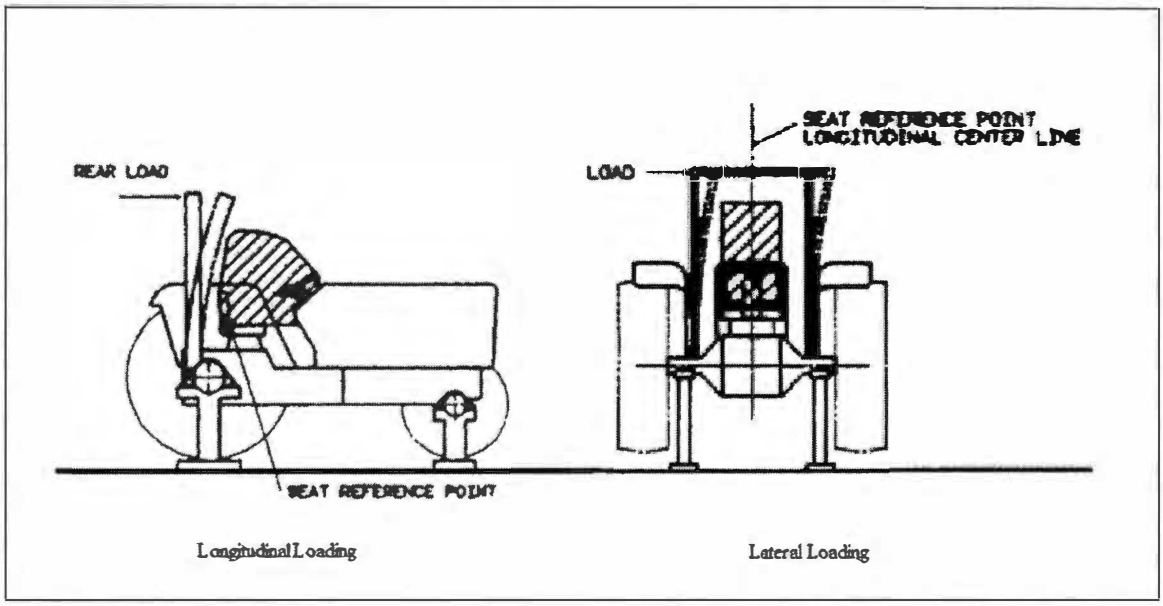


Figure 1. Static Loading Location for a Two-Post ROPS (ASAE, 2004b).

$$E_2 = 1.75 * m_t \qquad (2)$$

where

$E_2 = Energy\,(Joules)$

$m_t = tractor\ mass\ (kilograms)$

After laboratory test sections were completed, field upset testing was discussed. This test was discussed as an optional method for testing ROPS. To conduct this test, a normally ballasted tractor was driven onto a soil test ramp until either a longitudinal or lateral roll was produced. A longitudinal roll was produced by driving a tractor up a 60 degree slope at a rate of 5 to 8 km/h. Once the tractor was on the ramp, a roll would occur as it continued moving upward. Lateral rolls were achieved by driving a tractor at a rate of at least 16 km/h along and into a ditch bank of 50 degrees (ASAE 2004b). While these tests were the most realistic form of test, it possibly meant causing costly damage to the tractor.

The next issue covered by the standard was ROPS material requirements. To ensure that brittle fracture of ROPS was not seen at low temperatures, temperature material requirements were set. To pass all ROPS tests, material temperature should be at -18°C or below. Alternatively, material tested above that temperature was allowable if it exhibited certain Charpy V-notch impact strengths at -30°C. Allowable strength requirements for various size specimens were given in the standard.

The final issue covered by the standard was seat belt anchorage test. To ensure that operators stay within the clearance zone, a type 1 seat belt system was required. This is

the common lap belt. Belt system anchors had to withstand 4,448 N of tensile force in an upward and forward direction at 45 degrees. Also, anchors must withstand 2,224 N of tensile force in an upward and rearward direction at 45 degrees (ASAE 2004b).

2.3.2 SAE J1040

An additional ROPS testing standard often used was SAE J1040. This standard gave ROPS performance criteria for construction, earthmoving, forestry, and mining machines. A section in this standard addressed ROPS criteria for industrial tractors. While this standard's methods for testing were similar to J2194, there were a few notable differences. In addition to a required energy amount, a minimum force was also to be applied in both longitudinal and lateral static tests. Force requirements were met at 4.8 and 6.0 times the tractor mass for longitudinal and lateral directions respectively. Also, while longitudinal energy requirements were identical for both standards, lateral energy requirements were different in this standard. Lateral energy requirements were defined by Equation 3 (ASAE 2004a). This was based upon a different study that found this empirical equation. Temperature material and seat belt requirements were the same as in J2194.

$$E_3 = 12{,}500 * (m_t / 10{,}000)^{1.25} \qquad (3)$$

where

$E_3 = Energy\,(Joules)$

$m_t = tractor\ mass\ (kilograms)$

2.4 ROPS Retrofitting

A study by Myers and Snyder (1995) estimated that at the end of 1993 there were 4,800,000 ± 475,000 (mean ± standard error) tractors in use throughout the United States. Model years ranged from 1924 to 1994 with an average age of 22.8 ± 0.53 years. Of these tractors, it was estimated that approximately 62% were being operated without any ROPS device. And the most popular non-ROPS tractor without an aftermarket ROPS was Allis Chambers' D-17. And, there was over 41,000 D-17 tractors in use, all without rollover protective structures (Myers and Snyder, 1995).

A distribution of ROPS usage by tractor age is shown in Table 2. Data illustrated that new tractors were more likely to be equipped with a ROPS. This was likely due to the 1985 voluntary tractor manufacturer's agreement to include rollover protective structures on new tractors. These data also show that tractors manufactured before 1970, $\geq$ 25 years of age, were mostly without a ROPS. This is discouraging since studies have suggested that these tractors were the cause of most overturns (Lehtola et al., 1994). The large numbers of these older tractors being used also suggest that tractors are not retired from service. This is seen in the fact that tractors manufactured before 1970 average over 100 hours of annual use.

Another survey by Myers (2003) was conducted in 2001. From this survey it was found there were around 4,700,000 tractors in use throughout the U.S. This survey found that for the first time the number of ROPS equipped tractors was about the same as non-ROPS tractors. He assumed this was due to the voluntary ROPS agreement and large

Table 2. Estimated Number of Tractors and Percentage without ROPS by Age (Myers and Snyder, 1995).

Age Groups (years)	Tractor Numbers	Tractors without ROPS (%)	Average Annual Hours
0 to 4	320,657	8.5	430
5 to 9	386,565	23.9	475
10 to 14	627,751	35.2	425
15 to 19	762,587	42.5	375
20 to 24	683,567	65.5	310
25 to 29	606,742	84.4	250
30 to 34	391,949	92.6	225
35 to 39	319,813	97.8	120
40 to 44	376,712	98.0	110
45 to 49	152,236	98.0	105
$\geq$ 50	81,527	99.0	115

number for smaller tractors being produced with a two-post ROPS. The average age of tractors was 25.7 years. This again shows that tractors are not being retired from use since average age is increasing.

With proper maintenance, tractors were able to be used for many years. By retrofitting older tractors with rollover protective structures, safety could be extended from new tractors down to older models. This was and still is a difficult task to accomplish. Cost of designing and testing mounts for each style of tractor was a major concern (Etherton et al., 1991). Since many models produced before 1965 did not have ROPS designs, mounting locations were limited and difficult to work with (Johnson and Ayers, 1994).

The idea of retrofitting a ROPS onto tractors was not a new thought. Early in the 1960's U.S. military personal were pushing for retrofit ROPS on all Army tractors. Engineers evaluated and listed major issues that needed to be addressed for retrofitting a ROPS. These issues were as follows:

- Determine if chassis strength could withstand ROPS loads applied
- Determine which vehicles needed a ROPS first
- Determine what standard to use for testing
- Minimize cost
- Create a style that used simple and proven manufacturing techniques that could be done at all Army bases.

Unfortunately this project lost interest due to lack of funds and war in Vietnam (Hopler and Stewart, 1973).

Evaluating similar issues, agricultural engineers began studying the possibility of using rollover protective structures on older tractors. Tractors made before 1970 were typically not designed with a ROPS structural supports on their axle housing, a common ROPS mounting location. These tractors were known as pre-ROPS tractors. Some axle housings from these pre-ROPS tractors were thought to likely fail under loads seen in a vehicle overturn if ROPS was attached to them (Ayers, 2003). Schnieder (1990) found evidence that early retrofit rollover protective structures could pass standard testing methods. However, when these structures were tested in field upset tests, there were failures due to axle housing twisting and cracking. This gave operators a false sense of security when using a retrofitted ROPS. Ayers (2003) suggested that he had a concern with standard methods for testing since repeated measurement and chassis structural strength were not evaluated in standardized testing.

Other studies were conducted to show that retrofit rollover protective structures were effective when designed correctly. Effective ROPS mounting meant the difference in useable and unusable ROPS (Ross and DiMartion, 1983). Johnson and Ayers (1994) used finite element analysis (FEA) to find weaknesses of retrofit mounting areas. Their study showed that axle housings were able to withstand large loads before failure, but mounting styles were highly critical. Minor changes to a mounting structure designs showed large changes in the FEA analysis. This analysis was then confirmed through

lab testing of axle housings and mounting systems. A new ROPS utilized this improved mounting system and passed both static and field upset testing requirements.

In the U.S., the cost effectiveness of retrofitting a ROPS would be equivalent to the addition of airbags in automobiles (Myers et al., 1998). Other countries such as Sweden and Australia have been requiring farmers to replace or retrofit their tractors for years (Altamore et al., 1996; Thelin, 1998). Unfortunately, time loss and annoyance associated with retrofitting, along with lack of concern, have slowed Americans from purchasing retrofit protective structures (Hallman, 2005).

2.5 Axle Housing Strength

Attaching a ROPS onto older tractors required axle housings and mounting designs with ample strength. This was shown by Johnson and Ayers (1994) in their study of a retrofit ROPS. This study found that an axle housing from a Ford 8N could easily withstand loads seen during longitudinal ROPS testing. Other studies have also been conducted to evaluate if axle housings had adequate strength for ROPS loading.

Wen et al. (1994) examined a John Deere 720 axle housing by estimating loads seen during overturns through FEA. He considered side roll and crush testing more important than longitudinal rolls; therefore his study did not look at longitudinal rolls. This was done since more loading was placed onto single axle housings during side overturns as compared to rear rolls. Peak stresses in side evaluations were found to be 15,258 kPa. Crush evaluations were done by estimating that two times the mass of the tractor would

be placed upon the ROPS if a vehicle were upside down. Peak stresses in this test were 89,842 kPa. It was estimated through FEA that John Deere 720 axle housings could withstand stresses up to 241,325 kPa. Far above stresses produced during side or crush FEA testing. No lab testing was done to validate any of these estimated stresses.

Li (1997) conducted tests on Ford and Farmall axle housings to determine if simple axle configurations had sufficient design margins to withstand rear ROPS loading. He considered rear loading more critical since higher torques were being placed onto the axle housings during this style of loading. He used a testing apparatus that was constructed with a base frame for securely mounting axle housing and a hydraulic cylinder for applying loads. A force sensor and potentiometer gave continuous data of cylinder load and travel. After an axle housing was mounted to the base frame, a two meter tall I-beam was then attached to axle housing using conventional ROPS mounting techniques. Loads were then applied to the I-beam from the hydraulic cylinder. This caused a torsional load about the center axis of the axle housing.

Loads were applied until a failure was seen or full cylinder extension had been reached. After evaluating force and deflection data from each test, a strength or yield torque was found. If failure occurred while still in an elastic deflection range, then torque at failure was considered Strength Torque. An example of this was seen in Figure 2. If failure occurred after plastic deformation had been reach, Yield Torque was calculated. This author chooses to use torques as a bases of comparison and did not list any stresses. To calculate yield torques, torque versus axle housing degree of twist curves were

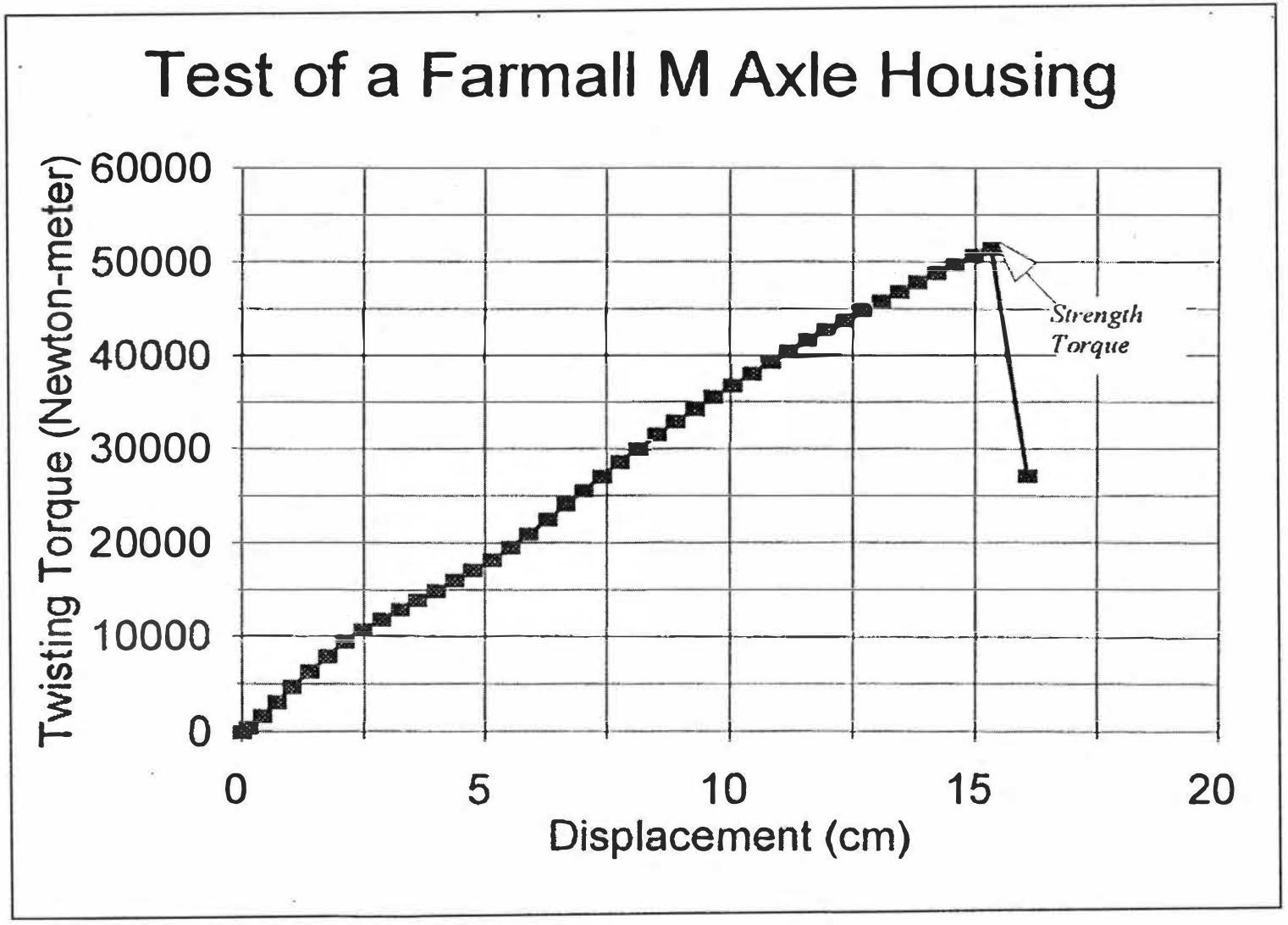


Figure 2. Example of Strength Torque Location on Farmall M Axle Housing (Li, 1997).

made. A line was then drawn parallel and 0.72° right of the elastic loading line. 0.72° was 0.2% of 360°. This was based upon engineering practice of 0.2% proof offset to find yield points (Gillam, 1969). Figure 3 gave an example of a torque versus axle housing twist curve used to find yield torque by 0.2% proof offset.

Once axle housing strength had been established, a ROPS was constructed for each style of axle housing tested. These rollover protective structures were then subjected to longitudinal static testing in accordance to SAE J2194. After conducting these tests, torques applied to reach energy requirement was calculated. Having this value and axle housing strength values allowed Li to calculate a design margin for each axle housing set. Three equations were used to calculate design margin. Equation 4 was used if ROPS entered a clear plastic deformation during static testing.

$$DM_1 = \frac{YT}{0.5 * ST} \qquad (4)$$

where

$DM_1 = design\ margin$

$YT = Yield\ Torque\ (Newton\ meter)$

$ST = Static\ Torque\ at\ energy\ requirement\ (Newton\ meter)$

This equation assumed that equal loading was placed on both axle housings since the ROPS was into plastic deformation. When a ROPS did not clearly enter a plastic deformation stage, loading could not be uniformly distributed. Li assumed an extreme case that one post bore 75% of the total load. This can be seen in Equation 5.

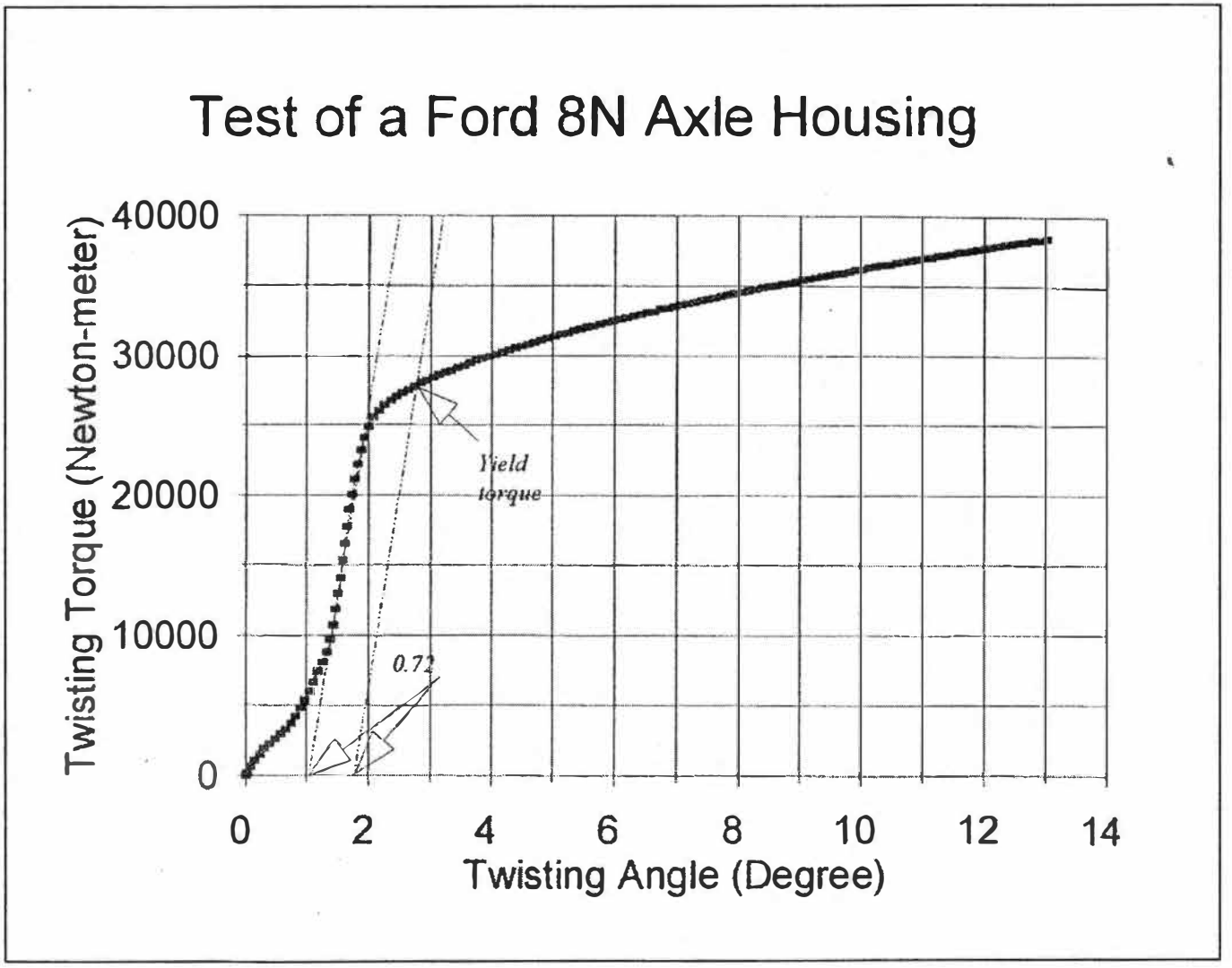


Figure 3. Example of Yield Torque Location on Ford 8N Axle Housing (Li, 1997).

$$DM_2 = \frac{YT}{0.75 * ST} \qquad (5)$$

where

$DM_2 = design\ margin$

$YT = Yield\ Torque\ (Newton\ meter)$

$ST = Static\ Torque\ at\ energy\ requirement\ (Newton\ meter)$

Equations 4 and 5 were only applicable to axle housings that yield torque or longitudinal static torques calculated. Another method for calculating design margin was established that used either yield or strength torque (Li, 1997). This method assumed the minimum load placed on a ROPS was 4.8 times the mass of the tractor. This was based upon SAE J1040 longitudinal force requirements of 4.8 times the mass of the tractor. Also, this method assumed that vertical ROPS height, torque moment arm of the ROPS, was 2.5 meters, higher than most rollover protective structure. This procedure is shown in Equation 6.

$$DM_3 = \frac{T * n / h}{4.8 * M_t} \qquad (6)$$

where

$DM_3 = design\ margin$

$T = Yield\ Torque\ or\ Strength\ Torque\ (Newton\ meter)$

$n = number\ of\ axle\ housing\ (2)$

$h = vertical\ ROPS\ height\ \ (meter)\ (2.5m)$

$M_t = tractor\ mass\ (kilogram)$

Li used these equations to find design margins for all axle housings he tested. Results from his testing are given in Table 3. Ford 8N and Farmall M were the only axle housings that had static ROPS testing completed. Therefore, these were the only axle

Table 3. Summary of pre-ROPS Axle Housings Tested (Li,1997).

Manufacture	Model	Years Produced	Mass (kg)	Yield/Strength Torque (Nm)	Design Margin 1	Design Margin 3
FORD	2N	1943-1947	N/A	26,250	N/A	N/A
	8N	1947-1952	1232	28,850	2.22	3.9
FARMALL	M	1939-1949	2204	50,600	1.41	3.8
	450	1956-1958	3119	53,500	N/A	2.9
	H	1939-1949	1676	42,400	N/A	4.2
	460	1958-1962	2747	72,600	N/A	4.4

housings that could have design margins calculated based upon Equation 4. And only equation 4 was reported since the author considered all ROPS deflections to be into plastic deformation. All axle housings had design margins calculated based upon Equation 6 except Ford 2N which did not have a tractor mass listed. Li's work showed that pre-ROPS axle housings were able to carry loads needed to support a ROPS. His work also set a method for calculating a design margin of axle housings.

Ayers (2003) conducted research similar to Li, however his work was focus on axle housings that mount structures were more complex. These axle housings were more complex due to the fact that there were no flat or square surfaces to mount a ROPS onto. One housing that Ayers examined was the Allis Chambers D-17. This axle housing was a cylindrical shape that made compression mounting difficult. Compression fittings easily rotated on cylindrical shaped axle housings. To alleviate this problem, compression fittings interlocked on top of fender mounting lobes on the axle housing. An example of this mounting style was shown in Figure 4.

D-17 axle housings were tested to find yield strengths (Ayers, 2003). Next, static testing was conducted to find if a D-17 axle housing could withstand loads seen in standard ROPS testing. From static testing, it was found that energy requirements for longitudinal and lateral tests were met at deflections of 23 cm and 26 cm respectively. These values were well below the allowable deflection amounts of 41.5 cm longitudinal and 40 cm lateral. During both styles of static testing, no failures of the axle housings were observed.

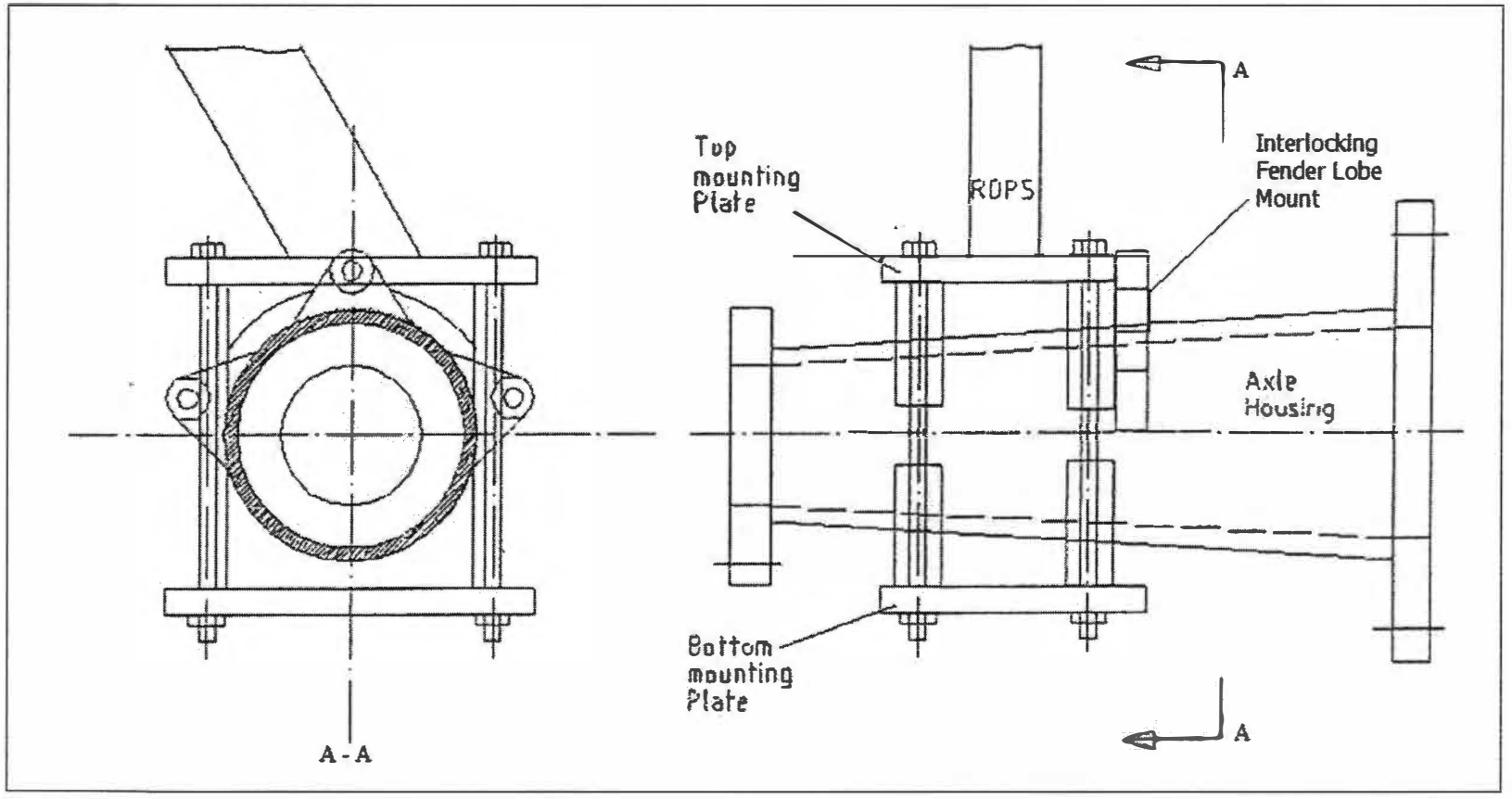


Figure 4. D-17 Axle Housing Mount System (Ayers, 2003).

Having conducted axle and static testing, Ayers (2003) calculated a design margin of 1.6 for the D-17 based upon Equation 4. This value showed that D-17 axle housings had ample strength to withstand loads needed to support ROPS. This was verified by conducting a field upset test using this mounting system. During this test no axle housings failures of permanent twisting was seen. While this axle housing has proven to be able to withstand loads seen during tractor overturns, a ROPS still do not exist for this axle housing due to its interlocking mounting system. This interlocking system had tight tolerances and detailed machining needed to fit on top of the fender lobes. This has lead Ayers to believe that an innovative, conveniently mounting system is needed for complex pre-ROPS axle housings and axle housings where a commercial ROPS are no longer available.

2.6 Engineering Plastics

In the last half of the twentieth century, usage of plastics in everyday life increased very rapidly. It was estimated that polymer plastics were the second most important material group at the end of last century. They were second behind steel products, but were expected to surpass steel during this century (Feldman, 2002). This rapid increase was caused by two factors. One reason being the ever reducing amount of natural resources available to produce structural metals. The other reason was the flexibility and light weight of most polymers. Epoxies, for example, are often used in aerospace industries due to its light weight, high strength, and processing additives that improve the materials performance. Improvements can be easily made to modulus of rigidity, tensile strength,

thermal stability, and other structural parameters during manufacturing of the material (Chang et al., 2004).

Polymers are very common in automobiles. This was partially due to regulations that required a certain amount of materials within the car to be recyclable (Luda et al., 2002). Polymers gave producers three key areas of improvement over metals; design flexibility, weight savings, and reduced production cost. Since polymers are typically liquids during the mixing and molding stages, cast can mold around and into virtually any shape (Luda et al.). Also, it was estimated that by using a composite, component weight was typically reduced by 35%. Examples of polymers in cars can be seen in body panels, bumpers, engine valve covers, air intake manifolds, airbags, oil sumps, roof and trunk racks, floor boards, and V-belts (Stewart, 2004).

Stewart's (2004) study of polymer uses in automobiles showed that by using these composite materials, many advantages were seen. When replacing aluminum oil sumps with plastic sumps, not only was weight lowered by 50%, but also oil holding capacity was improved by 30% and noise was lowered by one decibel. Roof racks made of plastic reduced weight and production time by eliminating a need to forge, weld, and paint steel. Chapman and Vesey (2000) found that body panels produced from polymers improved durability, had mold-in-color, and better dimensional stability. And, by using injection molding, producers were able to make a robust, repeatable process that lowered assembly time and save money. These added advantages to the weight savings all come from the polymers ability to consistently mold into various shapes.

For many applications, plastics have begun to surpass other materials in transportation industries. Designers, however, must anticipate working condition requirements when choosing a material. Often times a compromise between performance, cost, and manufacturability was needed. A clear definition of product requirements in life expectancy, size, color, shape, and strength aids in choice of materials (Rosato et al., 2001).

Epoxy composites were favored for many engineering plastic applications due to their good mechanical strength and hardness (Zhang et al., 2004). These materials were widely used as adhesives, coatings, composites, and laminates because of high strength, low creep, extremely low curing shrinkage, environmental resistance, and low frequency vibration damping (Ratna et al., 2004). Dobrzanski et al. (2005) found that epoxies with additional metal flakes gave excellent corrosive resistance in harsh environments.

American Society of Metals (ASM International, 1987) stated in their Engineering Materials Handbook that epoxies offer a broad range of physical, mechanical, and processing capabilities that made them invaluable by comparison to other composites. Epoxies were easily produced and formed no by-productions during curing. Typically epoxies were filled with a fiber or flakes that enhanced physical properties of the epoxy matrix. Often times, a matrix will strongly reflect the physical properties of its filler due to epoxy's strong adhesion (ASM International, 1987). It has been found that by using small flakes or ground fibers higher impact resistance and energy absorption resulted in epoxies (Jacob et al., 2004).

One major weakness of epoxies was their poor ability to withstand UV weathering (ASM International, 1988). Weathering was defined as "the undesirable change produced by outdoor exposure" (Feldman, 2002). This weathering or degradation arose from a wide variety of environmental factors. UV weathering was just one factor that degrades polymers. Common effects were discoloration, tackiness, loss of material, and brittleness. These changes were irreversible and slowly weaken the matrix and its physical properties (Feldman, 2002).

Many studies have been conducted to evaluate polymer performance through a number of environmental conditions. Luda et al. (2002) studied the outdoor ageing effect on strength properties in older car bumpers. This study found that bumpers at ten years old have only slightly lower elastic modulus, yield stress, and failure stress than newer bumpers. However, the toughness, found by impact loading, was up to three times lower than newer bumpers.

Temperature effects were the second leading cause of degradation behind UV light. Lopez-Anido (2004) looked at temperature cycling effects on shear strength and adhesion in epoxies. This study found that after only 20 freeze-thaw cycles, shear strength was reduced by 40%. It also found that failure modes changed to more brittle fracturing after temperature cycling. However, these effects were tested on a fiber-reinforce epoxy. Other methods of reinforcement may show different effects from temperature cycling. Chang et al. (2004) found that impact resistance had little variation under different temperature levels. Only after the material was heated to over 80°C did

impact resistance degrade. This may prove to be an issue for engineering plastics placed onto tractors.

2.7 Summary

Tractor overturns account for over 50% of annual deaths in the agricultural industry. The use of rollover protective structures has aided in reducing overturn deaths and injuries. This is because ROPS struck the ground and absorbed energy while protecting operators. Numerous testing standards gave guidelines on material, energy absorption, and seat belts. These were done to limit severity of injury during a vehicle overturn and ensure that all tractors meet the same requirements.

After rollover protective structures were seen to aid in preventing overturn deaths, many people began exploring placing them on older tractors. However, many people were concerned with placing loads seen during overturns onto frames or axle housings of tractors that were not designed for these loads. Axle housing strength testing was conducted and resulted in a design margin calculation that would show if axles housing could withstand ROPS loading. It was found that even when axle housings could withstand loads, the design and fabrication of a ROPS was often too costly. An innovative mounting process was needed.

Engineering plastics could supply that innovative mounting system. These products have been growing rapidly into daily life over the past decades. Aerospace and automotive manufactures are constantly looking at ways to incorporate these materials

in their products. Engineering plastics, such as epoxies, offer high physical strength, ease of processing, and are widely used for their broad range of applications. Environmental weathering could be the only draw back to this type of mounting system.

Chapter 3. Objectives

The objective of this study was to evaluate the feasibility of using an engineering plastic as a rollover protective structure (ROPS) mounting aid on tractor axle housings. The concept of this project was to develop a metal box that encloses axle housings at the location where the ROPS was to be mounted. This mounting box was then injected with engineering plastic to adhere to the axle housing and provide support to keep the box from rotating. The ROPS would then be attached to this mounting box. The specific objectives of this project included four main evaluations:

1) Evaluation of strengths and cost of different engineering plastics
2) Evaluation of longitudinal torsional strength of a full-scale axle housing mount on an Allis Chalmers D-17 axle housing
3) Evaluation of Allis Chalmers D-17 axle housing mount during a static longitudinal and lateral ROPS tests
4) Evaluation of Allis Chalmers D-17 axle housing mount during longitudinal field upset test.

A D-17 was chosen due to it being the most popular pre-ROPS tractor without any form of an aftermarket ROPS. Also, this tractor has had an axle housing ROPS mount produced that could be improved through an engineering plastic mounting system. To fulfill the objectives of this project, research was broken down into four main phases that followed the four objectives.

The first phase was to examine various engineering plastics and choose a material(s) that would meet project needs. The second phase of testing was to conduct torsional test of axle housing and plastic filled mounting box combinations. This was done to see if the combined system would have adequate strength to endure a rear vehicle roll. The loads for this test needed to be greater than 38,000 Nm. These values were based upon loads from static ROPS tests conducted on the D-17 as found in previous testing (Ayers, 2003).

The third phase was to take combined axle housing and plastic filled boxes and implement them on to a D-17 tractor chassis. This ROPS will then be subjected to a static longitudinal and lateral ROPS tests as specified in SAE J2194 standard. Finally, phase four testing was done to validate testing from phase three. In this phase, a fully functional D-17 tractor was implemented with a plastic filled mounting box and ROPS combination. This tractor was then subjected to field upset test as specified in J2194.

This current study was focused on aiding in protecting the operator during both longitudinal (rear) and lateral (side) roll situations. Yet, rear rolls could be more dangerous to operators due to torsional failures in an axle housing and plastic filled box combination. During a side overturn, these torsional loads about the axle housing's center line are not present. For this reason, phase two and four will only evaluate longitudinal testing.

Chapter 4. Phase One: Material Selection

In selecting a material, there were many engineering plastic data sheets examined. Conditions needed from the data sheets were:

- Low viscosity to pour at ambient temperatures, below 3000 cp
- Curing in reasonable time under ambient temperatures and pressures, approximately 60 minutes
- High tensile strength to resist failure during applied loading, greater than 24 MPa.

Three materials were selected based upon these criteria. Devcon Liquid Metal (Devcon 10210) was a steel-filled liquid epoxy. Freeman Liquid Steel (Ren RP132) was a fast-curing steel-filled polyurethane. And Freeman Liquid Iron (Ren RP 3215-3) was a iron oxide-filled variable hardness epoxy.

4.1 Methods and Materials

Testing was completed to determine if any of these engineering plastic(s) had a high enough strength to be used for this project. To make this determination, plastics were subjected to two different types of testing. The first of these tests was a shear strength test. The second test was a scale model torsion test. This test was approximately a 1/3 scale simulation of full scale torsional testing that would be conducted later. Compression testing was later conducted to gain additional knowledge of the material chosen from the first two material tests.

4.1.1 Shear Testing

A completely randomized design experiment was setup to compare mean shear stresses between each of the three engineering plastics. Each engineering plastic was used as a different treatment. Eight to fifteen repetitions were made for each plastic examined. Knowing that there were three treatments and a maximum of fifteen repetitions, SAS (SAS, 2002) was used to randomly assign treatments.

Treatments were assigned to tubes that allowed the plastic to cure into a small cylinder, approximately 0.75 cm. A cylinder was then placed into a hole in the shearing mechanism. This shearing mechanism was attached to a MTS universal test machine model Alliance RT/30. The shearing attachment consisted of stationary outer edges and an inner section that moved with the test machine's crosshead. As the crosshead moved upward, material specimens would shear at the contact points of the outer sides. Since there were shears at two contact points, this was considered to be a double shear (Yu, 2004). The double shear box and universal testing machine is shown in Figure 5.

Testing followed the randomized layout produced by SAS. As each specimen was sheared, cylinder shear plane area was recorded. Using shear load and area, shear stress was calculated for each specimen. A completely randomized analysis of variance model was conducted in SAS and used to compare the means (SAS, 2002). Mean separations were done through the default least significant difference (LSD). Data were normal, had equal variance, and were without outliers; therefore no transforms or adjustments were made.

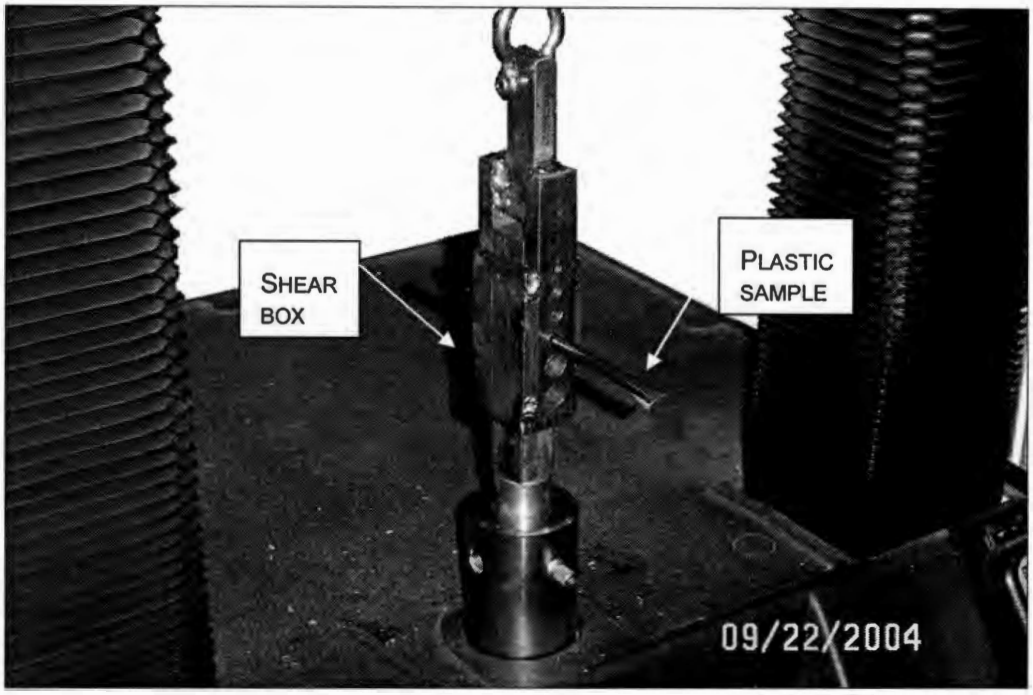


Figure 5. Universal Test Machine with Shearing box.

4.1.2 Torsional Scale Testing

To further investigate strength of the materials, approximately 1/3 scale models of axle housing and mounting box combinations were made. This would allow for torsion testing to be conducted that resembles loading applied by the ROPS. Axle housing models were produced by using 50.4 cm diameter steel cylinders and four lobes, 1.9 cm tall by 1.7 cm wide by 0.8 cm thick, that were welded to the steel cylinder. Figure 6 shows an example of this scale axle part. The groove shown in the figure was where the lobes were placed and was welded shut.

Once a box was assembled with a scale part in place, engineering plastic was poured in and cured. A lever arm was attached to the end of scaled axle parts that extended out of the box. Forces were applied on this arm to produce a torsional load. A load cell and linear variable differential transducer (LVDT) measured force and deflection at the lever arm. These measurements were logged into a Campbell Scientific 23X data logger. The set up of this scale model on a hydraulic press is seen in Figure 7. Measurement accuracies for the system were calculated to be ±20 N and ±0.04 cm.

After conducting each torsional test, boxes were cut open and examined for causes of failure. By cutting open these boxes, it could be seen that shearing was not a major cause of failure within the boxes. Examination showed that material compression strength was resisting lobes movement easily through the engineering plastic.

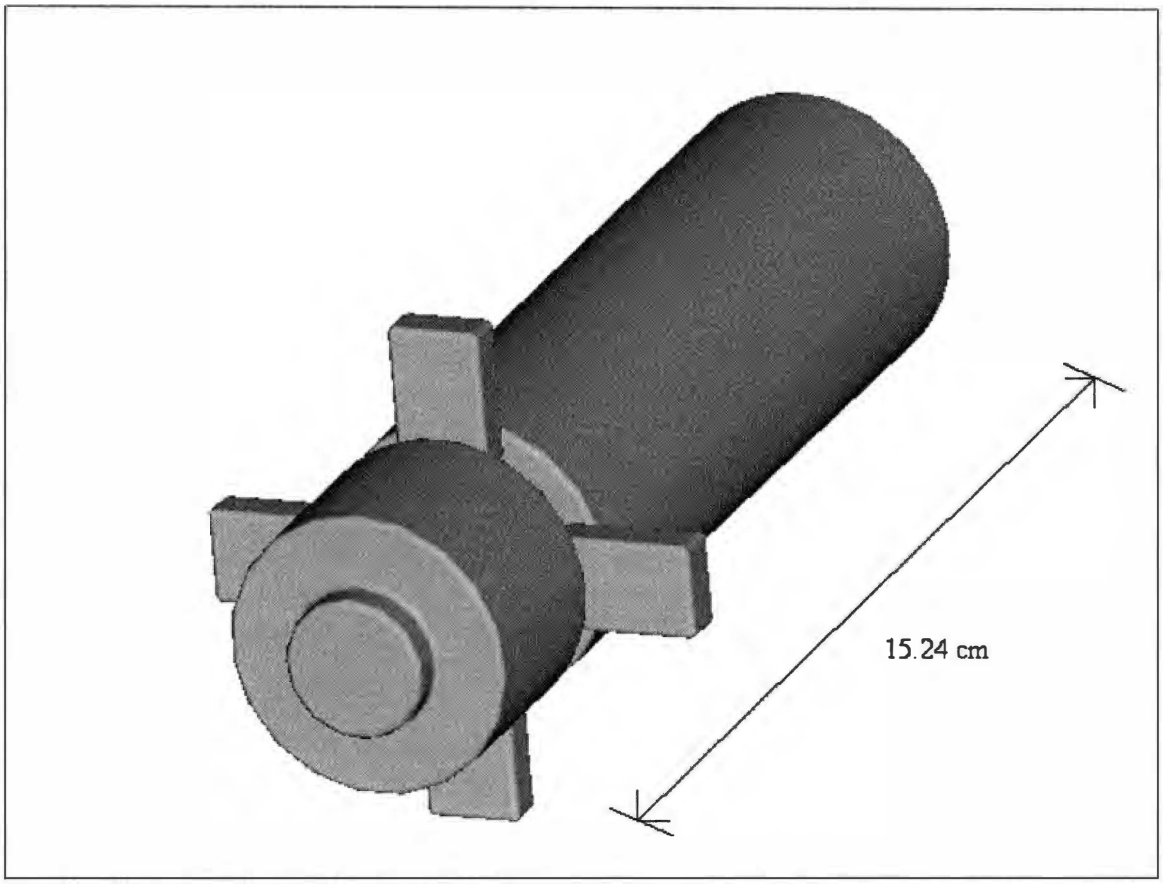


Figure 6. CAD rendering of Scale Axle Part.

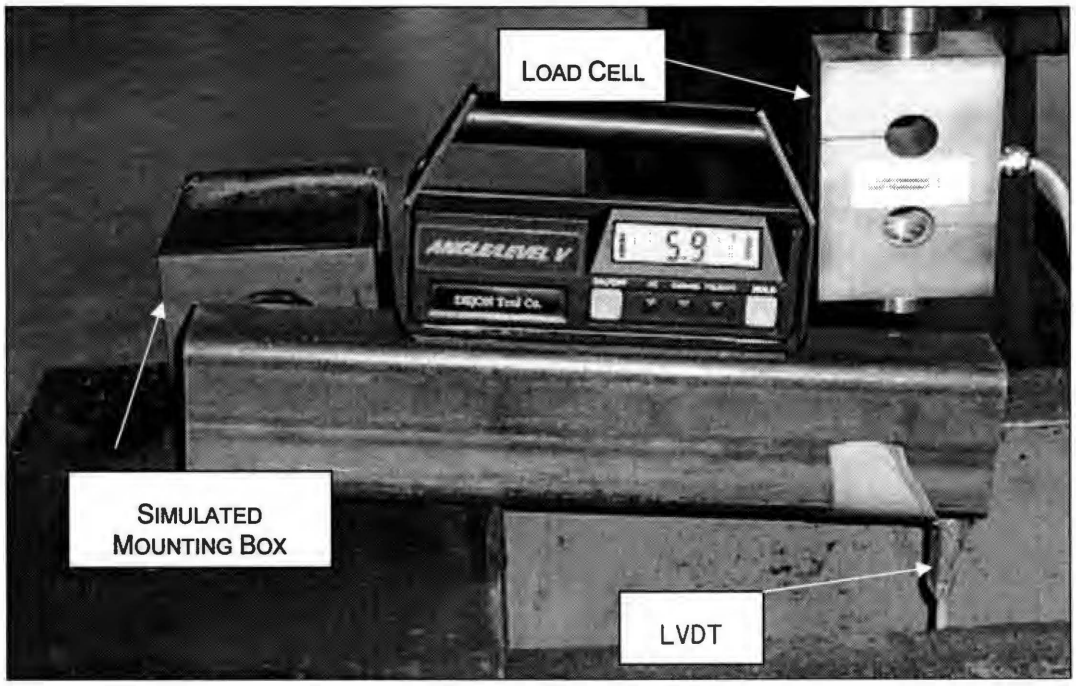


Figure 7. Torsion Test Set Up on Hydraulic Press.

A mathematical model was used to calculate the compressive stress on each lobe face during testing. This model assumed that compression resistance was the lone parameter keeping the scale part from moving through the plastic material. To calculate this compressive stress, torque values were multiplied by the moment arm to the center of each lobe. The force on each lobe was then divided by lobe face area to find the compressive stress. Figure 8 shows a view of the scale model part with it loading points.

4.1.3 Compression Testing

Compression testing was conducted to gain additional knowledge on material behavior. Early in this research, it was thought that lobes might shear through plastics placed inside the box. Scale model testing showed that this was not occurring and material was compressing. For this reason, compression testing was conducted on the material chosen from the shear and torsional test results.

Manufacturers of all materials evaluated listed material specifications on their Web sites. Compressive strength was one of the properties listed. Testing used for finding this property, American Society for Testing and Material Standard (ASTM) D695, called for an unconfined cylinder of material. This test was performed by placing 1.27 cm diameter by 2.54 cm length cylinder (½ in by 1 in) samples between two large parallel surfaces. Loading was applied downward onto a sample causing a compression. Loading continually increased until rupturing occurred (ASTM, 2004).

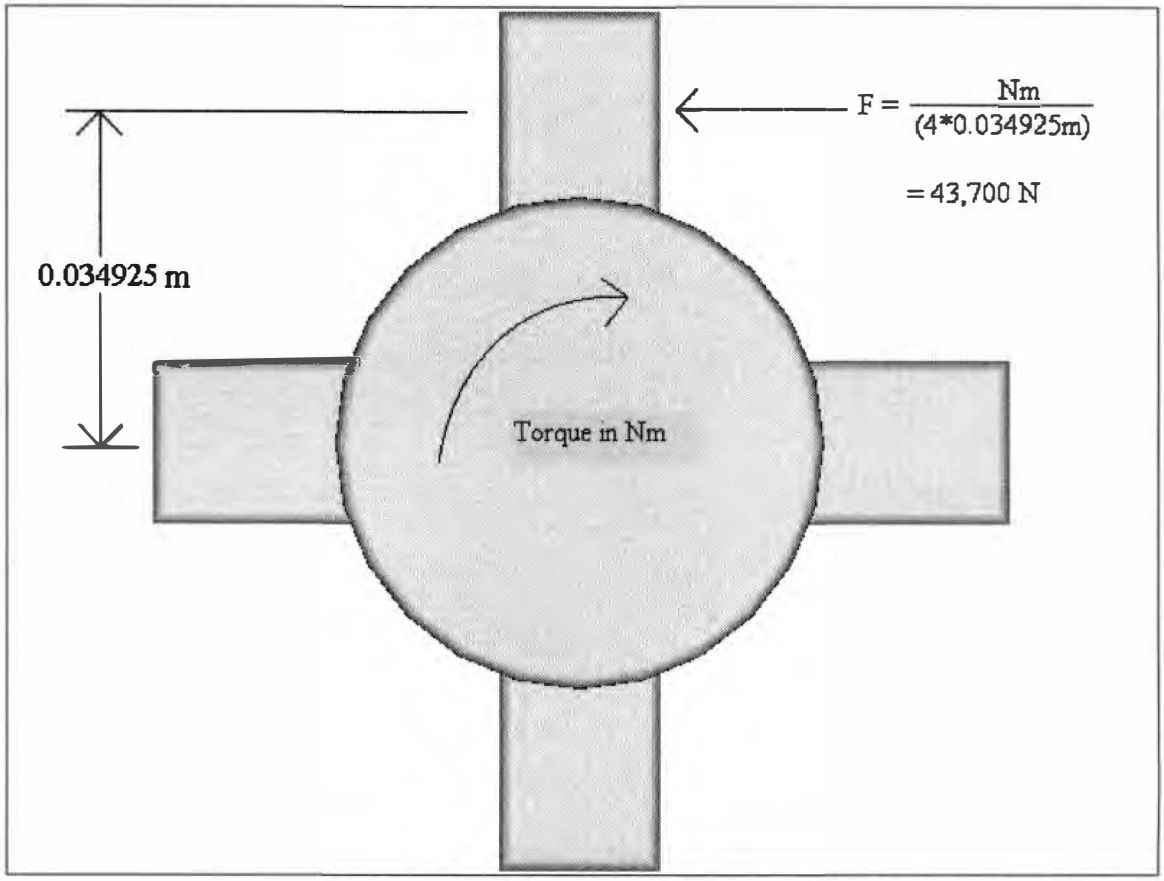


Figure 8. Torque and Force Location on Scale Model Lobes

ASTM D695 testing was conducted on 10 specimens of the chosen material by a universal testing machine and compared to stated compressive strength given by the manufacturer. This comparison was conducted in SAS using Student's t-test for one sample mean. It was expected that the mean of the tested specimen would be slightly lower than the value given by the manufacturers. This was due to a different amount of resin to hardener used in this study than used by the manufacturers. Manufacturers used a 10:3 ratio while this study used a 10:2.5 ratio. This increase in total amount of resin used which increased total amount of iron oxide flakes within the plastic's volume. Chang et al. (2004) found that increasing filler amounts aided adhesion and wear resistance, but slightly lowered impact and compressive strengths.

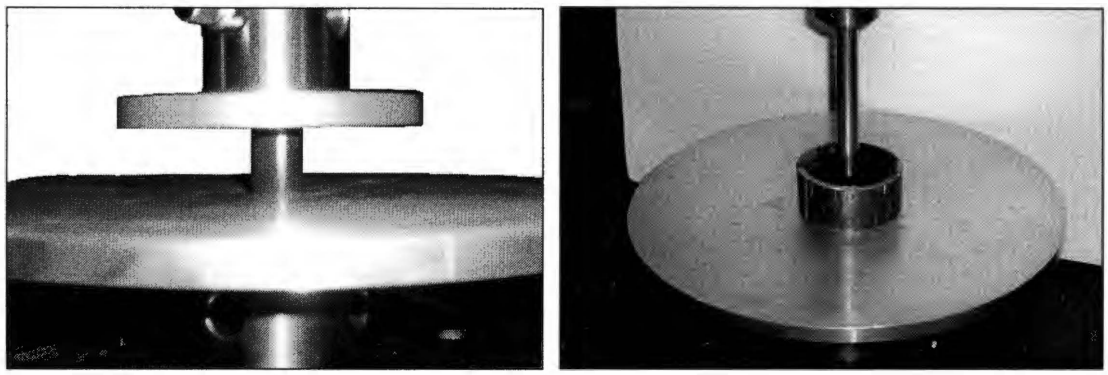
Since materials being used in this test were unconfined, it was felt that values given by manufactures do not reflect compression loading applied in this study. A method for testing that used a small surface compression onto a larger confined area of plastic was needed to represent axle housing lobes imbedded in plastic.

To find a confined compressive strength of material used in phases two and three of this study that better represents our confined compression, material specimen were poured into a steel container and left to cure for seven days. The steel container used was a 5.08 cm inside diameter tube at a height of 3.18 cm and a wall thickness of 0.48 cm. Once curing time had passed, samples were placed upon the universal testing machine platform. A 1.27 cm diameter probe was then lowered onto the material samples by the cross sectional head at a constant travel speed of 1.3 mm/min. Peak force and force

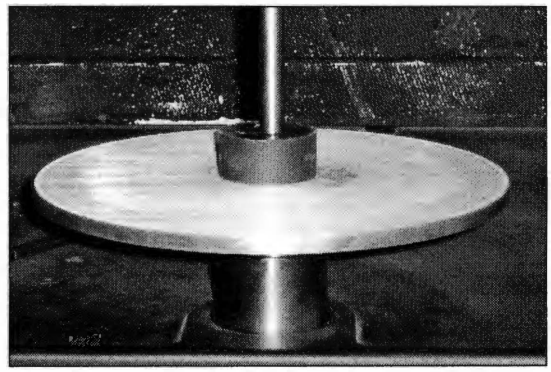
versus deflection curves were acquired during each test. Eight specimens were tested for confined compressive strength.

Since this style of testing is not standardized, results from this test may be different from ASTM D695. Differences could be from influences caused by metal rings allowing materials to not fracture or bulge. Similar testing was conducted without metal rings to determine if there was any influence on compression strength from the ring. This similar test was not considered confined since metal rings were not present. This was designated as a semi-confined test since compression applied through 1.27 cm probe was confined by its surrounding epoxy material. Eight specimens of semi-confined material were tested for compressive strength. An example of all three compression tests is shown in Figure 9.

During confined and semi-confined compressive testing, it was found that the universal testing machine used was unable to reach peak material strength. The load sensor reached its maximum before a failure point was observed. Using force versus deflection curves from these confined and semi-confined tests, a compressive index was calculated. This was available since there was a constant linear section on each curve. Forces were converted to stresses by dividing by probe tip area. An example of a compression stress versus defection curve is given in Figure 10. Compressive index/slope was calculated for each confined and semi-confined sample tested. This slope amount is shown in Figure 10 by the trend line's equation.



(a) (b)



(c)

Figure 9. Compression Testing Setup on Universal Testing Machine: (a) Unconfined (ASTM Test), (b) Confined Test, and (c) Semi-Confined Test.

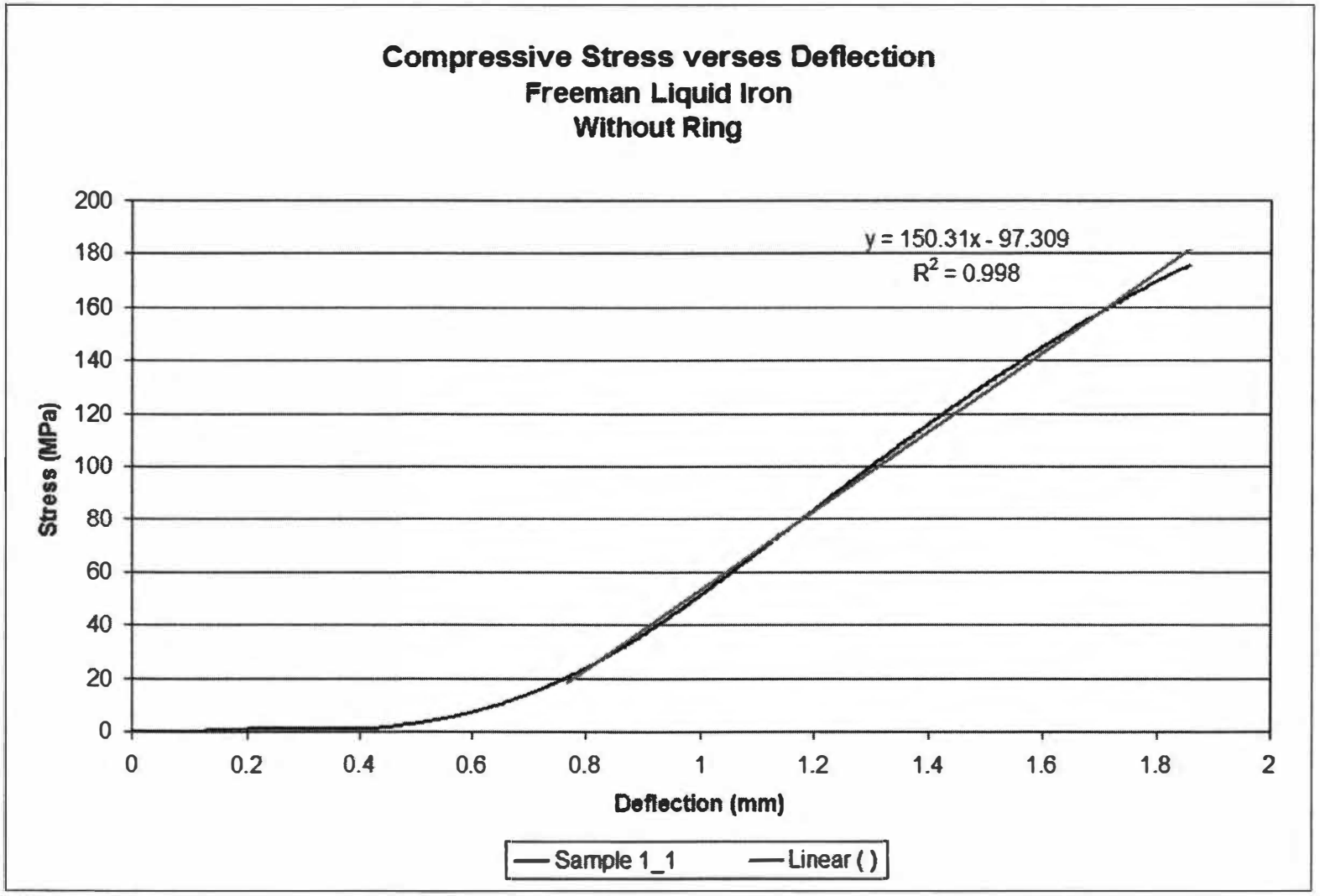


Figure 10. Compressive Stress versus Deflection with Linear Fit Line and Slope.

Student's t test for dependent samples was used to compare mean compressive indexes for both the confined and unconfined specimens. Confinement, with a metal ring, and semi-confinement, without a metal ring, were used as treatment factors to be compared. The eight specimens tested for each treatment were used as replications. SAS was used to conduct the Student's t test and test for normality and equal variance. If no differences were seen between the confined and semi-confined tests, it would show that compression was centralized to load bearing area.

4.2 Results

Table 4 gives the results from this data analysis. It can be noted that mean shear stresses of each material was statistically different according to analysis of variance testing ($P < 0.05$) (SAS, 2002). Devcon Liquid Metal and Freeman Liquid Iron mean shear stresses were close to each other but not statistically similar. Freeman Liquid Steel had a much lower shear stress than the other two materials. The Freeman Liquid Iron had the highest shear stress of all materials at 28.7 MPa.

4.2.2 Torsional Scale Testing

Comparisons were based upon compressive stress on each material at the lobe face. This stress was calculated using the compressive model discussed during section 4.1.2. From this model it was found that Freeman Liquid Iron had the highest peak compressive stress at 298 MPa. Second highest compressive stress was Devcon Liquid Metal at 220 MPa. The lowest compressive stress was from Freeman Liquid Steel at 173 MPa. Figure 11 provides results from torsional load testing. Freeman Liquid Iron

Table 4. Epoxy Mean Shear Stress and Standard Error.

Material	Mean Shear Stress (MPa) with Mean Separation Value		Std. Error (MPa)	N
Devcon Liquid Metal	26.67	B	0.81	10
Freeman Liquid Steel	21.18	C	1.14	8
Freeman Liquid Iron	28.62	A	0.60	15

*Mean Separation Values given from LSD separation technique.

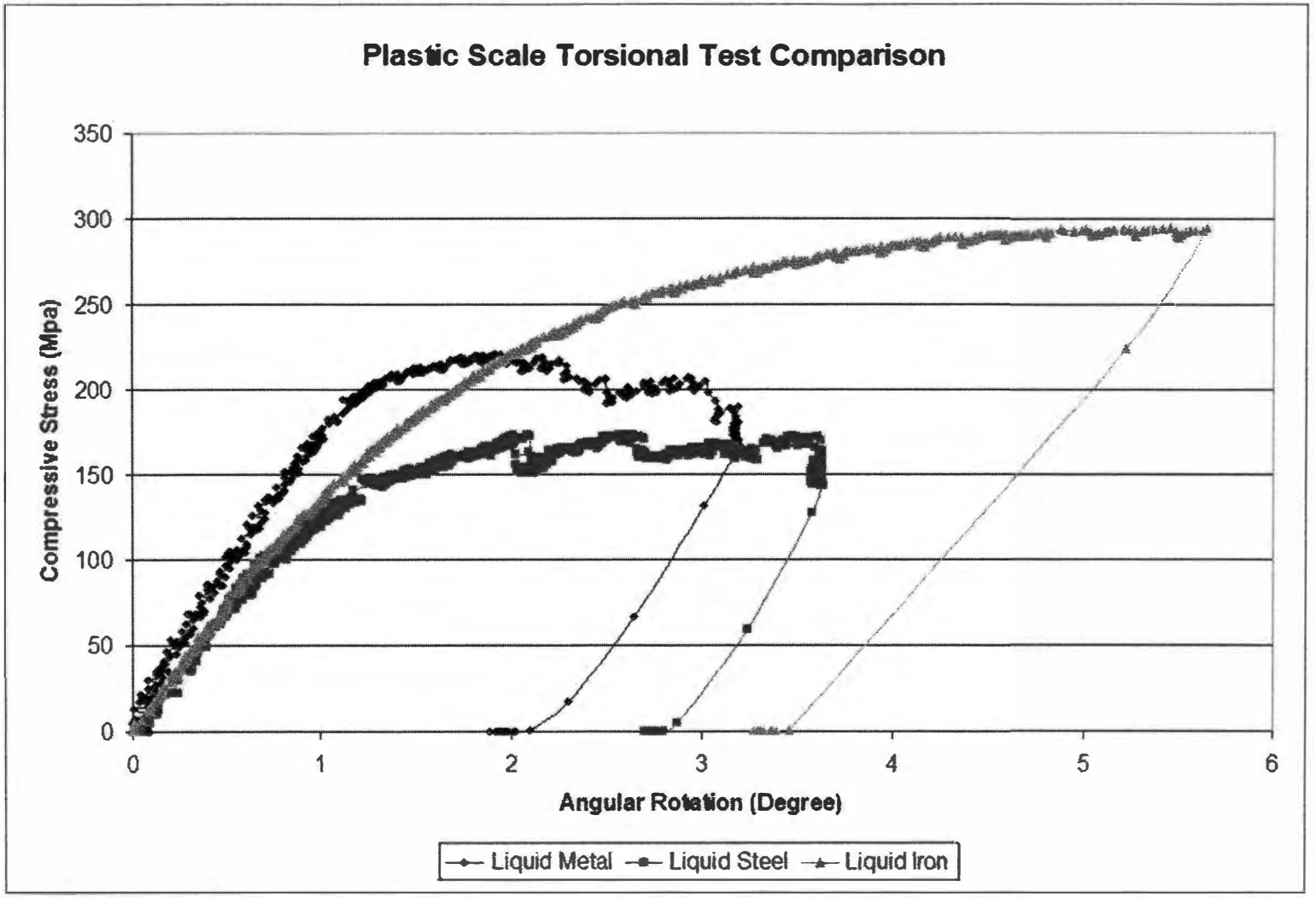


Figure 11. Comparison of Results from Torsion Test.

had the highest strength in both shear and torsional tests followed by Devcon Liquid Metal.

Based on data gathered, it was observed that Freeman Liquid Iron had the highest compressive stress at 295 MPa. This material also had the lowest cost. Devcon Liquid Metal costs $0.06 per cubic centimeter of epoxy while Freeman Liquid Iron costs $0.02 per cubic centimeter of epoxy. Since Freeman Liquid Iron showed highest shear strength and compressive stress, as well as lowest cost, it was chosen as the material to use for further testing.

After completing these tests, scale model boxes were cut open and examined for any signs of failure. Upon opening the boxes, it was observed that Freeman Liquid Steel had large amounts of lobe deflections and plastic cracking. Devcon Liquid Metal also had large amounts of cracking but smaller amounts of deflection. Freeman Liquid Iron showed no signs of cracking and a small amount of deflection. This gave even more confidence that Freeman Liquid Iron was the material that needed to be used in future testing phases.

Also, shown in Figure 12 was shear stress versus angular rotation curve for Freeman Liquid Iron without any lobes. This test was done to find adhesive ultimate stress of this material. Maximum torque at adhesive failure was 2,970 Nm on the 1.27 cm radius of the smooth mild steel scale part. This gave a force of 117,000 N at the surface of the scale part. Dividing this force by surface area, 0.008 m^2, gave adhesive shear strength

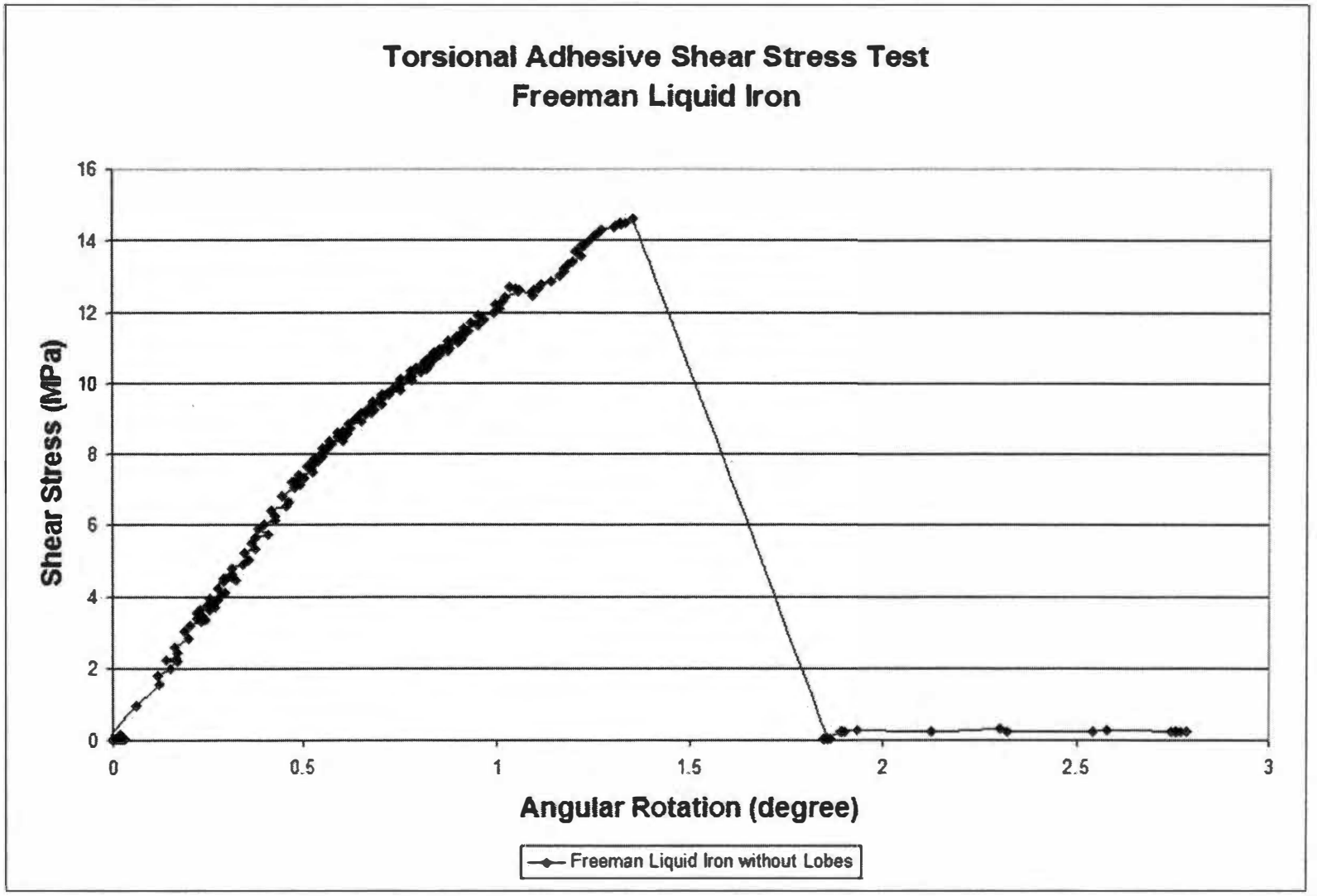


Figure 12. Adhesive Shear Stress Test Results.

of 14.6 MPa. Compared to this materials shear strength of 28.6 MPa, adhesive strength was just over half its shear strength. This shows that adhesion was dependent on surface roughness since its adhesion was much lower than shear strength.

4.2.3 Compression Testing

ASTM Standard D695 testing was conducted on Freeman Liquid Iron material. Mean ultimate compressive strength for this test was found to be 82.1 MPa ±1.2 (mean ± standard error). According to manufacturer's specification tables, this material had a ultimate compressive strength of 86.9 MPa. Student's t test for one sample results found that the measured compressive stress was significantly less than the manufacturer's reported value ($P<0.05$), as was expected (SAS, 2002).

Table 5 shows results from this test. The mean compressive index for semi-confined samples was slightly lower than confined samples. However, these values were not statistically different from each other ($P > 0.9$) (SAS, 2002). This means that compressive strength was not being aided by the metal ring. Thus, the load bearing area was carrying a majority of the compressive stress. And, confinement did not aid material compressive strength. Figure 13 gives a comparison of all three compression test. These curves were the average values for each test along with its variation amount.

Table 5. Compressive Index Comparison for With and Without Metal Ring.

Treatment	Compressive Index (MPa/mm) Mean Separation Values *	Standard Error (MPa/mm)	N
Confined	157.10 A	4.41	8
Semi-Confined	152.34 A	4.41	8

* Mean Separation Values using LSD separation technique

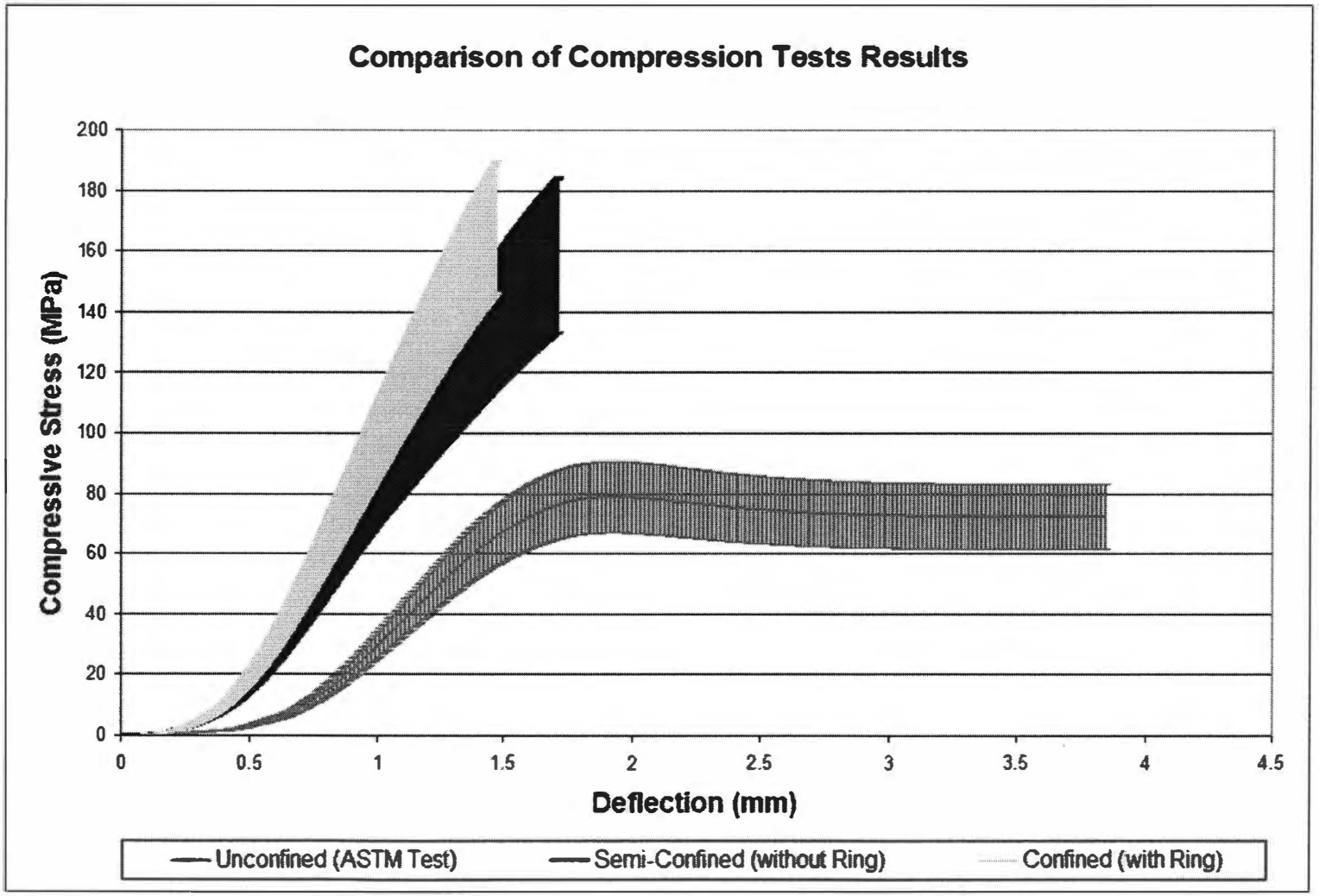


Figure 13. Comparison of Stress versus Deflection for All Compression Tests.

Chapter 5. Phase Two: Axle Housing Testing

5.1 Methods and Materials

Phase 2 testing of axle housing ROPS mount was completed at the Agricultural Engineering Research Center located at Colorado State University in Fort Collins, CO. Parts used for this test were an Allis Chalmers D-17 axle housing and mounting structure injected with engineering plastic. Mounting boxes were fitted around the axle housing where fender mounting flanges were located. Once boxes were in place and sealed, plastic was poured into them through open holes on top of each box. During testing, axle housings and box combinations were subjected to torsional loads until a failure occurred or loading surpassed previous D-17 maximum torque loads.

5.1.1 Mounting Box Preparations

To fit a box around fender mounting lobes, critical housing measurements were needed for the fender lobe sizes and housing radius. Box dimensions were made to fit these critical measurements inside. Boxes were 25.4 cm by 15.2 cm base with a height of 21.6 cm. Wall thickness was chosen to be 0.95 cm (3/8 in.) to reduce possible bending of box sides during testing. Boxes were made into two sections, a top and bottom, which fit around the axle housing and were then welded together.

Having each half of the box produced, the next step was to produce top and bottom plates. Since this test used an I-beam that was bolted to each box, a thick top would be needed to give ample number of threads to hold the I-beam in place. There were eight

bolt holes drilled and tapped along with two 2.54 cm holes drilled though this plate. These two 2.54 cm holes allow for plastic pouring inlets. A bottom plate was made of 1.27 cm thick cold roll steel. Both top and bottom plates were welded to their respective halves. These two assemblies were then placed around an axle housing and welded together. Figure 14 gives a CAD rendering example of the mounting box around an axle housing. To seal areas where boxes and axle housing touched, expandable foam was injected around the housing's diameter. Once this foam cured, engineering plastic was poured into the boxes.

Boxes completely surround the material and were left in place during testing. This aided in eliminating the concern of UV weathering since plastic was incased within the box. However, this study did not evaluate any weathering or temperature effects on the plastic's performance.

5.1.2 Plastic Preparations and Pouring

Freeman Liquid Iron was made from a two part epoxy. This two part epoxy was made by combining an iron oxide-filled resin with a hardener. Total volume of plastic poured into each box was estimated by subtracting the approximate volume of the axle housing inside the box from the known box volume. Estimated total volume of plastic was around 5,500 cubic centimeters for each box.

Engineering plastic chosen had a volume to weight ratio of 0.65 cubic centimeters per gram, meaning each box would hold approximately 9 kg of engineering plastic.

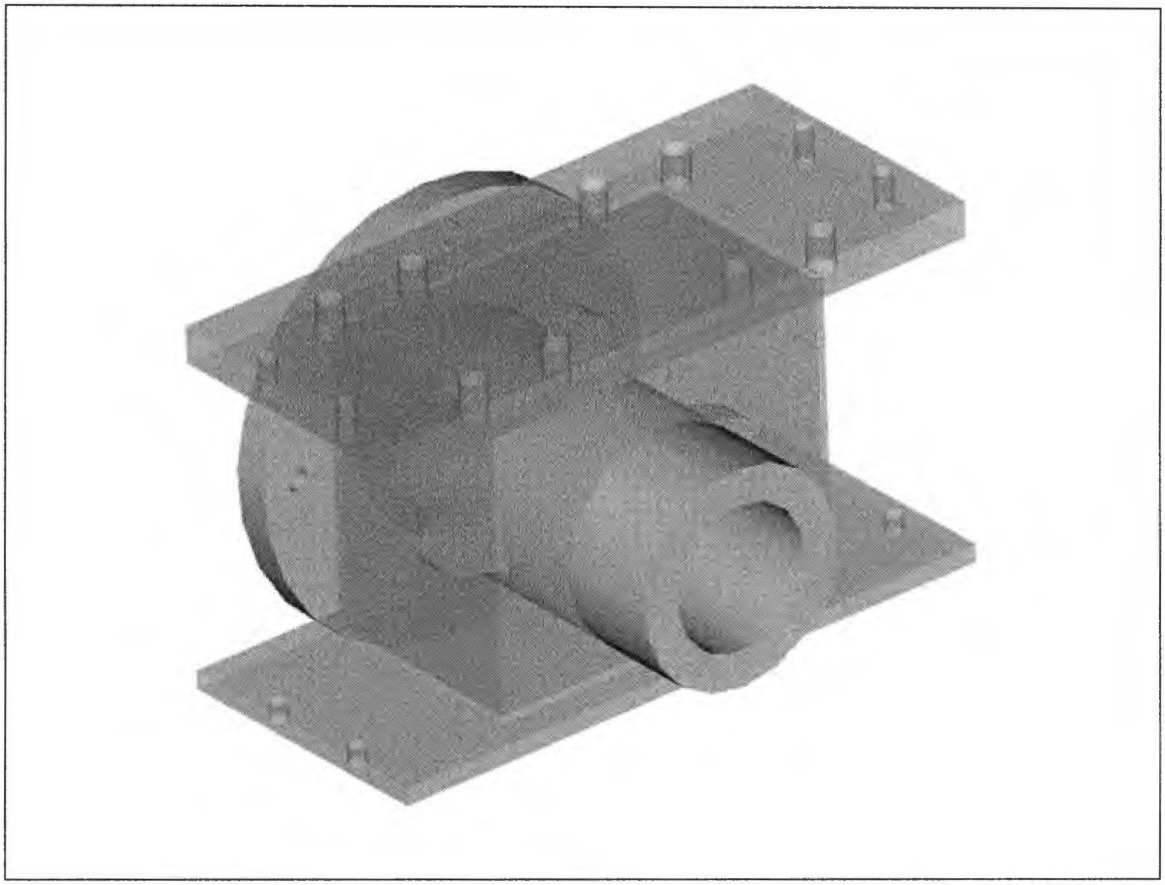


Figure 14. CAD Isometric View of Mounting Box Around Allis Chalmers D-17 Axle Housing.

Following a mixing procedure given by the manufacturer, a mix ratio of 10 parts resin to 2.5 parts hardener by weight, amounts for each epoxy part was weighed out and combined to make 9 kg of engineering plastic. These two parts were mixed with a plunge mixer attached to a large drill. This ensured that resin and hardener were mix together well. Plastic was then slowly poured into the 2.54 cm open holes on top of the box and left to cure for the recommended time. Figure 15 gives examples of the mixing and pouring processes. Once plastic was cured, combined box and axle housings were ready for testing.

5.1.3 Testing Setup

To conduct axle housing torsional load testing, an axle housing's differential end was bolted onto a mounting bracket of the testing apparatus base frame. This was done to prevent movement of the housing relative to testing apparatus. The wheel end was held up with a pin mount (allows rotation, but not translation) by a second mounting structure to reduce bending loads. With an axle housing mounted securely, an I-beam was bolted to the box's top mounting plate. An example of this set up can be seen in Figure 16.

A hydraulic cylinder was attached to the I-beam at 1.98 m above the centerline of the axle housing. This distance was chosen based upon height restrictions caused by roof height of the test facility. The hydraulic cylinder had a piston diameter of 10 cm and a stroke of 60 cm. Cylinder displacement was at a rate of approximately 0.35 cm/s. A 90,000 N load cell was in-line with the hydraulic cylinder to measure loads applied to the I-beam. As the cylinder extended, a potentiometer was used to measure length of

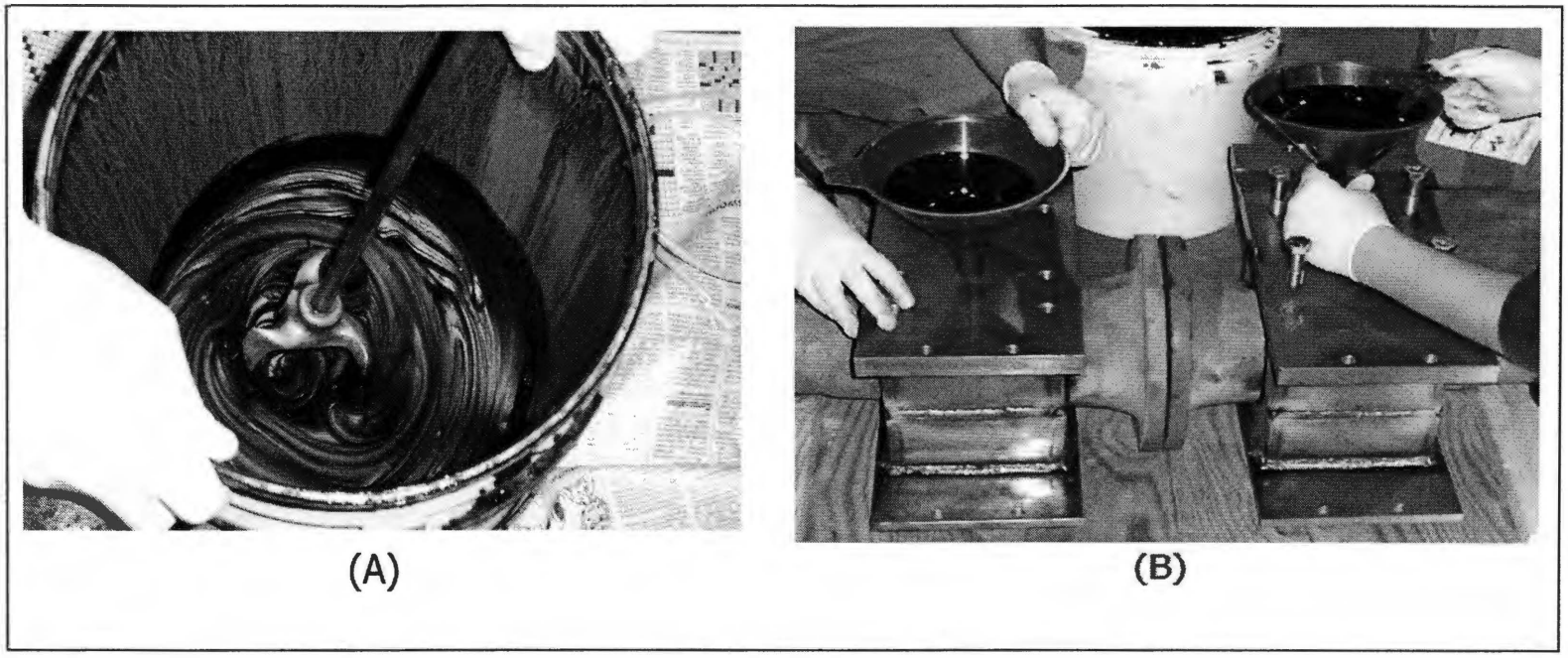


Figure 15. (A) Mixer Combining Resin and Hardener. (B) Pouring Mixed Epoxy into Mounting Box.

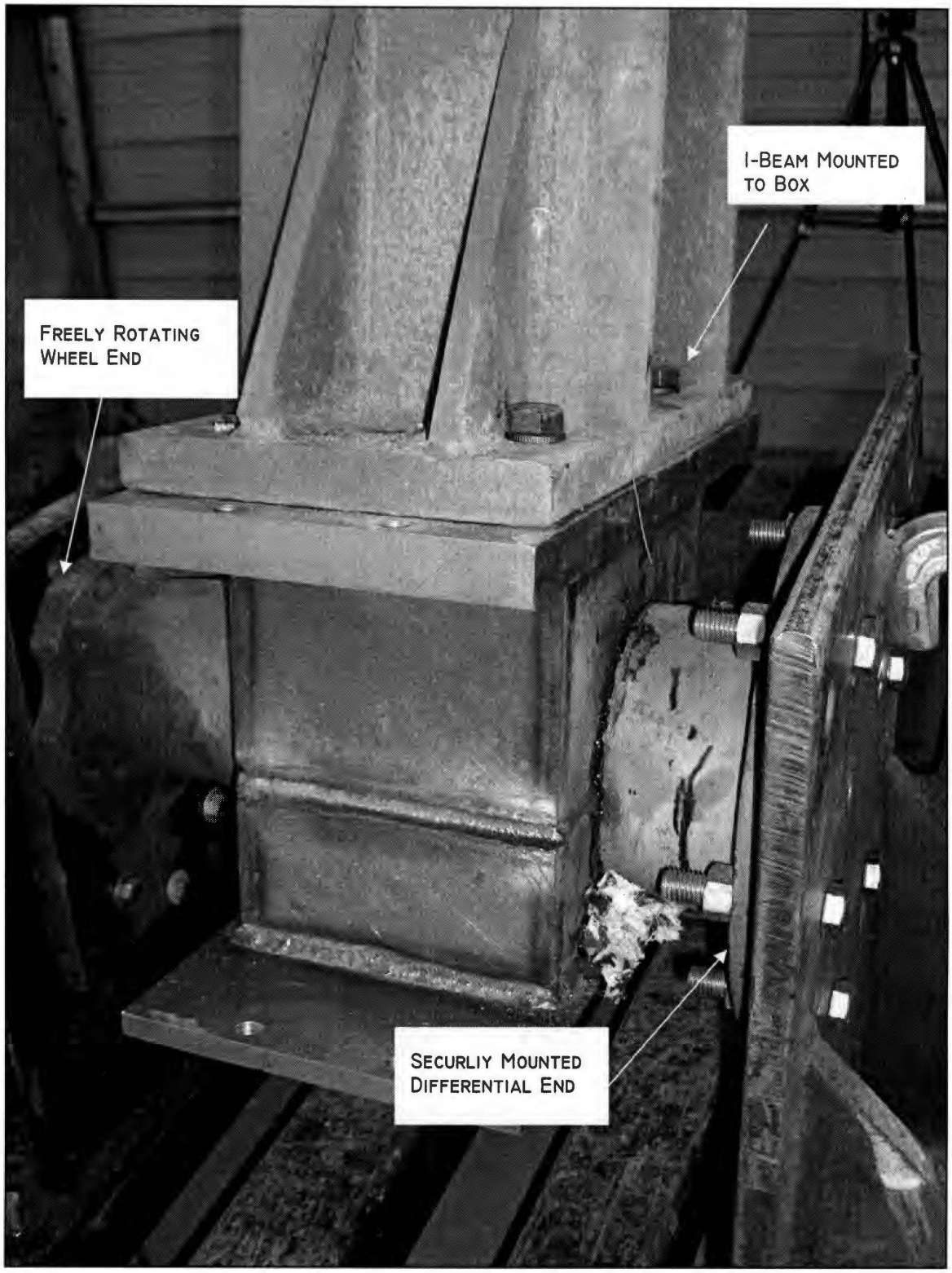


Figure 16. Mounting Box with I-Beam Setup for Axle Housing Strength Test.

extension. Both the load cell and potentiometer measurements were logged into a Campbell Scientific 23X data logger. The logging rate was once every 0.2 seconds. Measurement accuracies for this system were 41 N and 0.1 cm. Figure 17 showed cylinder and I-Beam placements during testing.

During cylinder extension, axle housing and box twisting was seen through axle housing marks and videotape review. After extending until failure or an amount that surpassed loading placed on previous D-17 axle housings, the cylinder was returned to a no load situation. Testing was conducted on 2 identical mounting box structures on 2 different axle housings. Observations were made after each test about the failure or amount of twist.

A mathematical model was used to calculate a compressive stress on fender lobe faces. To calculate this compressive stress, torque values were multiplied by the moment arm length to the center of the lobes to find a force on each lobe. Unlike the squared lobes used during the torsional scale testing of phase one, fender mounting lobes were triangular shaped. This meant that to find the perpendicular force on each lobe face, geometry must be used. Once the perpendicular force was found, it was divided by the lobe face area of 16.1 cm^2. This would give the compressive stress on each lobe face.

Previous torsional testing research was conducted on D-17 axle housing by Ayers (2003). This data was available for comparison with the data collected during this phase of testing. Also, a yield torque, as described by Li (1997), was calculated based upon



Figure 17. Mounting Setup for the Torsional Test of an Axle Housing.

the 0.2% proof offset method. To calculate this value, a torque versus angular rotation graph was needed. A line was then drawn parallel and offset 0.72 degree from the elastic deformation range. A 0.72 degree offset was used since 0.2% of 360 degrees was 0.72. This value was calculated so that design margins could be calculated in later phases.

5.2 Results

After data were collected a peak torque of around 66,500 Nm was found on each test conducted. Each test of the two axle housing plastic mounts was nearly identical when comparing their torque versus axle housing twist curves. These curves also clearly illustrated the change where axle housings went into the plastic deformation range.

Using the peak torque value, compressive stress on the fender mounting lobes was found. When the perpendicular lobe face force was calculated, it was 103,200 N. This gave a peak compressive stress of 62 MPa on each lobe face. Figure 18 shows the loads and load locations calculated in this model. Also given in Figure 19 is a compressive stress versus angular rotation for the axle housing and box combination torsional test.

In Figure 20, results of previous tests that were conducted by Ayers (2003) on D-17 axle housings are compared to results form current testing. When compared to previous testing of axle housing strength, the addition of plastic filled boxes gave an increase in compressive stress with less angular rotation. Maximum stress applied was increased

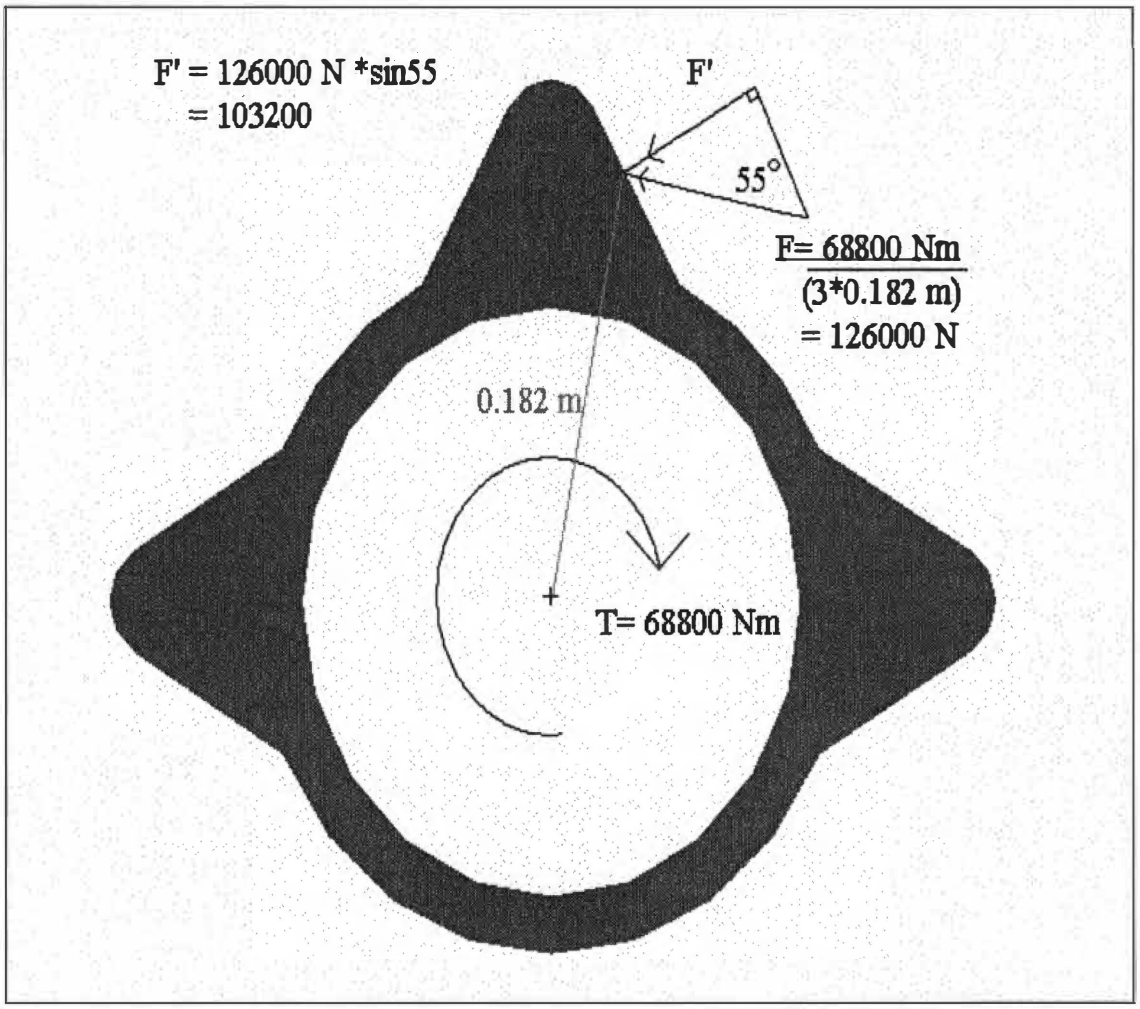


Figure 18. Axle Housing Section View with Force Locations.

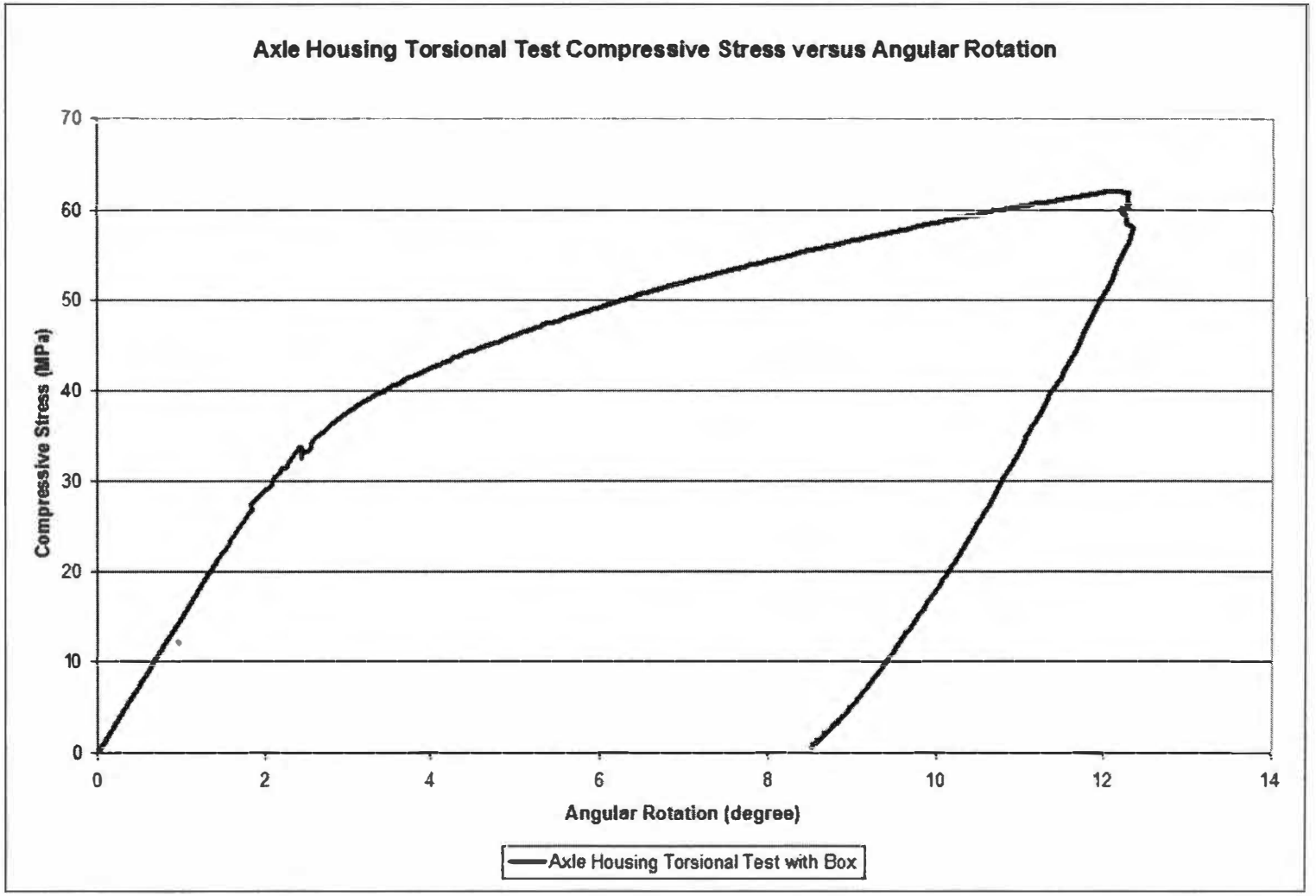


Figure 19. Compressive Stress versus Angular Rotation from Axle Housing Torsional Testing.

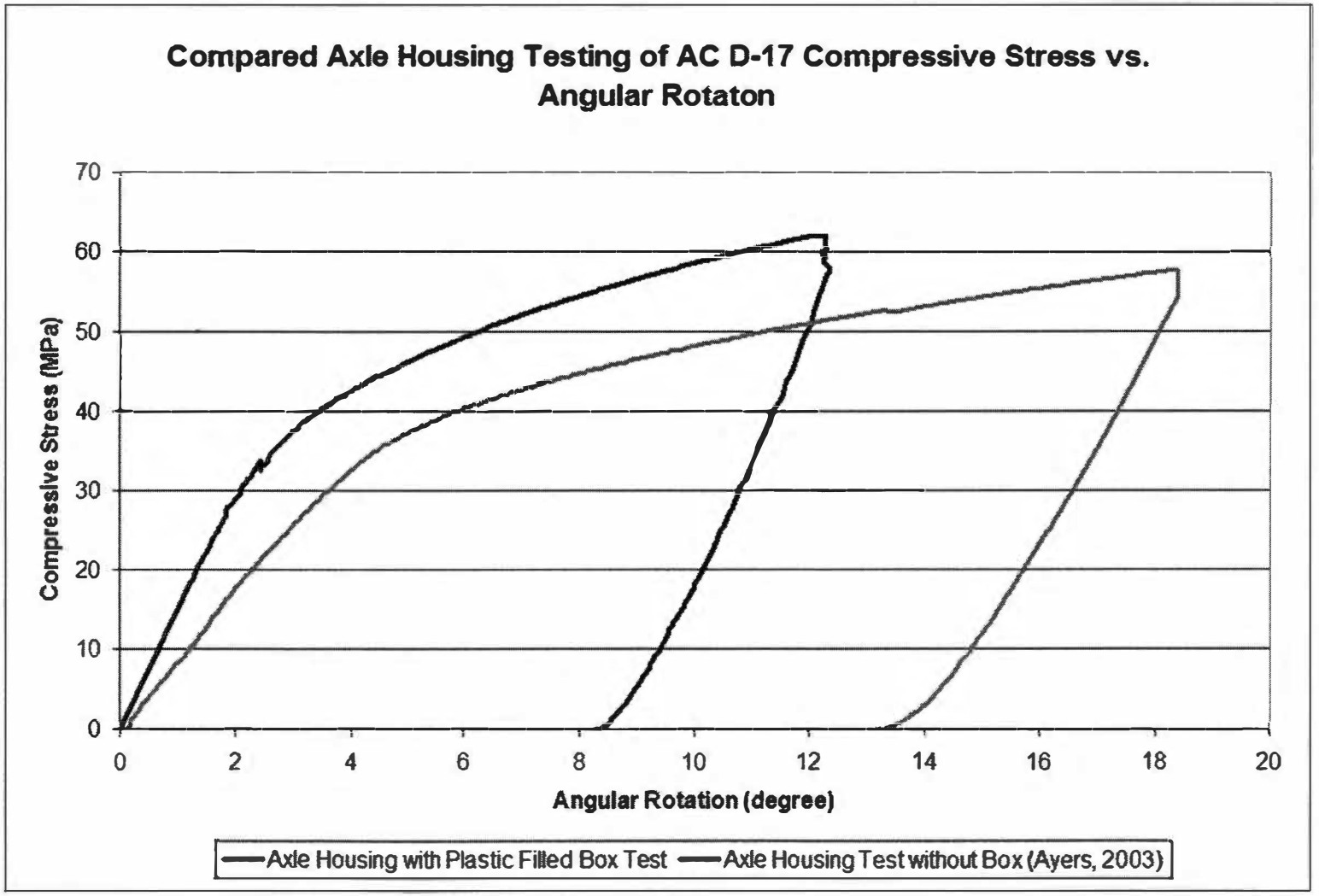


Figure 20. Torque vs. Angular Rotation Curves for Different Test Conducted on D-17 Axle Housings.

from 57.7 MPa to 62 MPa when plastic filled boxes were used. Angular rotations were decreased from 18.3 degrees to 12.3 degrees.

Torque versus angular rotation data for phase two testing are seen again in Figure 21. This figure shows how the 0.72 degree offset was applied to calculate the yield torque. This yield torque value was 44,700 Nm.

Plastic deformation was easily noted since axle housings did not return to their originating state. It was observed that twisting took place in line with fender mounting lobes. When looking down the inside of an axle housing, a cast line easily identified where twisting originated from. Since this twist came from fender mounting lobe's location, it was thought that lobes carried a bulk of loads applied to the housings, as expected. No visible permanent change in box position was viewed. However, if there was an amount of twist, it was minimal when compared to twisting of the housing. This indicates boxes stayed firmly in place and all twisting was due to housing deformation.

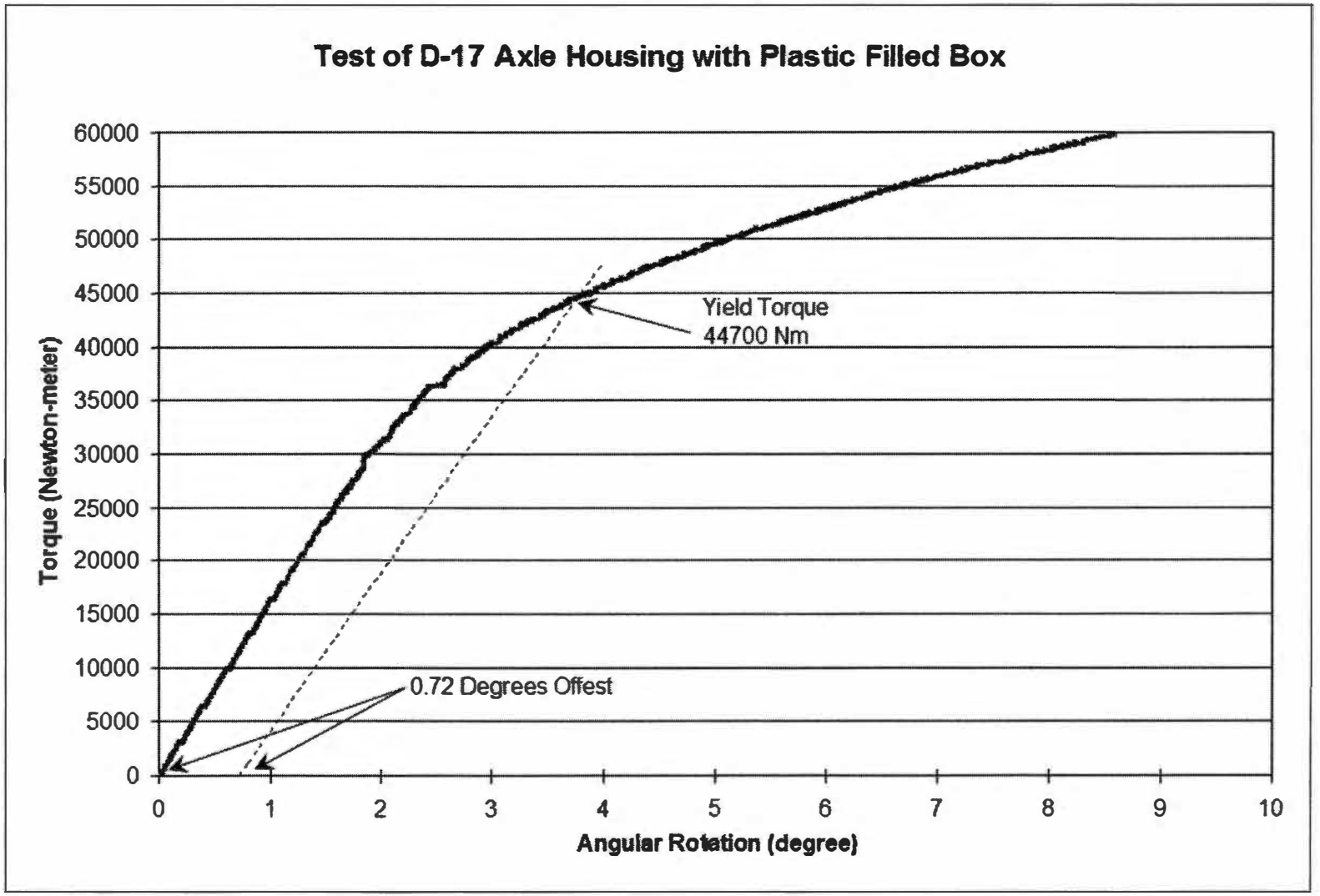


Figure 21. D-17 Axle Housing with Plastic Filled Box Yield Torque.

Chapter 6. Phase Three: Static ROPS Testing

6.1 Methods and Materials

The third phase of testing was also completed at Colorado State University's Agricultural Engineering Research Center. In this phase of testing, axle housing and mounting box combinations were tested to determine if they could withstand loads applied during SAE ROPS standard longitudinal and lateral static testing. The standard that was chosen was SAE J2194 (ASAE, 2004b). To conduct this test, a set of combined axle housings and boxes were attached to a differential housing from a D-17 tractor. A ROPS was then attached to mounting boxes so that loading called for by the standard could be applied.

6.1.1 Mounting Box Setup

Box dimensions for this testing were the same as used in phase two. Top and bottom plates of each half were reduced in thickness to 1.27 cm for this test. This was due to the fact that the ROPS was not bolted to the mounting box, but around it in a compression fit. Therefore plates did not have to be as thick since there were no threads required. During this phase, box halves were not welded together. When two halves were placed around an axle housings, they were compressed together using four bolts (15.875 mm by 35.56 cm grade 8). Expandable foam was again used to seal between each half and around axle housing diameters. With halves compressed and sealed, plastic epoxy was mixed and poured in a similar procedure that was used before.

6.1.2 ROPS Setup

ROPS used for this test was similar to ROPS used in previous D-17 testing. ROPS was of a two-post style that used 12.7 cm by 5.0 cm rectangular tubing with a wall thickness of 0.47 cm for each post and cross member. Posts were angled backward 20 degrees from vertical. ROPS height from top of posts to mounting plates was 161.3 cm vertically. Length of cross member between each post was 99.06 cm.

6.1.3 Testing Setup

Mounting box and axle housing combinations were bolted onto a D-17 axle housing that was securely attached to a test frame. The ROPS was then mounted in a compression fit around the mounting box. The first test conducted was a longitudinal static test. This test called for a continual load to be applied to the ROPS cross member at a rate of 5 mm/s (ASAE, 2004b). Loading continued until failure or energy applied meet what was required by Equation 1 (ASAE, 2004b). Tractor mass for a D-17 was 2,118 kg given by Nebraska Tractor Test #635 (Nebraska Tractor Test, 1957). This mass gave a required energy of 3,200 J when ten percent was added to the equation's given amount. Ten percent was added to compensate for test stand deflections.

The same test apparatus and hydraulic cylinder that was used in phase two was utilized for this test. A potentiometer and load cell again recorded force and displacement. Measurement accuracies for this system were 41 N and 0.1 cm. These values were

logged into a Campbell Scientific 23X data logger. The data logger program also calculated energy applied at each sampling interval along with total energy applied. Once total energy passed 3,200 J, loading stopped and the hydraulic cylinder was return. The test frame set up can be seen in Figure 22. This figure illustrates how a ROPS was mounted to axle housing and box combinations, as well as, the testing set up.

After conducting longitudinal test, a lateral static test was performed. This test is similar to longitudinal testing except loading was applied to upper most part of the ROPS side. Loading rate for the hydraulic cylinder remained at 5 mm/s. Loading would continue until energy applied meet standard requirement given in Equation 2 (ASAE, 2004b). Lateral energy required for a D-17 was 4,100 J when ten percent was added.

To successfully pass there static tests, two events must happen. First, ROPS and mounting system must withstood loads needed to achieve energy requirements. Second, ROPS deflection during testing must not enter the Operator Protective Zone as described in SAE J2194 or expose the Operator Protective Zone to the ground (ASAE, 2004b). Maximum longitudinal and lateral deflections were found by using a model to evaluate exposure. This model from Ayers et al. (1994) took various measurements from tractor and ROPS and then calculated maximum allowable deflections. Maximum longitudinal and lateral deflections were 41.5 cm and 40.0 cm, respectively.

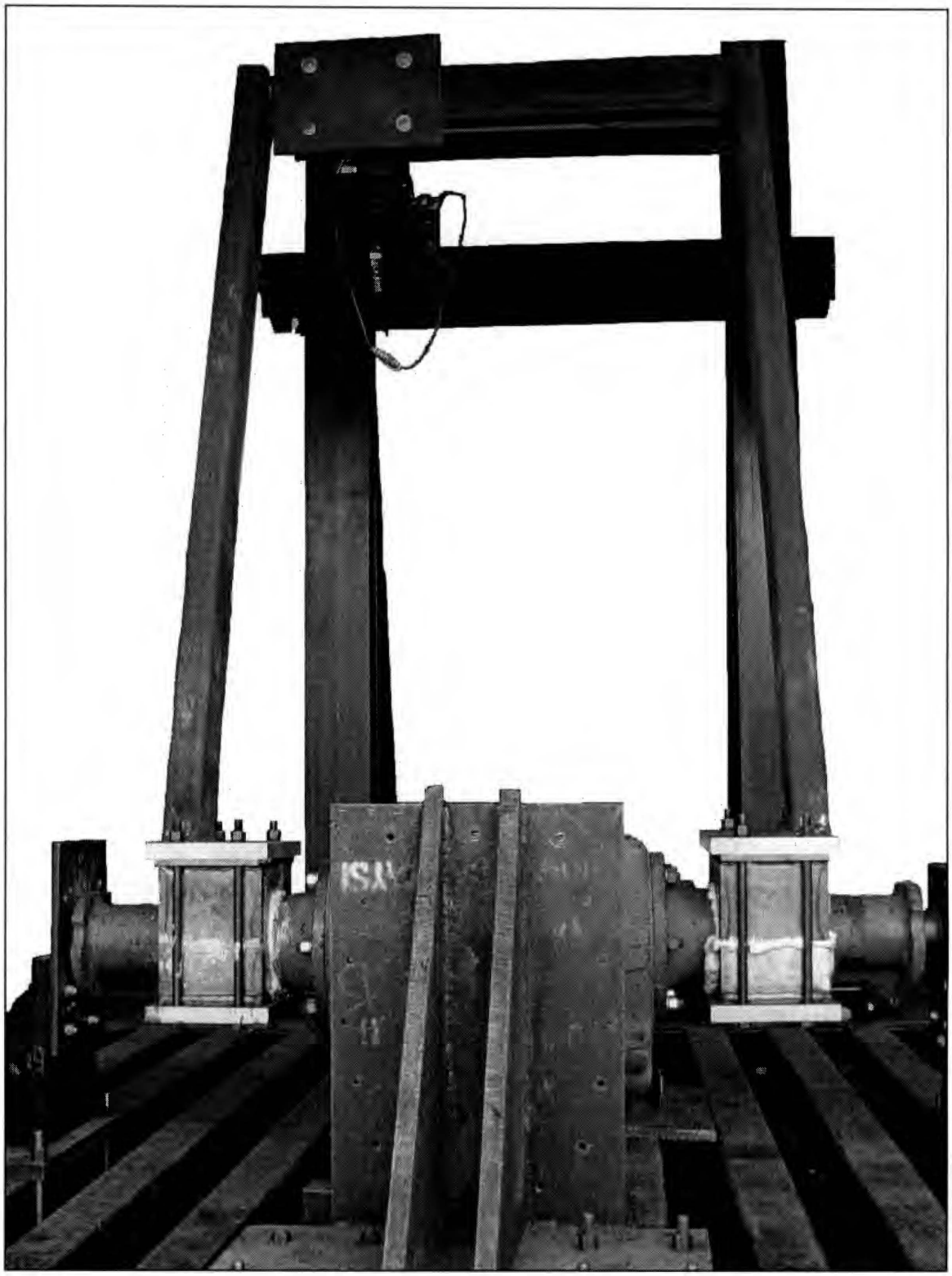


Figure 22. Static Longitudinal ROPS Testing Setup.

After conducting longitudinal and lateral static tests, force and ROPS deflection data was graphed for each test. These graphs were used to find energy applied to the ROPS during static testing. Energy was calculated by finding the area under the force – deflection curves. The ROPS deflection amount when energy applied meets the required energy amount is then compared to the maximum allowable deflections to determine if the ROPS passed the static ROPS tests.

After the ROPS passed static testing, a mathematical model similar to the model in Figure 18 from phase 2 testing was used. This model calculated the compressive stress on each fender mounting lobe face during longitudinal testing. Torque values were multiplied by the moment arm length to the center of the lobes to find a force on each lobe. Geometry was used to find the perpendicular force due to the triangular shaped lobe face. Once the perpendicular force on each lobe face was found, it was divided by the lobe face area of 16.1 cm^2. This would give the compressive stress on each lobe face.

At the moment that longitudinal energy requirements were met, force being applied was known along with horizontal ROPS deflection. ROPS vertical height above axle housing centerline was 1.65 m. During testing though, ROPS height will rise due to a rearward slope of the ROPS. Taking into account this rise in ROPS height, the height at energy requirement was calculated using geometry. This gave a torque moment arm that can be used to calculate the torque when energy requirements were meet. Knowing this value and yield torque value, a design margin was calculated using equation 4 described by Li (1997). This design margin is shown again in equation 7.

$$DM_1 = \frac{YT}{0.5 * ST} \quad (7)$$

where

$DM_1 = design\ margin$

$YT = Yield\ Torque\ (Newton\ meter)$

$ST = Static\ Torque\ at\ energy\ requirement\ (Newton\ meter)$

If the ROPS was not clearly into plastic deformation during static ROPS testing, equation 5 must be used to find design margin. Using this method, as shown in equation 8, a design margin was calculated. If either of these design margin values had gone below 1.0, the axle housing would be into a plastic deformation range. Values below 1.0 could still absorb energy, but axle housings would not return to their normal state.

$$DM_2 = \frac{YT}{0.75 * ST} \quad (8)$$

where

$DM_2 = design\ margin$

$YT = Yield\ Torque\ (Newton\ meter)$

$ST = Static\ Torque\ at\ energy\ requirement\ (Newton\ meter)$

6.2 Results

Results from longitudinal and lateral tests are shown in Figures 23 and 24. In these figures both force and energy absorbed are displayed. During both tests, energy requirements were met. Longitudinal testing demonstrated that energy requirements were met at a deflection of 22.3 cm and a force of 25,000 N. This deflection amount was

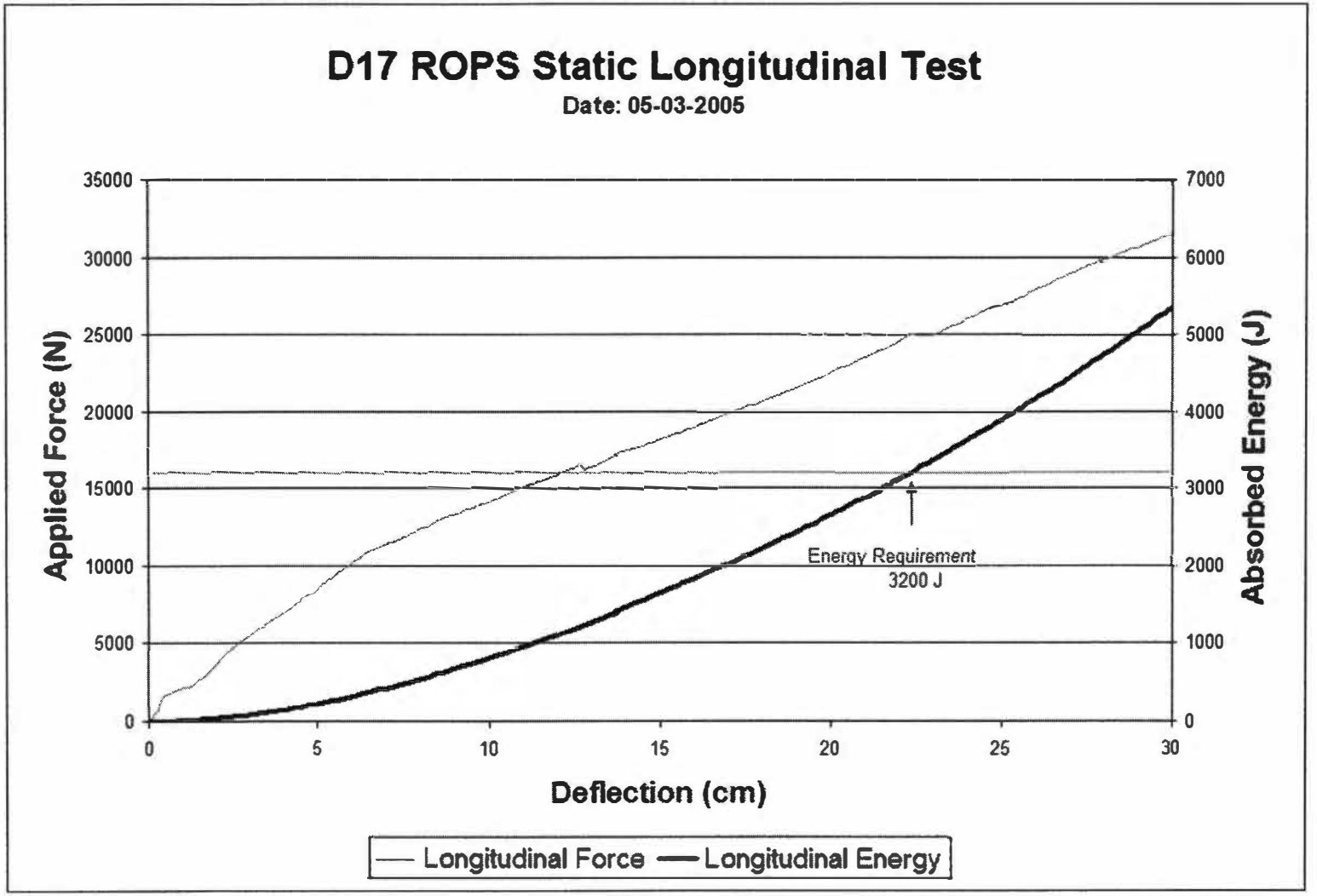


Figure 23. Static Longitudinal Test Results.

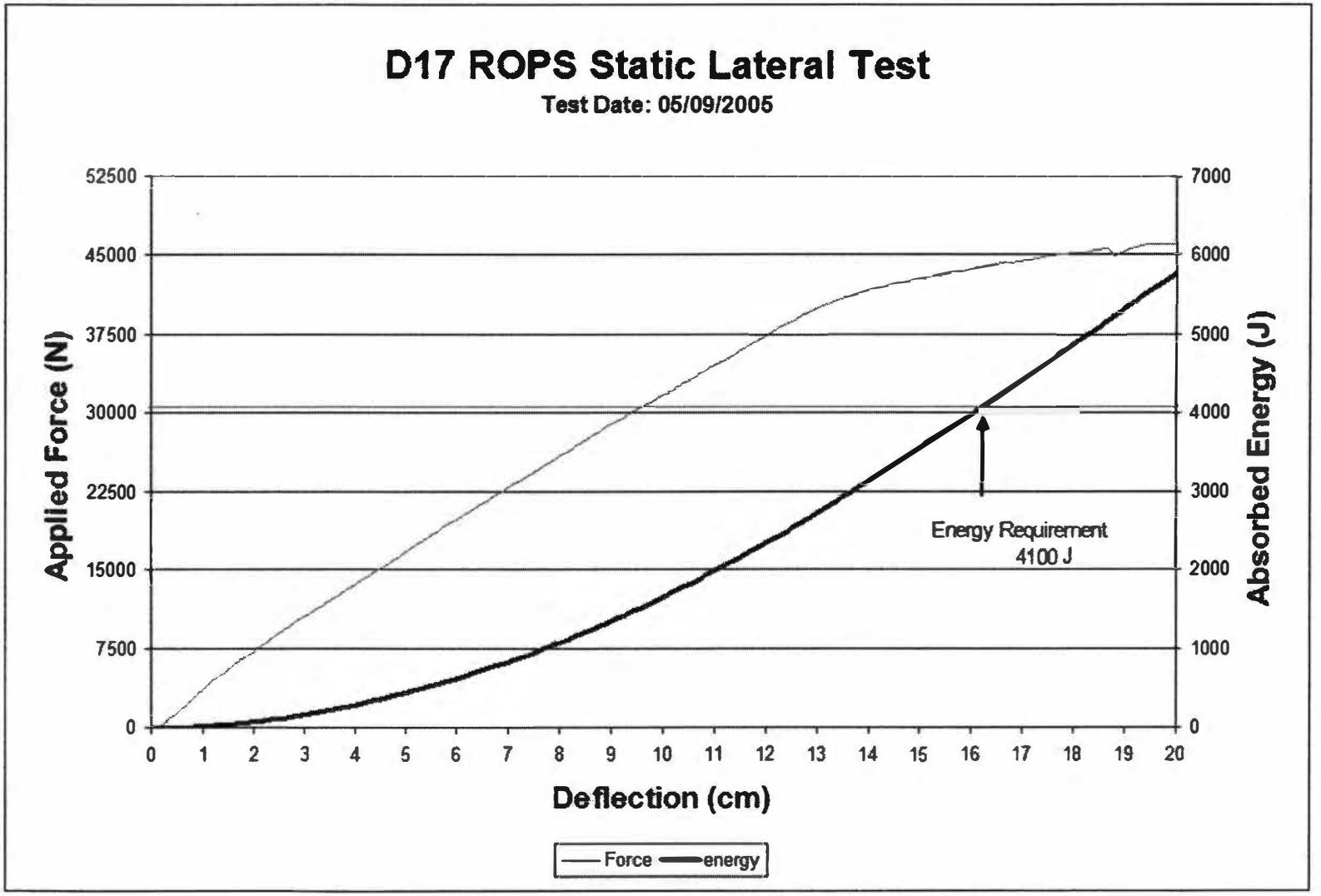


Figure 24. Static Lateral Test Results.

almost half the maximum allowable deflection of 41.5 cm which was needed to meet the intrusion and exposure criteria. Data from lateral test demonstrated that energy requirements were met at a deflection of 16.3 cm and a force of 44,000 N. This too was below maximum allowable deflection of 40.0 cm that was calculated by Ayers' (1994) model.

When longitudinal energy requirements were met, force being applied was 25,000 N. Vertical ROPS height had increased from 1.65 m to 1.71 m due to the rise of the rearward sloping side post. This gave a torque of 42,750 Nm produced on the mounting box. This max torque was used in the compressive model to find that compressive stress on each lobe face of 39.8 MPa. This model was also used to calculate a compressive stress versus angular rotation curve. It should be noted that the model assumed that the torque load was placed onto one axle housing. This graph is shown in Figure 25. This curve is also shown in figure 26 along with the compressive stress versus deflection curve from phase 2 testing. As it is shown in Figure 26, static compressive stress, that is applied to two axle housings, did not reach the maximum that was applied to a single axle housing of phase 2.

Since the torque at static energy requirement was known, a design margin was calculated to be 2.1 based upon DM_1 in equation 7. This value was much higher than the design margin found in previous D-17 testing of 1.6 (Ayers, 2003). Since the ROPS was not clearly into plastic deformation during static ROPS testing, equation 8 was used to calculate a design margin of 1.4. Since this value was above 1.0, it was safe to be used without concern of plastic deformation. It may also be noted that this design

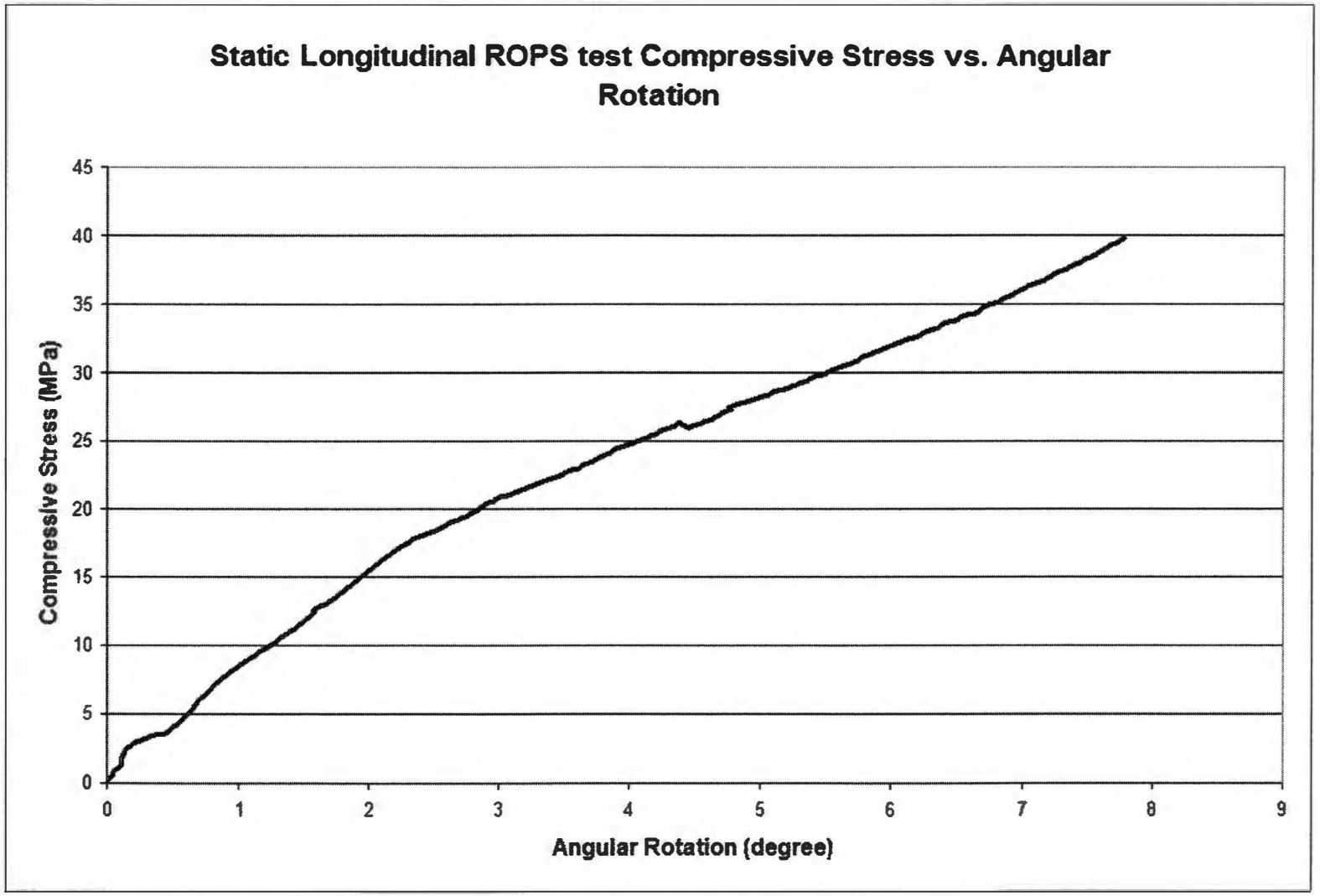


Figure 25. Longitudinal Static ROPS Test Compressive Stress versus Angular Rotation.

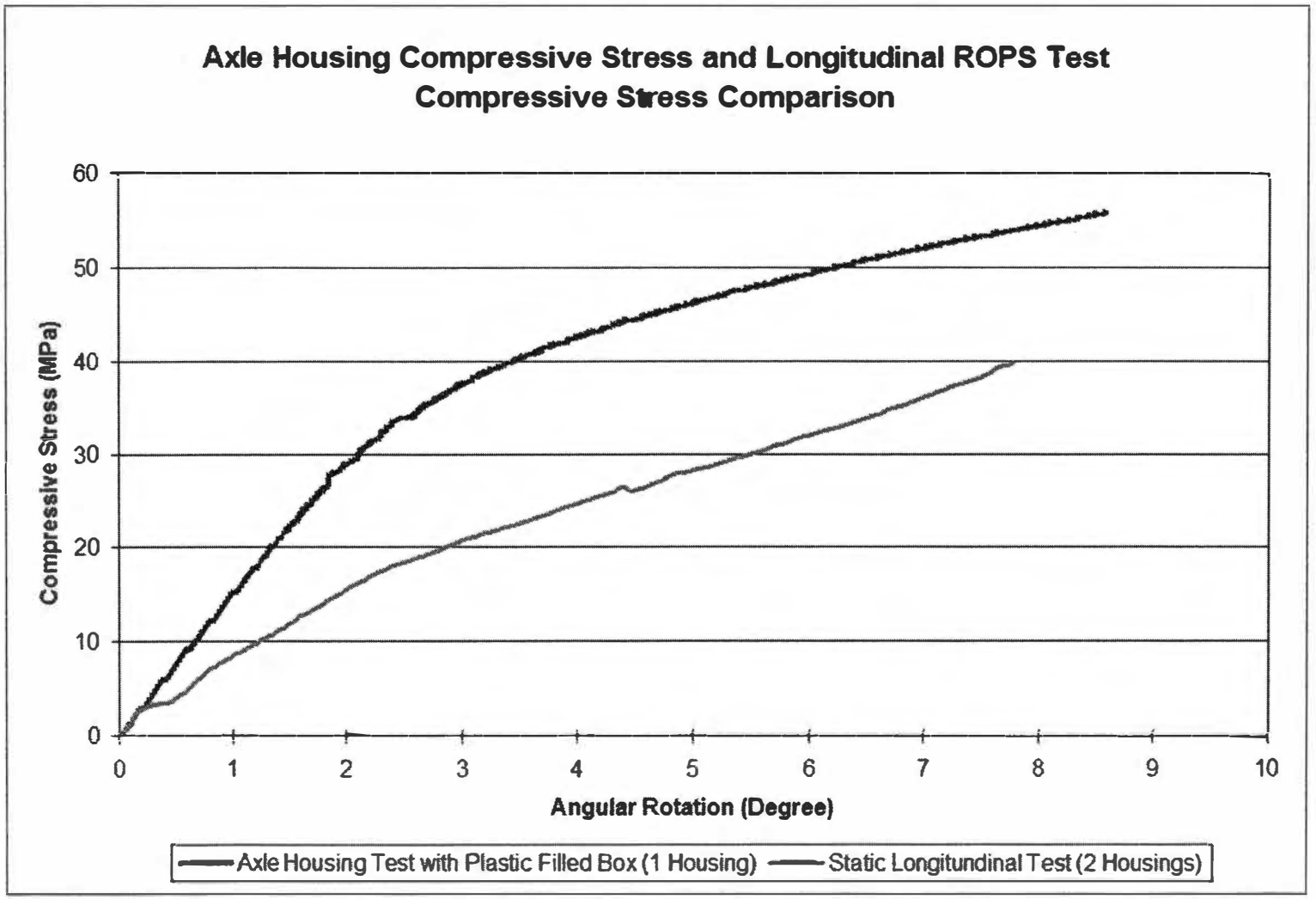


Figure 26. Axle Housing Yield Torque Compared to ROPS Testing Torque.

margin was equivalent to the design margin Li found for Farmall M tractors, (Li, 1997). The Farmall M tractor has commercially used axle housing mounts for ROPS retrofitting with design margin of 1.41.

Observations after testing confirmed that mounting boxes did not show any permanent rotation on the axle housing, just as none was seen during phase two. Also, there was no permanent axle housing twisting like was seen in phase 2. This could be attributed to having loads carried by two axle housing during this test. There were no signs of box failure during either style of static testing. ROPS material welding showed cracking under the high force and energy applied during the lateral test.

Chapter 7. Phase Four: Longitudinal Field Upset Testing

7.1 Methods and Materials

Longitudinal field upset testing was done to validate results from phase three testing. This test was a controlled method for executing a vehicle rollover based upon SAE J2194 requirements. To fulfill objectives of this test, a tractor was driven at a constant velocity, under its own power, onto and up a 60 degree inclined ramp (ASAE 2004b). As the tractor travels up the ramp, a rearward roll was produced. ROPS deflection was measured to ensure that operator's zone was not intruded or exposed by ROPS posts or ground. This was done through use of a slide bar attached to the top ROPS post. Slide bar location was shown in Figure 27. A slide bar was utilized since it measured both elastic and plastic deformations.

A ROPS was attached to the tractor in a similar fashion as during phase three. Two box halves were bolted around an axle housing in their mounting locations and foam strips sealed the seams between each half. Once boxes were in place, engineering plastic parts were mixed and poured into them. After recommended curing of seven days had finished, a ROPS was compression fitted around the mounting boxes. Figure 28 shows a ROPS mounted on to a D-17 tractor. ROPS material and dimensions were identical to those used in static testing. This meant that allowable ROPS deflections would be equal to 41.5 cm for a longitudinal roll.

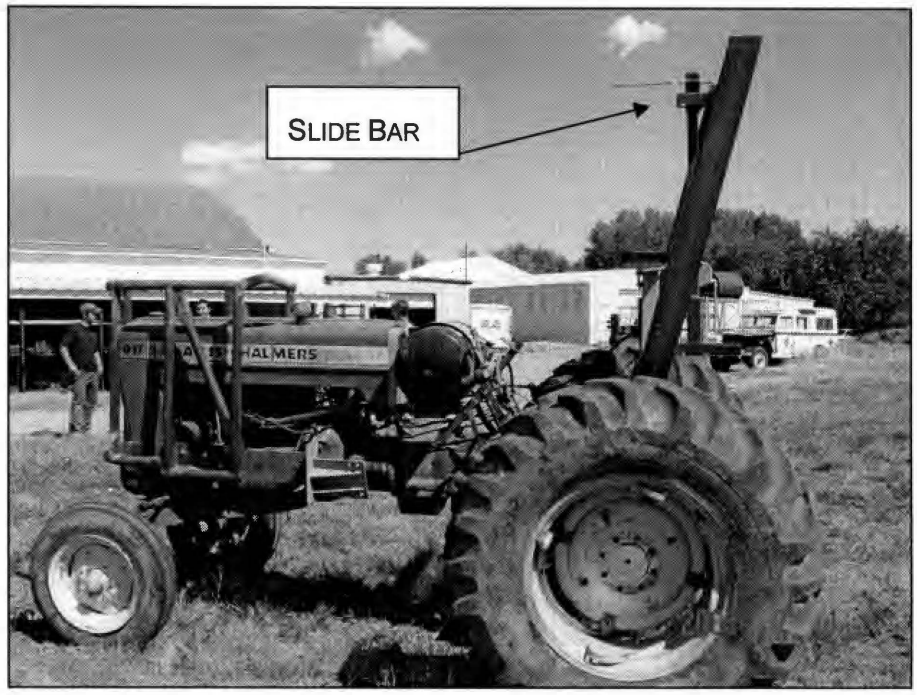


Figure 27. ROPS Mounted on Tractor with Slide Bar.

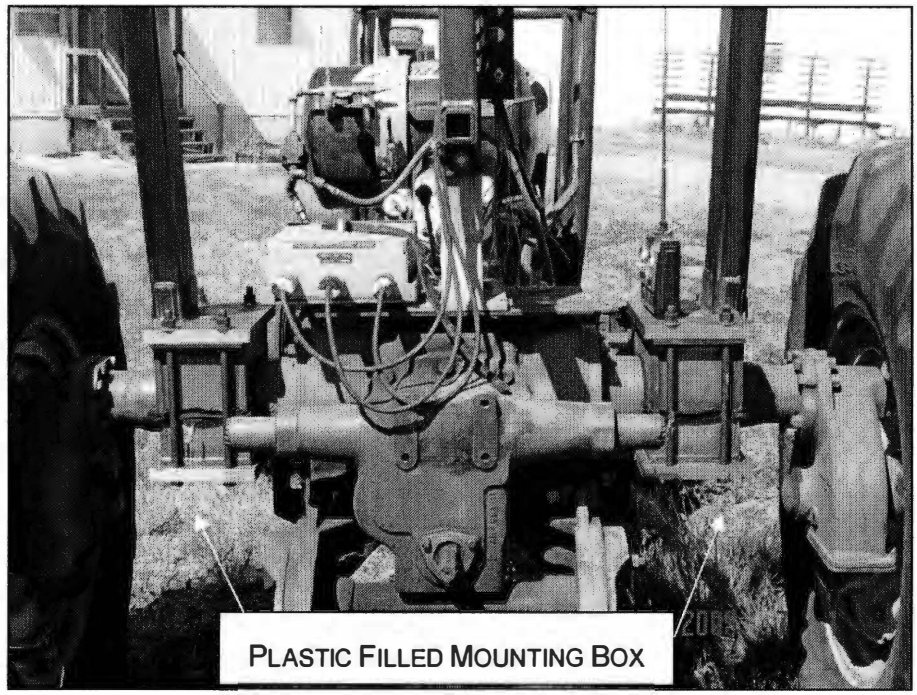


Figure 28. ROPS Mounting Box with ROPS Attached on a D-17 Tractor.

After a ROPS was attached, the tractor was instrumented so that it could be driven by remote control from a safe distance. Clutch and brake pedals were controlled through pneumatic cylinders and steering was controlled through a reversible DC motor. Also, a safety cutoff switch was included to kill the motor if needed.

7.2 Results

With a fully functional tractor in front of the ramp, a gear was selected to reach the desired speed. When the clutch and brake were released, the tractor moved towards and up the ramp until it rolled backwards. Slide bar measurements were taken after the successful roll. Elastic and plastic deformation was found to be 6.0 cm and 2.2 cm, respectively. This gave a total deflection of 8.2 cm, well below the allowable 41.5 cm. This amount was also much lower than the amount of deflection seen in static longitudinal testing (22.3 cm). No permanent box rotation was observed during testing. This indicated that plastic filled boxes had sufficient torsional strength to withstand impact forces produced during longitudinal rolls. Figure 29 showed the tractor after it came to rest on the ground following its rear roll.



Figure 29. Vehicle at Rest After Rear Roll.

Chapter 8. Modeling

Engineering plastic has proven to be effective in providing a ROPS mounting structure and able to withstand forces applied during both SAE standard static and field upset tests. In order to evaluate engineering plastic effectiveness for other axle housing designs, a modeling technique could be applied that uses the plastics properties. Material properties were found during the first phase of testing. Shear strength, compressive index, and adhesion strength were found during that phase of testing. Their values were 28.6 MPa, 157.1 MPa/mm, and 14.4 MPa respectively.

Axle housing surfaces carried various loads due to compression, adhesion, and shear loading. It was assumed that fender mounting lobe faces that twisted into the engineering plastic would support a compressive load. Cylindrical surfaces would support an adhesive load. Sides and top faces of fender mounting lobes would also be subjected to a adhesive loads. Shear loading could occur along the annulus of the lobes. Figure 30 shows the locations of these loading planes. Since loading during phase two testing indicated that rotation was occurring at fender lobe locations, a compressive model was used in previous phases of testing that assumed only compressive loading on lobe faces. Adhesion and shear strength effects were ignored.

8.1 Scale Model Testing Model

From scale model torsional testing results in section 4.2.2, it was known that the compressive model gave a compressive stress of 289 MPa on each lobe face. Knowing

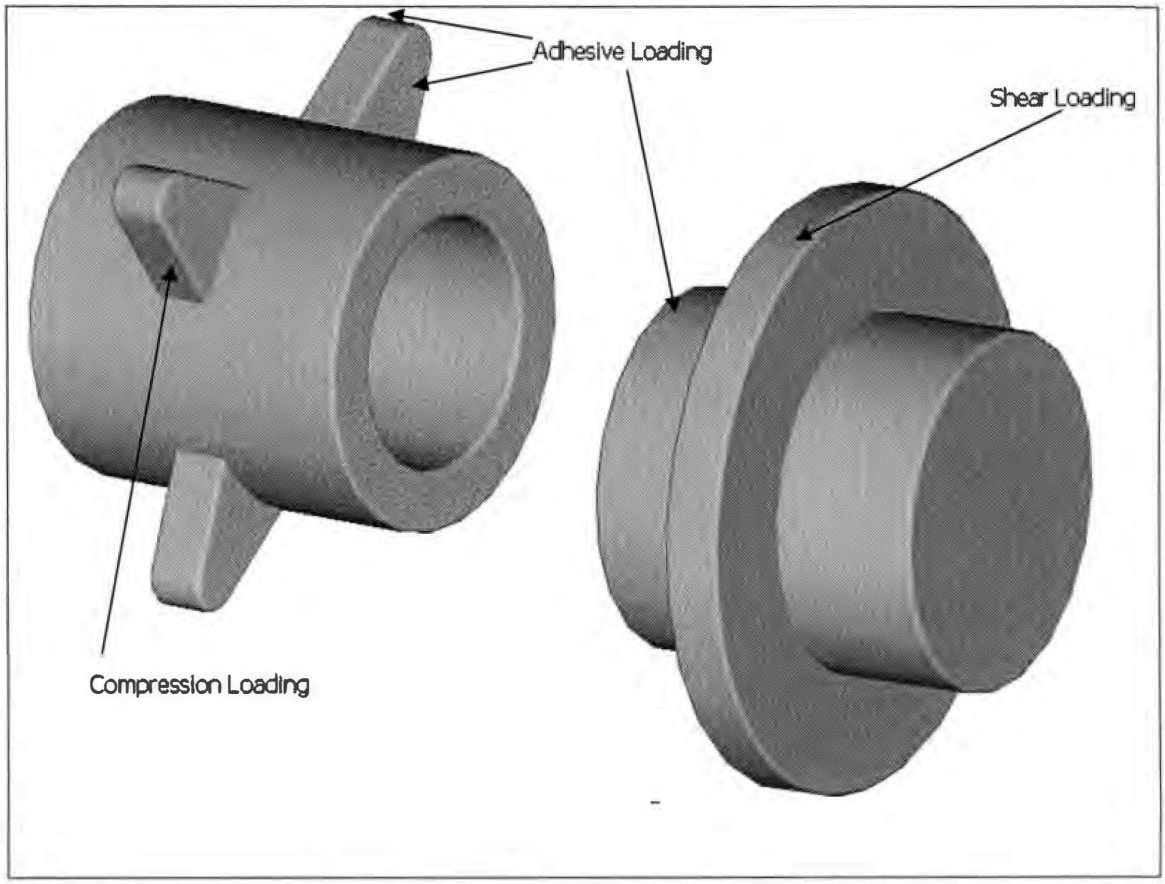


Figure 30. Loading Planes for Mathematical Model.

this compressive stress amount, estimated deflection amounts were calculated using the compressive index found during confined compressive testing. Dividing compressive stress by the compressive index (157.1 MPa/mm) gave a deflection amount of 1.8 mm. This means that each lobe moved into plastic material 1.8 mm when peak torque was applied. This includes both plastic and elastic deflections.

Scale model boxes were cut open and examined for deflection amounts. The lobes had a measured plastic deflection of 0.8 mm into the epoxy material. This can be seen in Figure 31. The calculated combined plastic and elastic deflection was 1.8 mm.

8.2 Axle Housing Testing Model

8.2.1 Compression Model

After examining scale model parts, the next step was to examine full scale axle housing deflections with the compressive model. It was found from section 5.2 that a compressive stress of 64 MPa was on each lobe face. Again using the compressive index, deflection amounts were calculated. This deflection amount was calculated to be 0.4 mm. This too seems to be a reasonable amount of deflection since during testing no permanent twisting of the boxes was visually seen. Figure 32 shows the wheel side of an axle housing before and after testing. As it can be seen, permanent twist was minimal if any.

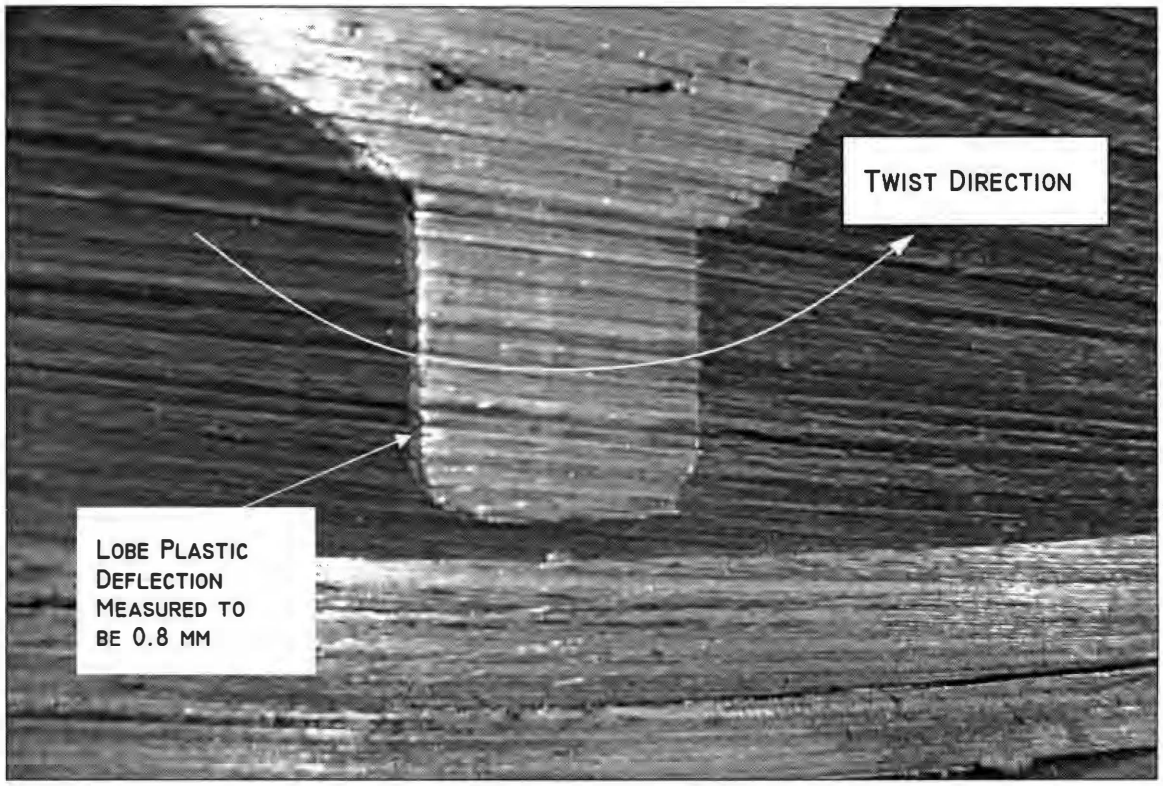
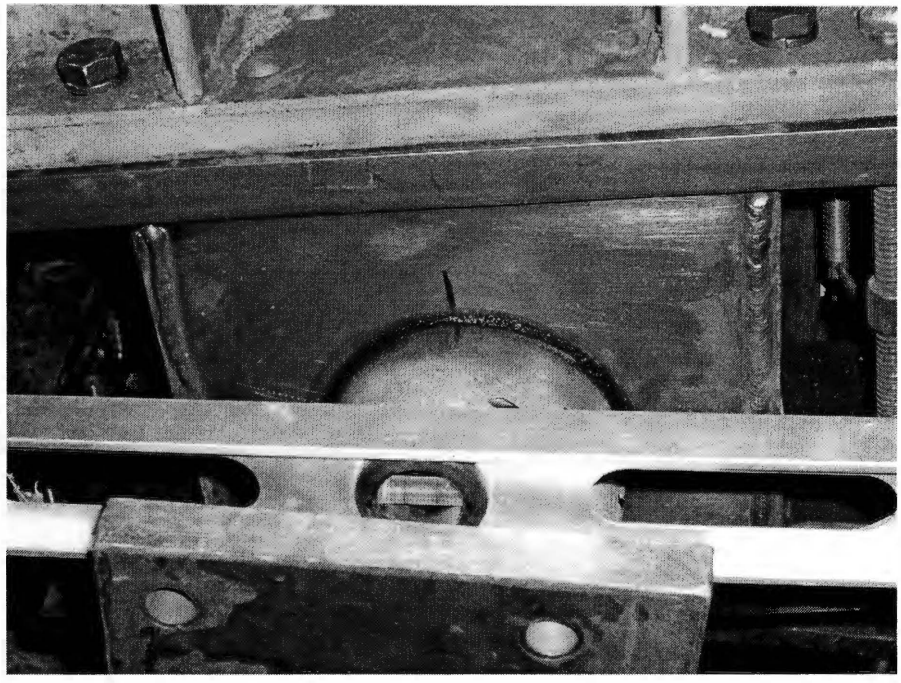


Figure 31. Scaled Box Cut Open Showing Plastic Deflection Amount.



(A)



(B)

Figure 32. (A) Wheel Side Before Testing. (B) Wheel Side After Testing.

8.2.2 Shear/Adhesive Model

This model assumed that shear and adhesive strength would be supporting all loads. In this case adhesion would be supported along the cylinder of the axle and lobe sides. Shearing would be supported through an annulus top caused by the lobes. The shear resistance along the annulus created by the lobes was ignored. Knowing the shear strength and adhesive strength, areas carrying loads, and lever arms to these areas, a total torque at catastrophic failure was calculated to be 169,138 Nm. This was well above the torque that was placed onto the mounting structure during axle housing testing of 68,800 Nm. This means that catastrophic failures of shear and adhesion do not occur.

8.3 Static ROPS Testing Model

8.3.1 Compression Model

During static ROPS longitudinal testing discussed in section 6.2, a compressive stress on each lobe face was calculated to be 39.8 MPa if loading was all placed onto one axle housing. Calculated deflection, using the compressive index value, was 0.25 mm if loading was placed onto one axle housing. If loading was even on both axle housings, the deflection would have been 0.13 mm. If loading was not even and 75% was assumed to be placed onto one axle housing, deflection on that axle housing would have been 0.19 mm.

8.3.2 Adhesion Model

If the lobes had not been present an adhesion only model could be used. Adhesive strength of 14.4 MPa provides a resisting torque of 80,100 Nm on one axle housing. This would have been a sufficient amount to allow for ROPS to pass static testing done on two axle housings. This adhesive strength was based upon the material of smooth steel used in adhesion testing. The adhesive strength could change when it is applied over different materials. Cast iron used on axle housing would likely increase the adhesive strength due to its rougher surface.

Chapter 9. Conclusion and Recommendations

The objectives of this study were to

1) Examine and select an engineering material based upon strength and cost,
2) Conduct axle housing and plastic mounting structure strength testing through torsional tests,
3) Conduct static longitudinal and lateral ROPS testing using a engineering plastic mounting structure, and
4) Conduct field longitudinal ROPS field upset testing using a plastic mounting structure.

Each of these objectives have been completed and evaluated through testing. These tests fulfilled all objectives confirmed that an engineering plastic mounting system is a strong and possibly very useful tool in ROPS mounting.

Plastic's strengths were examined through shear testing, scale model testing, compressive strength testing, and cost examinations. Materials examined were a steel-filled epoxy (Devcon Liquid Metal), a fast curing steel-filled polyurethane (Freeman Liquid Steel), and an iron oxide-filled epoxy (Freeman Liquid Iron). The iron filled epoxy was chosen based upon its higher strength and cost. This material showed the highest shear strength, 28.6 MPa, and maximum scale model compressive stress. Compression testing of this material showed that it had an unconfined ultimate compressive strength of 82.1 MPa and a compressive index of 157.1 MPa/mm. This material was also the least expensive at $0.06 per cubic centimeter of epoxy.

After material selection, a testing of the plastic's strength on an axle housing was conducted by combining an Allis Chalmers D-17 axle housing with a plastic filled box mounting structure. Plastic filled boxes were designed to be a two part box that fit around the axle housing and sealed shut. A D-17 was chosen due to this tractor still having a high number of estimated tractors in use, 41,000, with currently no commercial ROPS available for this tractor. And, previous studies were conducted that show D-17 axles could withstand ROPS loading (Ayers, 2003).

Torsional axle housing and plastic filled box tests revealed that the torsional strength of the ROPS mounting structure and axle housing combination had higher strengths with less twisting than the axle housing alone. Peak compressive stress applied was 62 MPa at an angular rotation of 12.3 degrees. Previous testing of a D-17 without a plastic filled box showed a peak compressive stress of 57.7 MPa at an angular rotation of 18.3 degrees (Ayers, 2003). This testing also gave a "yield torque" of the axle housing and plastic filled box combination to be 44,700 Nm.

An engineering plastic mounting structure was tested through static ROPS longitudinal and lateral tests done in accordance with SAE Standard J2194. From these tests, it was found that mounting structure and ROPS withstood energy requirements at ROPS deflections of 22.3 cm longitudinally and 16.3 cm laterally at a ROPS height of 1.65 m. These values are well below the calculated 41.5 cm longitudinal and 40.0 cm lateral maximum allowable deflections needed to keep the ROPS from exposing the operator's zone to the ground. A compressive stress on lobe faces during longitudinal testing was

found to be 39.8 MPa. Table 6 gives a comparison of the compressive stresses on lobe faces from each torsional test conducted.

Also at this point a design margin was calculated and compared to the previous design margin found. Design margin of this system was 2.1 while Ayers (2003) found a design margin 1.6 for the axle housing without plastic mounting structure. This addition of strength to axle housings with plastic filled boxes could aid other older tractors whose housings are currently too weak to with stand loads applied to them during an overturn.

The final test conducted was a longitudinal field upset test. This test placed a plastic mounting structure onto a working D-17 tractor with a mass of 2118 kg. This tractor was then tested in accordance to SAE J2194 field upset test requirements. Results from this test show a passing of the standards requirements since ROPS deflection was 8.2 cm. Again well below the maximum allowable longitudinal deflection of 41.5 cm.

In addition the work outlined through the objectives, mathematical modeling was conducted and showed that even under a compression only condition, compressive stresses inside each box do not reach unconfined ultimate compressive strength. Boxes do not show much twisting on the axle housing during either axle housing testing or static ROPS testing. Adhesive modeling showed that, even without lobes present, torques applied during static ROPS testing would have been resisted.

Table 6. Comparison of Compressive Stresses on Lobe Faces During Torsional Testing.

	Torque Applied (Nm)	Moment Arm to Lobe (m)	Lobe Face Area (m^2)	Compressive Stress on Lobe Face (MPa)
Scale Model Test	6,100	0.035	1.51	289
Axle Housing Test	68,800	0.182	16.1	62
Longitudinal Static ROPS Test	42,750	0.182	16.1	39.8*

* Compressive stress if one axle housing carried all loading.

While this engineering plastic has proven to be useful as a mounting structure for ROPS, further research is needed. Research should be conducted on life expectancy and material compatibility of the engineering plastic. In particular, how does this material handle vibrations caused by use on a tractor? As well as, how does the material react to extreme temperature and cycling temperature changes? Also, more modeling work could be done to examine reactions during a combined loading situation. Finally, a Charpy V-notch test for engineering plastic could be examined, like the one used for ROPS materials, to ensure that plastic materials will be effective at lower temperatures.

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Vita

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