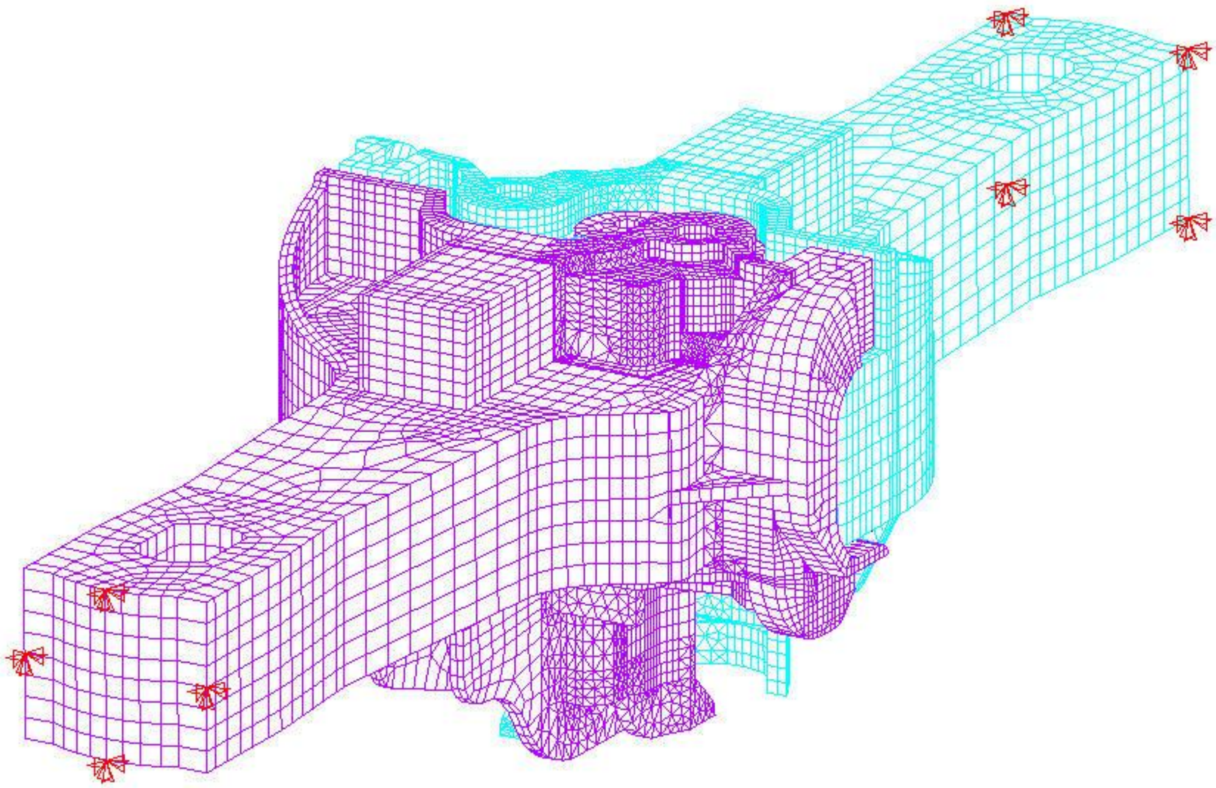
U.S. Department of
Transportation
**Federal Railroad
Administration**

Torsional Stiffness of Railroad Coupler Connections

Office of Railroad
Policy and Development
Washington, DC 20590



DOT/FRA/ORD-10/13

Final Report
November 2010

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Tank cars are required to use double shelf couplers to prevent the coupler of an adjacent car from puncturing the tank head in the case of overrides or accidents. However, the added torsional stiffness of such coupler connections might lead to tank cars 'taking down' adjacent coupled cars in rollover derailments. This project investigated the torsional behavior of freight car coupler connections through analysis and test. Tests included the following coupler combinations: no-shelf to no-shelf, shelf to no-shelf, and shelf to shelf combinations in both clockwise to counterclockwise directions. Coupler specimens of each test run were mounted in a test fixture with one coupler receiving a torque application. Results show a greater torsional stiffness for coupler connections that have couplers with the shelf feature over those that do not. The more shelves that are present in a coupler connection, the greater the transmission of torque, implying that the presence of a shelf coupler or couplers adds to the torsional stiffness of the coupler connection. What is still not clear is whether this stiffness will contribute to, or help prevent, rollovers of adjacent coupled cars. We recommend further study into the potential of stiff coupler connections for propagating car rollovers, specifically through the use of vehicle dynamics models that incorporate the effects car suspension and car structure, in addition to the stiffness of coupler connections.

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298-102

METRIC/ENGLISH CONVERSION FACTORS

ENGLISH TO METRIC

LENGTH (APPROXIMATE)

1 inch (in)	=	2.5 centimeters (cm)
1 foot (ft)	=	30 centimeters (cm)
1 yard (yd)	=	0.9 meter (m)
1 mile (mi)	=	1.6 kilometers (km)

AREA (APPROXIMATE)

1 square inch (sq in, in^2)	=	6.5 square centimeters (cm^2)
1 square foot (sq ft, ft^2)	=	0.09 square meter (m^2)
1 square yard (sq yd, yd^2)	=	0.8 square meter (m^2)
1 square mile (sq mi, mi^2)	=	2.6 square kilometers (km^2)
1 acre = 0.4 hectare (he)	=	4,000 square meters (m^2)

MASS - WEIGHT (APPROXIMATE)

1 ounce (oz)	=	28 grams (gm)
1 pound (lb)	=	0.45 kilogram (kg)
1 short ton = 2,000 pounds (lb)	=	0.9 tonne (t)

VOLUME (APPROXIMATE)

1 teaspoon (tsp)	=	5 milliliters (ml)
1 tablespoon (tbsp)	=	15 milliliters (ml)
1 fluid ounce (fl oz)	=	30 milliliters (ml)
1 cup (c)	=	0.24 liter (l)
1 pint (pt)	=	0.47 liter (l)
1 quart (qt)	=	0.96 liter (l)
1 gallon (gal)	=	3.8 liters (l)
1 cubic foot (cu ft, ft^3)	=	0.03 cubic meter (m^3)
1 cubic yard (cu yd, yd^3)	=	0.76 cubic meter (m^3)

TEMPERATURE (EXACT)

[(x-32)(5/9)] °F	=	y °C

METRIC TO ENGLISH

LENGTH (APPROXIMATE)

1 millimeter (mm)	=	0.04 inch (in)
1 centimeter (cm)	=	0.4 inch (in)
1 meter (m)	=	3.3 feet (ft)
1 meter (m)	=	1.1 yards (yd)
1 kilometer (km)	=	0.6 mile (mi)

AREA (APPROXIMATE)

1 square centimeter (cm^2)	=	0.16 square inch (sq in, in^2)
1 square meter (m^2)	=	1.2 square yards (sq yd, yd^2)
1 square kilometer (km^2)	=	0.4 square mile (sq mi, mi^2)
10,000 square meters (m^2)	=	1 hectare (ha) = 2.5 acres

MASS - WEIGHT (APPROXIMATE)

1 gram (gm)	=	0.036 ounce (oz)
1 kilogram (kg)	=	2.2 pounds (lb)
1 tonne (t)	=	1,000 kilograms (kg)
	=	1.1 short tons

VOLUME (APPROXIMATE)

1 milliliter (ml)	=	0.03 fluid ounce (fl oz)
1 liter (l)	=	2.1 pints (pt)
1 liter (l)	=	1.06 quarts (qt)
1 liter (l)	=	0.26 gallon (gal)
1 cubic meter (m^3)	=	36 cubic feet (cu ft, ft^3)
1 cubic meter (m^3)	=	1.3 cubic yards (cu yd, yd^3)

TEMPERATURE (EXACT)

[(9/5) y + 32] °C	=	x °F

QUICK INCH - CENTIMETER LENGTH CONVERSION

Inches: 0 1 2 3 4 5

Centimeters: 0 1 2 3 4 5 6 7 8 9 10 11 12 13

QUICK FAHRENHEIT - CELSIUS TEMPERATURE CONVERSION

°F	-40°	-22°	-4°	14°	32°	50°	68°	86°	104°	122°	140°	158°	176°	194°	212°
°C	-40°	-30°	-20°	-10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°

For more exact and or other conversion factors, see NIST Miscellaneous Publication 286, Units of Weights and Measures. Price $2.50 SD Catalog No. C13 10286

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Executive Summary

Tank cars are required to use couplers with top and bottom shelves to prevent the coupler of an adjacent car from puncturing the tank head in the case of an undesired uncoupling, coupler override or accident. However, the added torsional stiffness of such coupler connections might be detrimental in certain other cases, especially where cars have leaned over and derailed, for example, low-speed derailments in classification yards. In such a scenario, a car that has derailed might “take down” an adjacent car with it, because of a stiff coupler connection.

Similarly, the transmission of torque from a rail car to the next coupled car as a result of a disturbing force such as high winds could result in “chain” derailments. In revenue service, strings of stationary, coupled tank cars equipped with double-shelf couplers have been reported rolling over on more than one occasion in a chain reaction as a result of wind force. This phenomenon propagated up to the point in the consist where no-shelf couplers were located.

In contrast, it is possible that in some scenarios, a mated shelf coupler may help resist and inhibit an adjacent car from rolling over. It is possible that for every “domino” effect multicar rollover, there are also some potential rollovers that are eliminated because of the shelf-type coupler.

This study evaluated the behavior of freight car coupler connections in regard to the connections tendency to transmit torque from car to car. The goal of this study was to determine the torsional stiffness characteristics of the coupler connections based on the presence of coupler shelves considering multiple combinations of coupler types. These combinations would include all those possible with no-shelf, shelf, and a combination of shelf and no-shelf couplers connected. Both clockwise and counterclockwise rotations were performed to ensure contact at both top and bottom shelves.

An analytical model was developed from a coupler connection using two F-type couplers and analyzed using the Finite Element Method. The torque-deflection results of this model were used as a comparator to controlled laboratory testing using actual couplers.

A physical test bed was designed and constructed for mounting two connected coupler test specimens and for actuating one with an applied and controllable torque. A vertical offset of 5 inches was provided between mating couplers to simulate the drop of a derailed car. Appropriate instrumentation was used to record the torque and the resulting angular displacement.

Results of the physical testing show that a significant difference exists in the torsional stiffness of the coupler connections depending on the combination of coupler types used in the connection. Generally, the more shelves that are present in a coupler connection, the greater the transmission of torque. In other words, a coupler connection between two couplers equipped with top shelves was the stiffest, torsionally. Conversely, a coupler connection between two nonshelf couplers was the most torsionally flexible. The combinations that included only one shelf coupler also showed less deflection than the combination without any shelves. This reveals that the presence of a shelf coupler or couplers will add to the torsional stiffness of the coupler connection.

The question as to whether shelf couplers add to the torsional stiffness of coupled connections has been answered in this study. What is still not clear is whether this stiffness will contribute to, or help prevent, rollovers of adjacent coupled cars. Therefore, we recommend further study into the characteristics of the coupler connection and its propensity of propagating car rollovers.

Through the use of the torque-displacement relationships established in this study, vehicle dynamics simulations should be performed using the appropriate dynamics software, such as Vampire. Such analyses can capture more variables likely to affect the outcome such as those in the car suspension and car structure. Analytical models of multiple rail cars coupled together can be constructed and analyzed to determine the actual propensity of one car to rollover, based on the types of couplers used.

1. Introduction

1.1 Background

Tank cars are required to use couplers with top and bottom shelves to prevent the coupler of an adjacent car from puncturing the tank head in the case of an undesired uncoupling, coupler override or accident. A tank head puncture could result in uncontrolled lading release leading to serious consequences, especially in the case of hazardous materials (hazmat). Certain hazmat commodities also require the use of head shields in addition to shelf couplers.

Although such shelf couplers are beneficial in cases where potential uncoupling and coupler override might occur, the added torsional stiffness of such coupler connections might be detrimental in certain other cases, especially where cars have leaned over and derailed, for example low-speed derailments in classification yards. In such a scenario, a car that has derailed might take down an adjacent car with it, because of a stiff coupler connection.

Similarly, the transmission of torque from a rail car to the next coupled car as a result of a disturbing force such as high winds could result in “chain” derailments. In revenue service, strings of stationary, coupled tank cars equipped with double shelf couplers have been reported rolling over in chain reaction as a result of wind force, on more than one occasion. This phenomenon can propagate up to the point in the consist where no shelf couplers were located.

Thus, examining the torsional behavior of shelf couplers is necessary to evaluate whether there is an increased potential for derailment (if any) of cars that use shelf couplers as compared with cars that use nonshelf couplers.

1.2 Objectives

The effect of shelf coupler torsional rigidity on the derailment potential of hazmat tank cars was investigated (i.e., to investigate whether a derailed tank car could take down an adjacent car). The first step in the process was to establish whether shelf couplers were significantly stiffer than nonshelf couplers, and if so, by how much. The stiffness would be studied by using appropriate analytical models and controlled laboratory testing of actual coupler designs, as appropriate. The results of this study will help the tank car industry evaluate the risks and benefits associated with using shelf couplers on hazmat tank cars.

1.3 Project Plan

The overall plan of action for the study was as follows:

1. Develop analytical models to study and evaluate the relative torsional rigidity of shelf couplers. The modeling and analysis would quantify the extent to which shelf coupler designs could contribute to increased derailment potential.
2. Evaluate the torsional stiffness of shelf couplers in controlled laboratory tests using actual couplers. A suitable test bed for the test effort would be designed and implemented, including suitable instrumentation to measure the torque applied and the corresponding rotations.
3. Compare the analytical results against the laboratory tests to verify reasonableness.

4. Prepare a suitable test/evaluation plan for additional work, if the analytical or physical evaluations indicate a safety implication.

This report describes the evaluations conducted and the results derived.

2. Analytical Model

2.1 Finite Element Analysis Model

In the early stages of this investigation, a virtual three-dimensional model was developed using an F-type coupler geometry, because at the time, this was the only detailed geometry available for modeling purposes. A model of two coupled F-type couplers was created. The basic geometry was developed using Pro/E software (see Figure 1 and Figure 2). This model was exported to Ansys® for meshing. The model was constructed using structural solid elements with contact surfaces defined at all possible contacting surfaces between the mated couplers. The contact elements chosen were three-dimensional surface-to-surface type. The shank end of one coupler was constrained, whereas the shank end of the mating coupler had varying torques applied. These torques were produced by applying a predefined angular displacement on the actuated coupler shank (see Figure 3 and Figure 4).

The development of this model and its successful convergence to produce meaningful results was a very lengthy process. The first attempts utilized a mesh of tetrahedron elements that produced too large of a model if an acceptable mesh refinement was used. A solid mesh of brick (six sided, eight nodes) elements was then constructed for the complex shape of the F-type coupler. This process necessitated the manual construction of many elements individually. Many trials were required in refining the mesh on the complex contoured surfaces of coupler knuckles and mating surfaces before successful runs were possible with the contact elements used. The double convex–concave surfaces at the knuckle interfaces contributed to the complexity of this model.

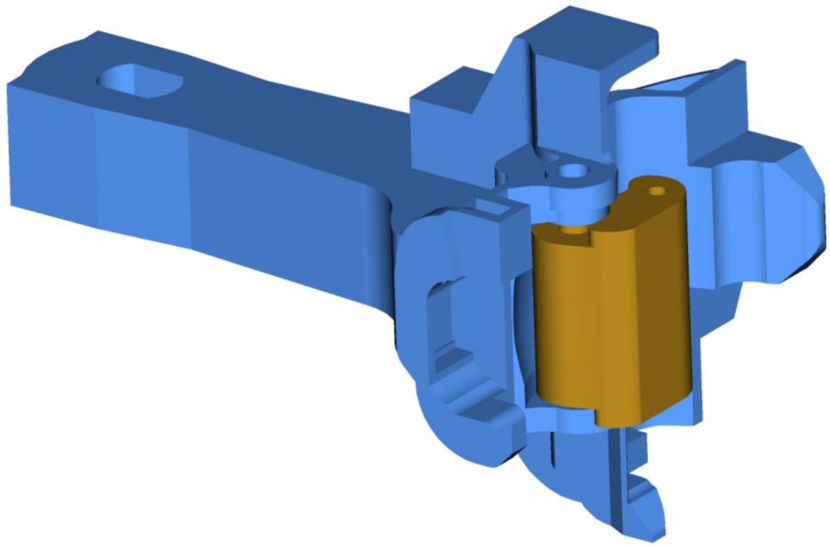


Figure 1. F-Type Coupler Model

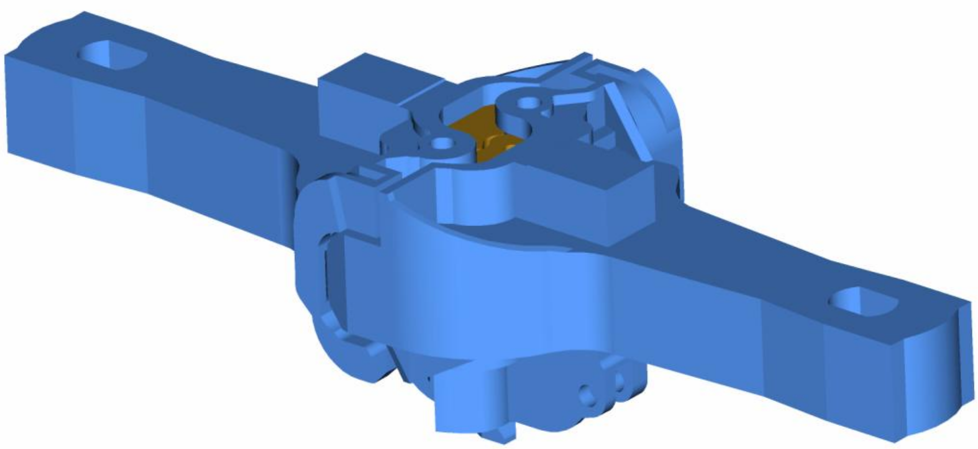


Figure 2. Coupled F-Type Couplers

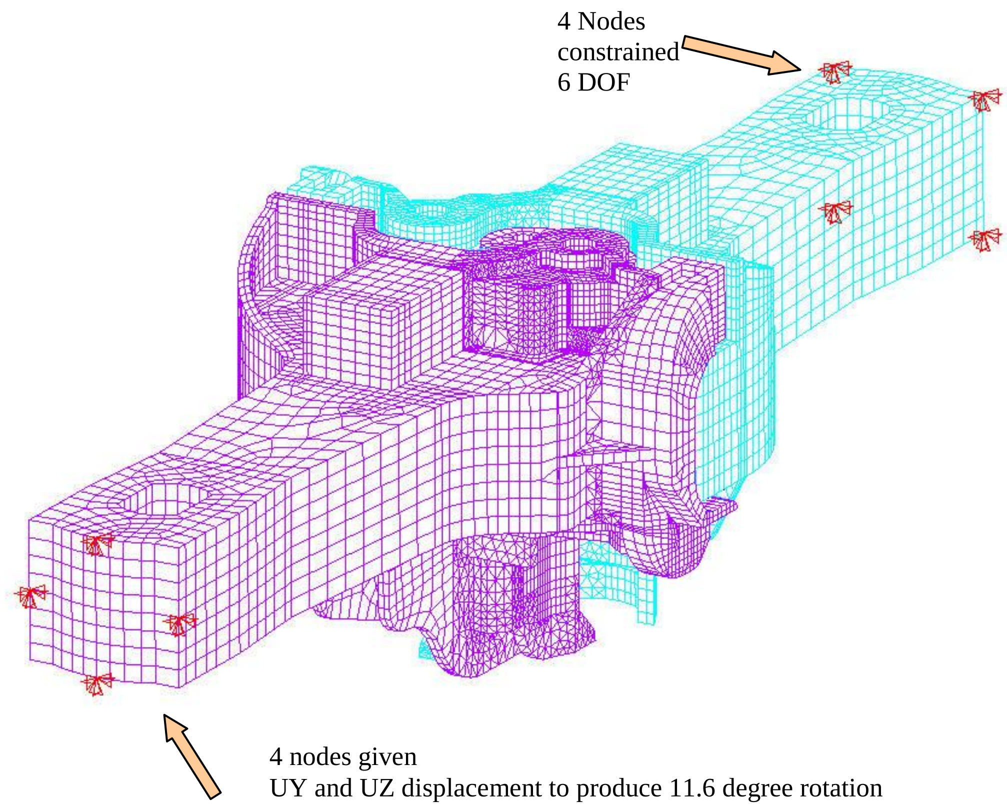


Figure 3. Constraints

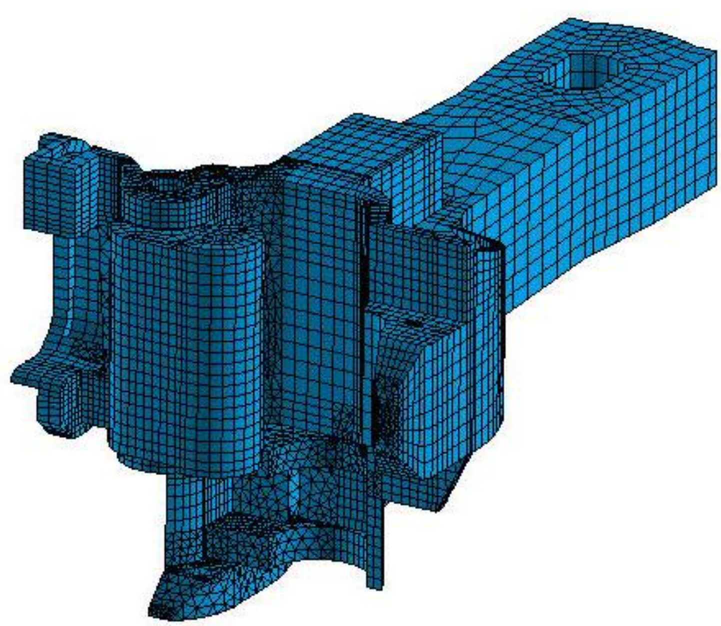


Figure 4. Mesh – Single Coupler

2.2 Finite Element Analysis Results

A relationship between angle of coupler deflection (at the rear end of the actuated coupler shank) and torque was obtained and shown in Figure 5. The plot reveals a mostly linear relationship with a slope of 12,337 pounds per feet (lb-ft)/degree by using liner regression along the inclined linear portion of the curve and about 4.8 degrees of rotation at 40,000 lb-ft of torque. The actuated coupler was displaced in the counterclockwise direction. The displacement contour plot and the von Mises stress contours are displayed for a single coupler in Figure 6 and Figure 7.

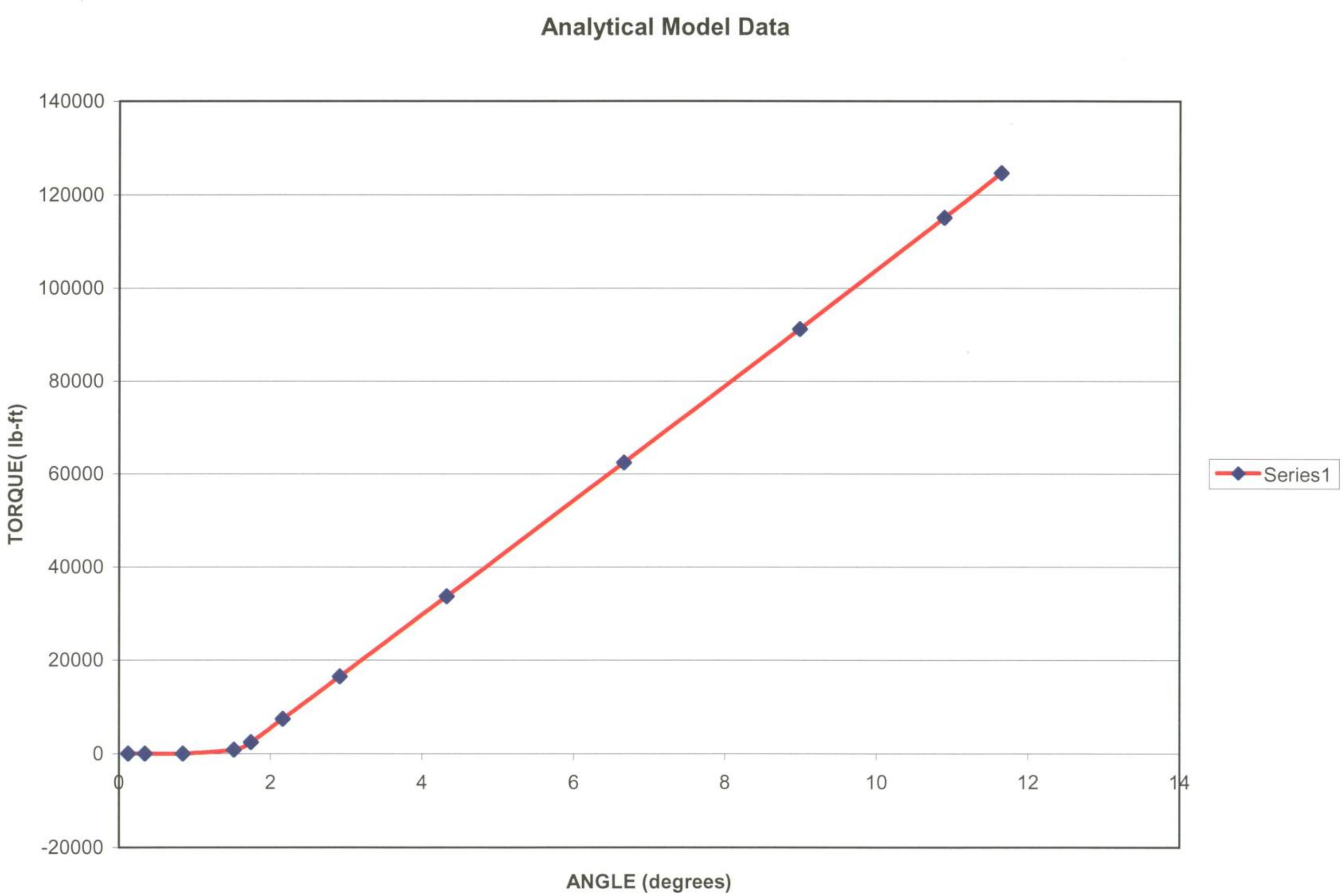


Figure 5. Analytical Model Data

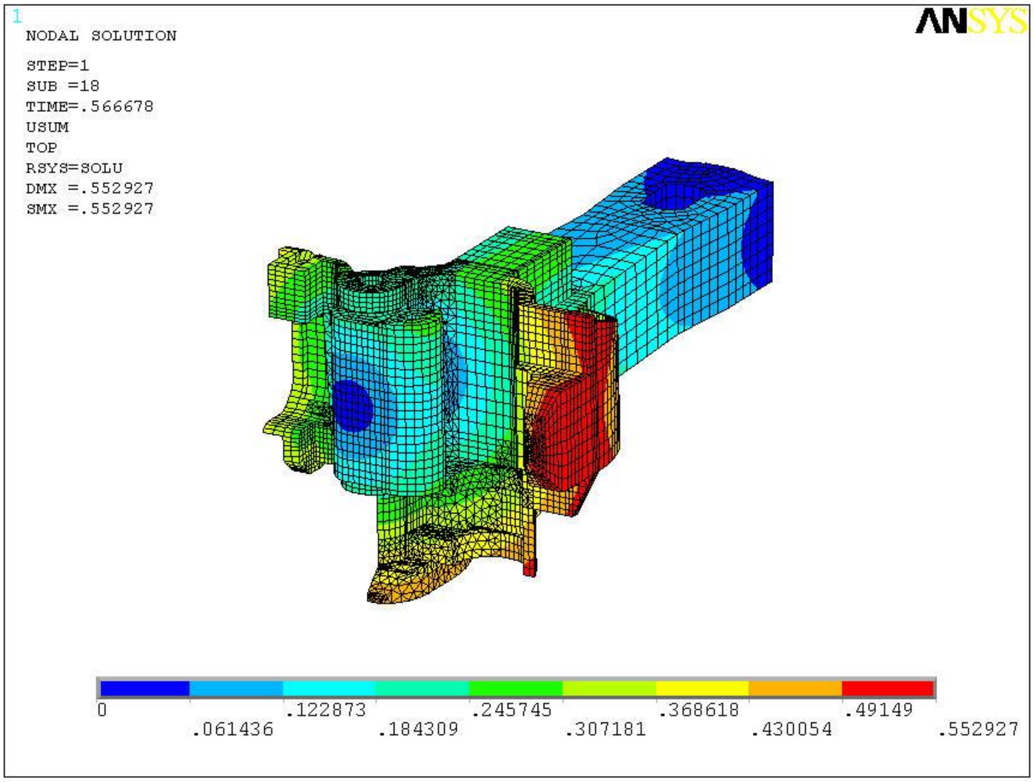


Figure 6. Total Displacement – Virtual Model

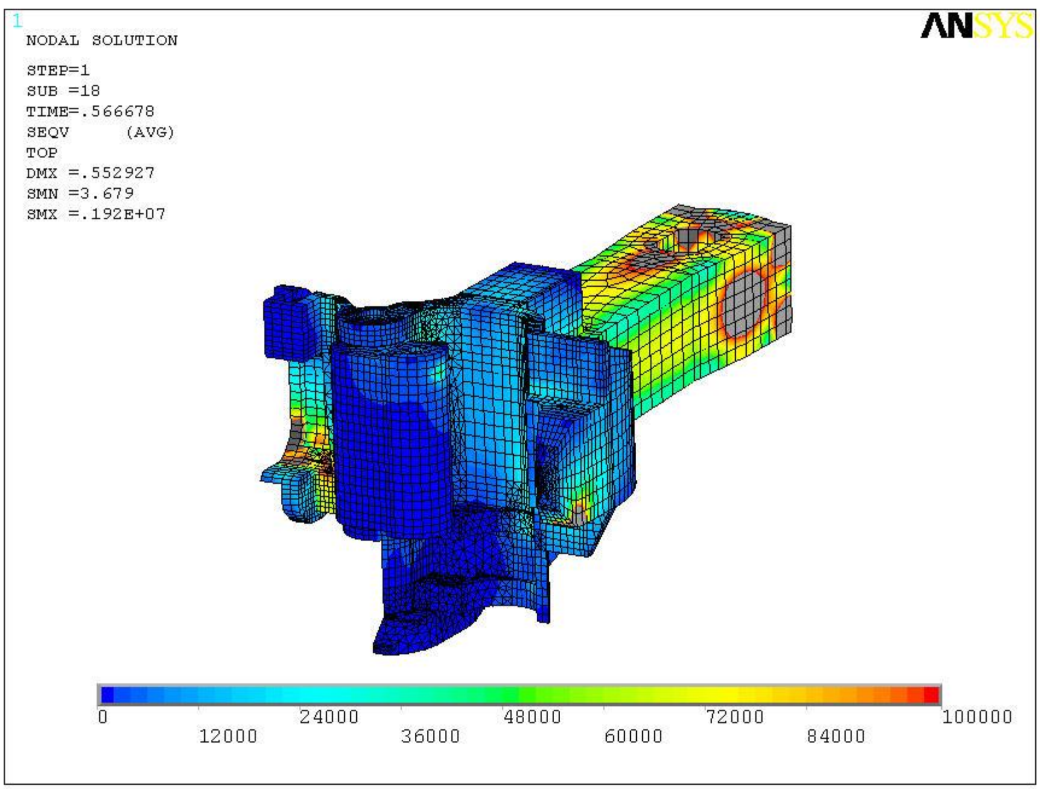


Figure 7. von Mises Stress – Virtual Model

3. Physical Testing

3.1 Introduction and Objective

Physical testing was performed to obtain torsional transmission stiffness data for four basic coupler combinations in both the clockwise and counterclockwise directions. Table 1 below summarizes these combinations:

Table 1. Combinations

	ACTUATED	To	REACTION
COMBINATION	No-Shelf (NS2)	↔	Shelf (S2)
	No-Shelf (NS2)	↔	No-Shelf (NS1)
	Shelf (S1)	↔	No-Shelf (NS1)
	Shelf (S1)	↔	Shelf (S2)

Resulting torque–displacement data from these test combinations should indicate whether any combination produces a stiffer torsional connection and possibly a greater tendency for overturning an adjacent coupled car. Use of both the clockwise and counterclockwise directions ensures that any contact at both the upper and lower shelves is taken into account.

3.2 Test Apparatus

A test apparatus was designed and fabricated consisting of two coupler support fixtures and a method of applying torque to one of the couplers. A welded steel actuation fixture and a reaction fixture were developed (see Figure 8 and Figure 9 and manufacturing drawings in Appendix B). The two fixtures face each other and can be moved relative to each other to allow the installation of the couplers into the fixtures and to allow the couplers to be mated and coupled together. The actuation fixture consists of a lower support frame and two upstanding stiffened webs, which locate and support bearings that accept the coupler shank of the actuated coupler. Provisions are made to attach a hydraulic cylinder to either the right or the left side of the fixture. A separate fabricated arm is provided. It grips the coupler shank of the actuated coupler and is given a force at its outer end by the hydraulic cylinder (see Figure 9 and Figure 10).

Clockwise or counterclockwise motion is possible by way of the cylinder supports at both sides of the fixture. Two cylindrical coupler shank adapters provided with tapered square holes and adjustable set screws allow the coupler shank to fit into the fixture at the bearing locations. The plain bearings are machined from Nylon 6 and fit between the adapters and the steel tubes that

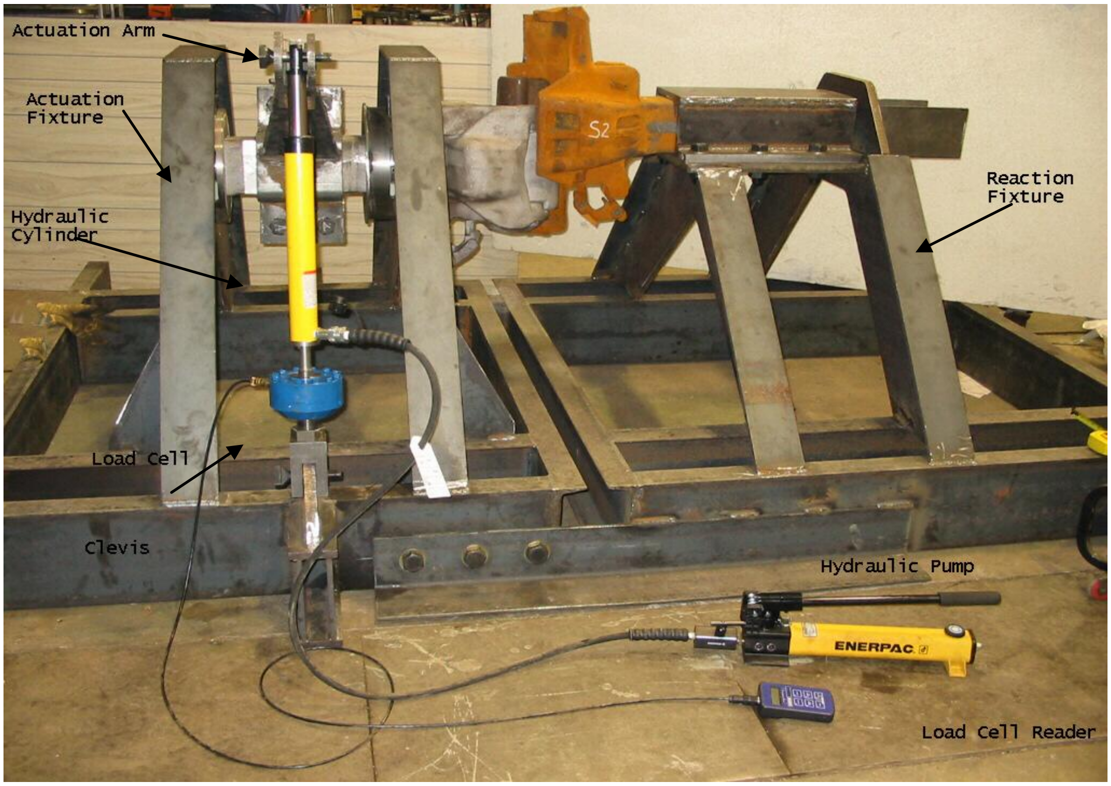
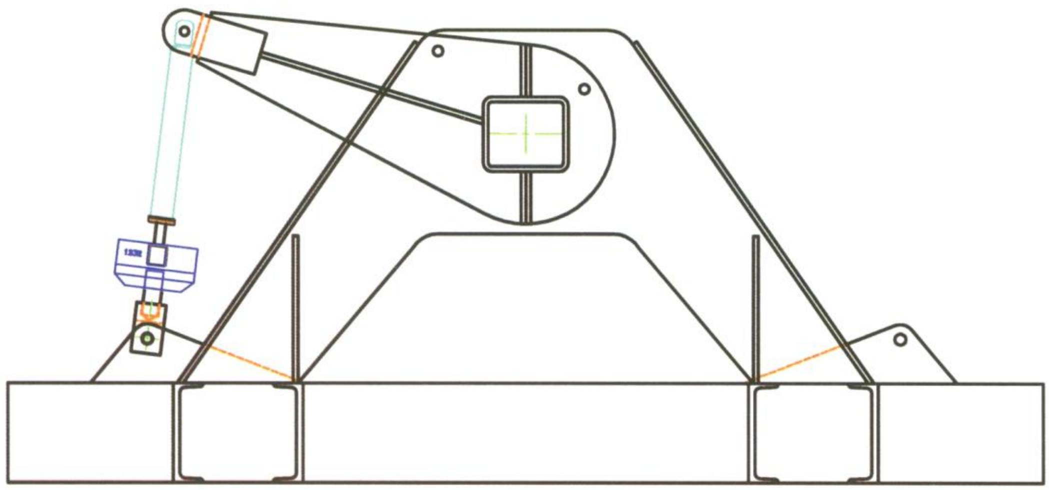


Figure 8. Test Apparatus

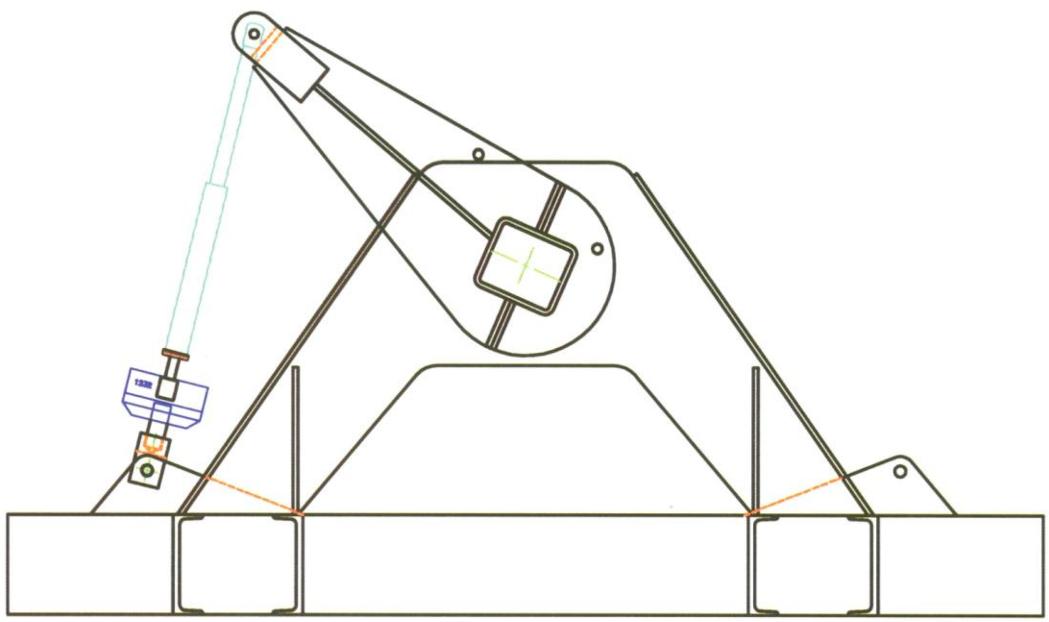
Figure 9. Test Apparatus

make up part of the fixture weldment. These bearings allow the actuated coupler to freely and easily rotate in the fixture with only hand force and remove any vertical moment out of the mated couplers upon actuation (see Figure 11 and Figure 12).

A theoretical range of motion of 23 degrees is possible by the use of an Enerpac® model RC-1014 single acting 14-inch extension hydraulic cylinder capable of providing 20,000 lb of force. In line with the cylinder is an Interface® model 1220 load cell with a 50,000-pound capacity. An Interface® model 9320 handheld load cell indicator is used to obtain force readings from the load cell. The cylinder, load cell, and two threaded adapters that connected the load cell to the cylinder and to a clevis pivot at the clevis on the mounting point of the fixture are shown in Figure 13.



INITIAL POSITION ARM AT 17°, COUPLER AT 0°



MAXIMUM EXTENDED POSITION ARM AT 40°, COUPLER AT 23°

Figure 10. Test Apparatus Motion

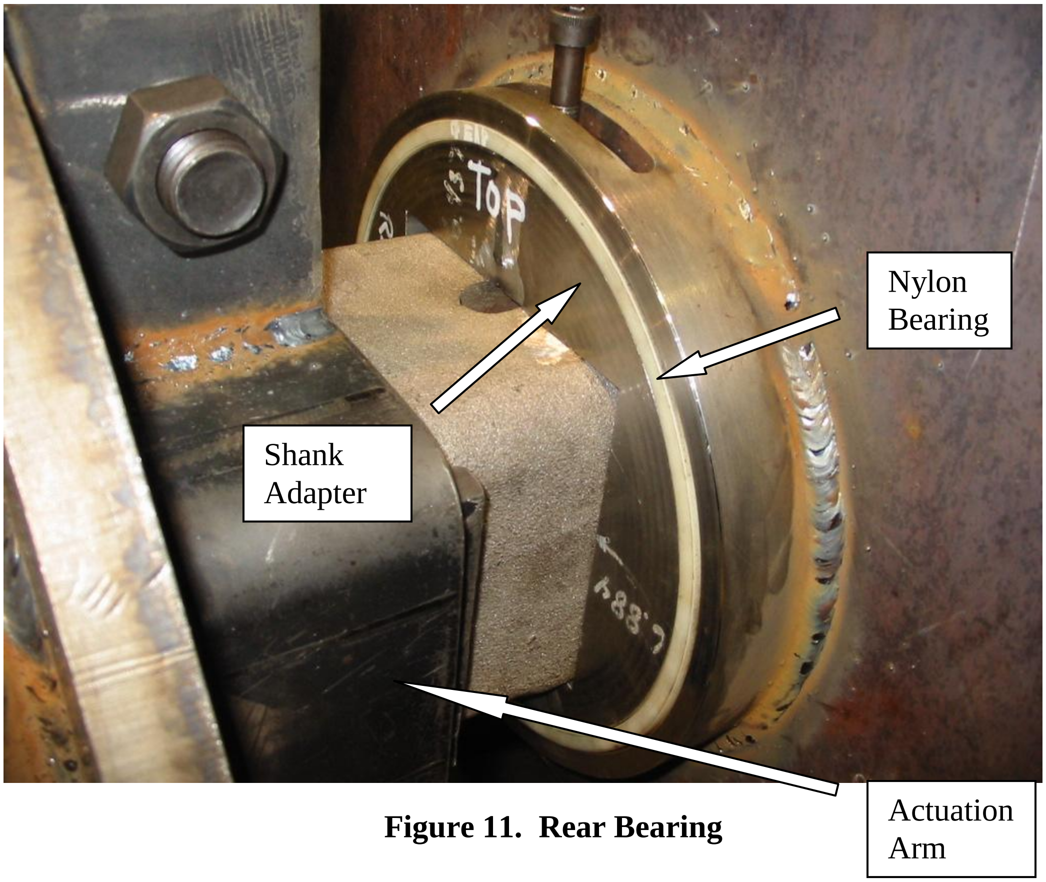


Figure 11. Rear Bearing

Figure 12. Front Bearing

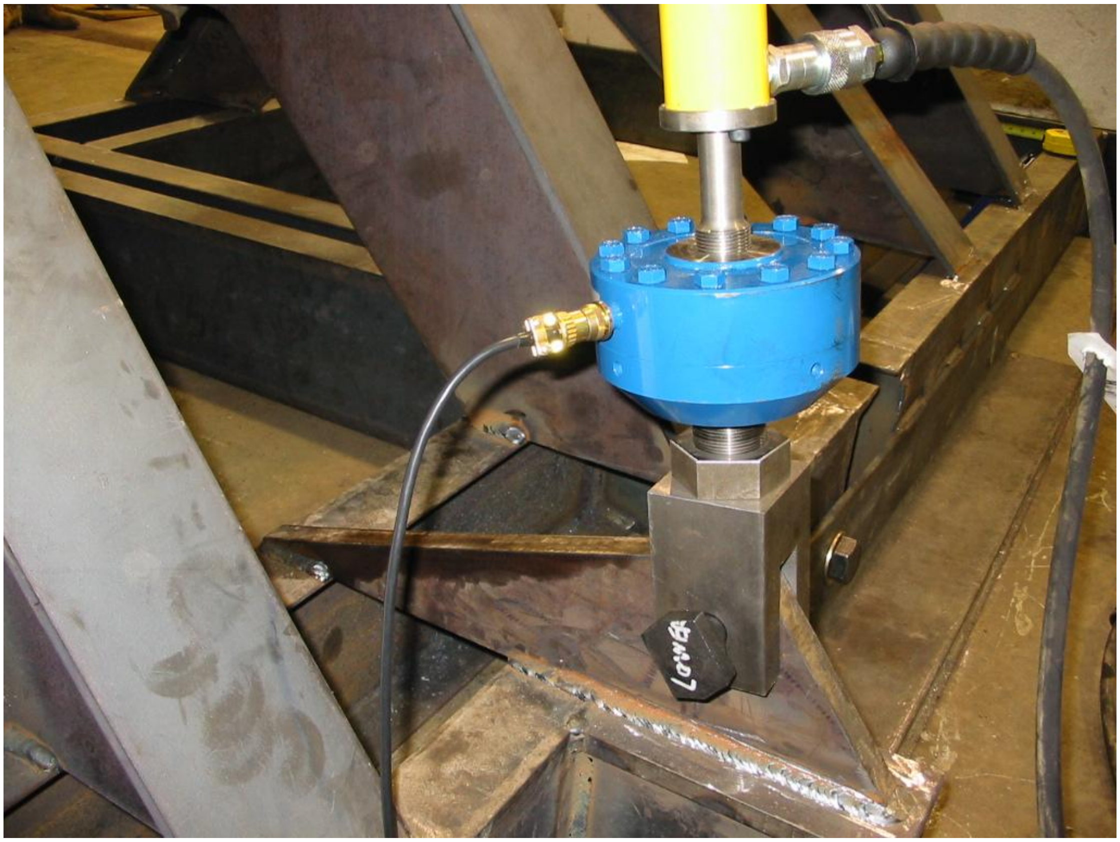


Figure 13. Load Cell and Clevis

An Enerpac® model P391 hydraulic hand pump is used in conjunction with a model V66 valve to actuate the cylinder and to provide the fine extensions of the cylinder plunger to obtain the angular displacement of the actuation arm.

The reaction fixture supports the nonactuated coupler and secures it by fastening a steel-hat section over the shank. The reaction fixture positions the coupler at a level 5 in above that of the actuated coupler to simulate the drop of one car relative to its mated neighbor in a derailment condition. Both fixtures were analyzed for acceptable stresses and deflections using the Finite Element Method with Ansys®. The actuation fixture is designed to slide into the reaction fixture and is bolted to the reaction fixture at the base.

3.3 Test Specimens

Four couplers were acquired for the test: two E68 and two SE68 types. All four couplers met AAR dimensional service standards and then had their shanks cut off near the rear at their narrowest point before the widening of the shank to allow them to fit into the fixture. The two E68 couplers were identified and labeled as S1 and S2. The two SE68s were identified and labeled NS1 and NS2.

3.4 Test Procedure

Four basic combinations of couplers were tested.

1. No-shelf with no-shelf (NS2 to NS1), NS2 actuated
2. Shelf with shelf (S1 to S2), S1 actuated
3. No-shelf with shelf (NS2 to S2), NS2 actuated
4. Shelf with no-shelf (S1 to NS1), S1 actuated

For each of these basic combinations, a clockwise and counterclockwise actuation was performed. Three runs of each direction were made for all four basic combinations. Force and angular displacement data were recorded typically at about 2,000-pound intervals. A Kell-Strom® Pro 360 digital protractor with magnetic base attachment was used for determining the angular displacement.

Shimming was used to remove any gaps or motion between the actuation arm and the coupler shanks and between the reaction-hold-down fixture and the coupler shanks. Before each run, initial data consisting of initial angle of coupler, initial force reading, and initial angles of the heads and shanks of both couplers were recorded. Similar readings were taken after stabilizing the maximum load and after unloading the system.

After each appropriate combination of coupler specimens were installed in the fixtures and the arm/cylinder assembly was securely attached and shimmed, the fixtures were moved together, and the couplers were coupled and the frames bolted together. The hydraulic hand pump was then carefully operated until the target force increment was approximately attained. The V66 valve was then closed, and a reading of the load cell was recorded along with the corresponding displacement. The valve was then opened, and the next increment of displacement was attained and so on. A maximum force reading of about 15,000 lb, equivalent to about 43,000 lb-ft of torque, was set as the limit for the test.

The recorded data points were of the input force through the hydraulic cylinder and the resulting angle of the arm. These data were reduced to obtain output torque on the coupler shank and the corresponding angle of the coupler shank. This was accomplished by using a spreadsheet with the appropriate trigonometric relationships.

3.5 Results – Physical Testing

Figure 14 and Figure 15 show the second run for each of the four basic coupler combinations plotted on a single graph. It shows that a significant difference exists among the four basic combinations. Regardless of rotational direction, the no-shelf to shelf combination produced the greatest angular displacement for a given torque, followed by the shelf to no-shelf, then the no-shelf to shelf and the shelf to shelf showing least angular displacement (see Table 2 and Table 3).

In the counterclockwise case, the shelf-to-shelf combination exhibited contact between the mating upper shelves through the duration of the tests (see Figure 16). The slopes of all the counterclockwise runs range from about 11,000 lb-ft/degree to about 18,000 lb-ft/degree (see Table 2).

The clockwise results are similar to the counterclockwise results except that:

1. The shelf-to-shelf and no-shelf-to-shelf curves show significantly more displacement for the clockwise direction.
2. The clockwise runs have slopes ranging from about 12,000 lb-ft/degree to about 15,000 lb-ft/degree (see Table 3).

The greater displacement of item 1 above is probably due to the top shelf contact, which occurs after a relatively small rotation in the counterclockwise direction. The top shelves do not make contact in the clockwise direction. Because of their shape and location, the bottom shelves appear to be less of a factor in effecting the couple rotation.

Irrespective of rotational direction, the shelf-to-shelf combination shows the steepest slope and therefore the greatest stiffness. These slopes most likely represent a combination of the fixture stiffness after the relative angles between the couplers have stabilized at their maximum values and stiffness of the coupler itself. Slope values were obtained by performing linear regression on all but the first two data points of each curve. The offset between curves represents the most significant difference in behavior of the chosen couplers.

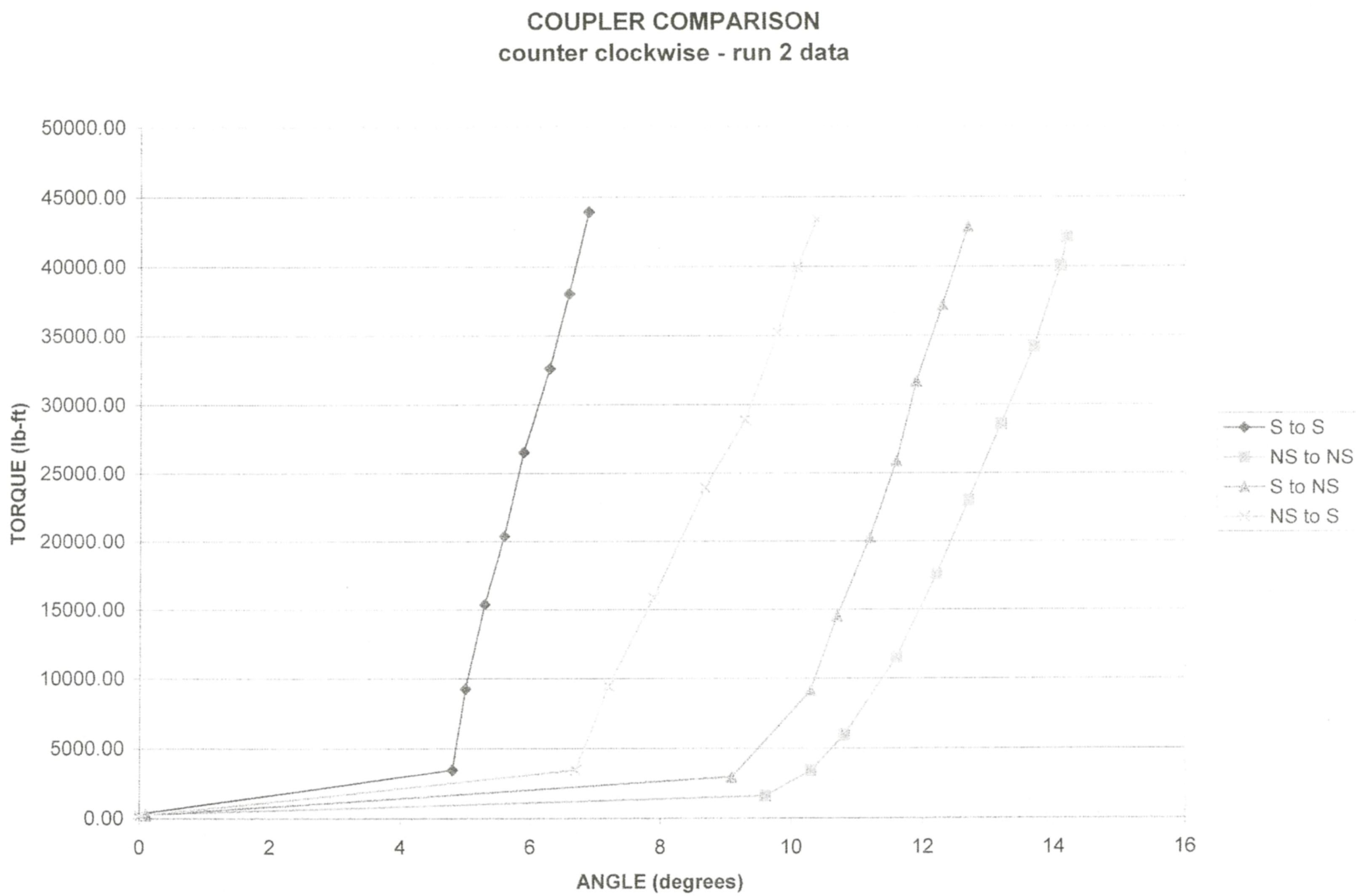


Figure 14. Results Comparison – Counterclockwise

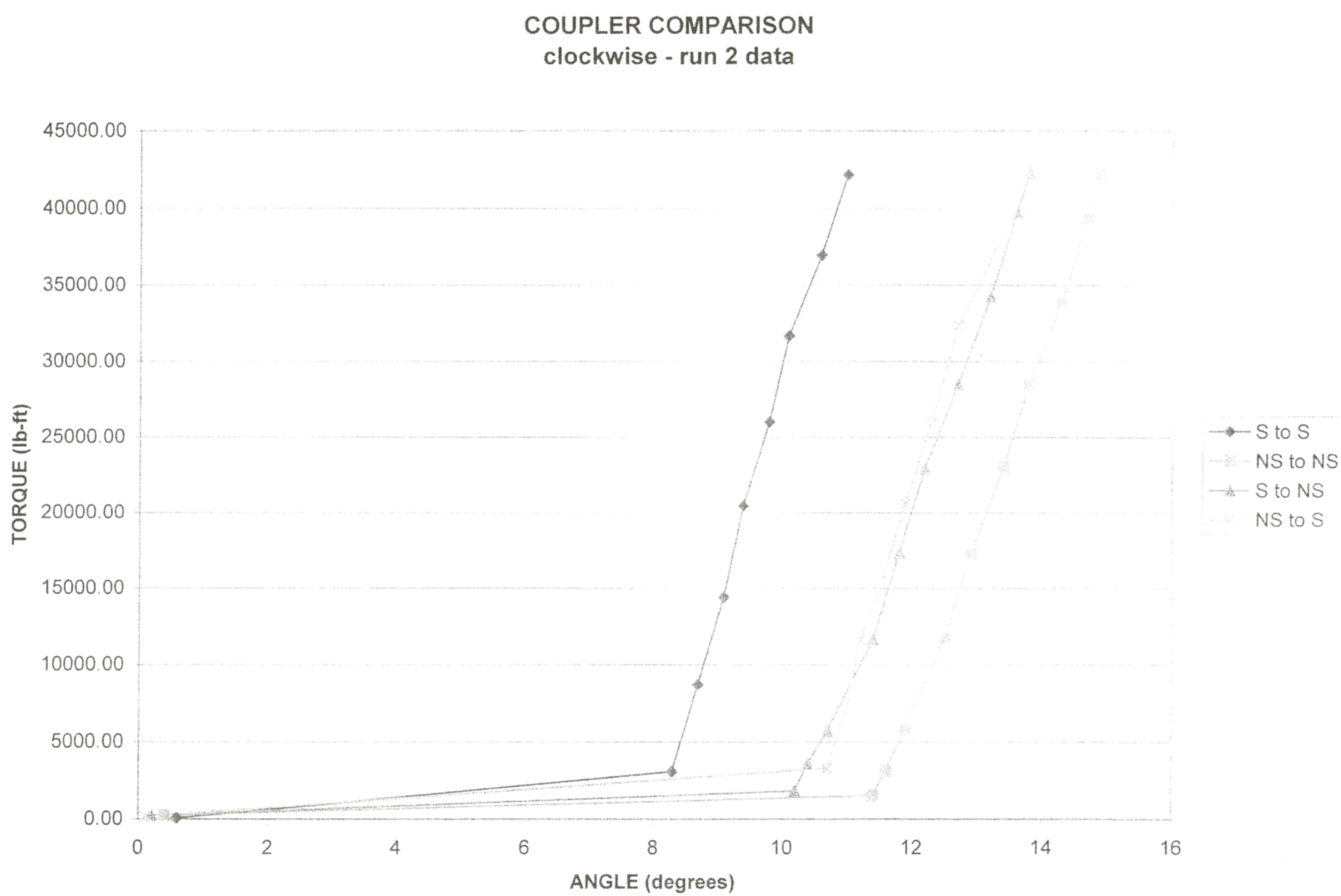


Figure 15. Comparison of Results – Clockwise

Table 2. Results Comparison – Counterclockwise

COUNTERCLOCKWISE (run 2)				
	S to S	NS to NS	S to NS	NS to S
Max. angle (deg)	6.9	14.2	12.7	10.4
Max. torque (lb-ft)	43,923	42,122	42,872	43,368
Slope (lb-ft/degree)	17,942	10,760	14,194	10,538

Table 3. Results Comparison – Clockwise

CLOCKWISE (run 2)				
	S to S	NS to NS	S to NS	NS to S
Max. angle (deg)	11	14.9	13.8	13.7
Max. torque (lb-ft)	42,166	42,162	42,292	42,516
Slope (lb-ft/degree)	14,708	11,943	11,649	12,884

The virtual model was made of an F-type to F-type coupler connection. However, comparison of the physical testing with that of the virtual model will tell us whether the results of each are similar. Comparing the results from the physical testing with that of the virtual model of the F-type couplers, we can see that the test slope values agree well with the simulated 12,337 lb-ft/degree value obtained. The actual value of angular displacement (used as reference) from the virtual model of 4.8 degrees at 40,000 lb-ft of torque is somewhat lower (greater stiffness) than that obtained in the physical test of the E-type couplers. This is most likely due to the presence of the interlocking aligning wing feature of the F-type coupler which will reduce the allowable angular displacement between couplers.

Appendix A contains data plots for all the runs and shows comparisons of the first through third runs (or first and second if only two runs were made) for each direction and combination. Most of the curves show a distinct break-in that occurs during the first run, followed by more consistent results at a greater displacement for the subsequent runs.

It should be noted that visual observation during test runs revealed that the buff to draft relationship of the knuckles in contact tended to change as the torque was applied or released. See Figure 17 for a view of the typical contact between couplers. The exception to this was the shelf-to-shelf case where the contact of the upper shelf to upper shelf maintained the relative longitudinal position of the knuckles constant throughout the run after starting the test run in a buff position (see Figure 16). Figure 18 and Figure 19 show side views of the shelf to shelf and no-shelf to no-shelf test runs.

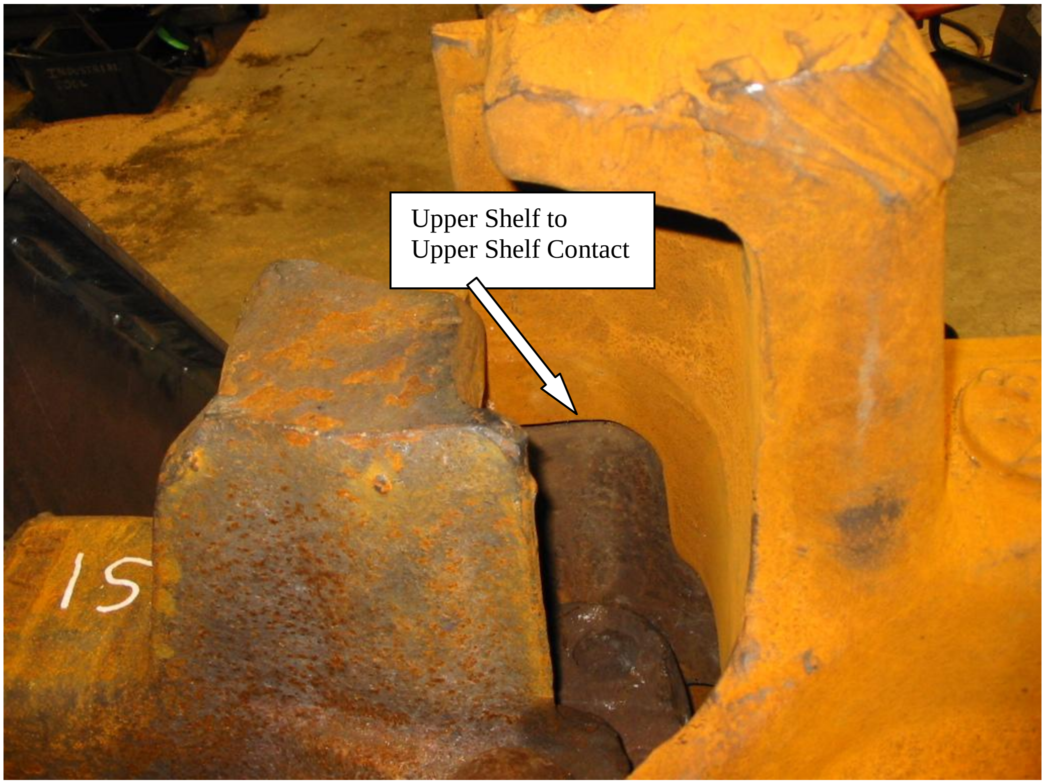


Figure 16. Upper Shelf to Upper Shelf Contact

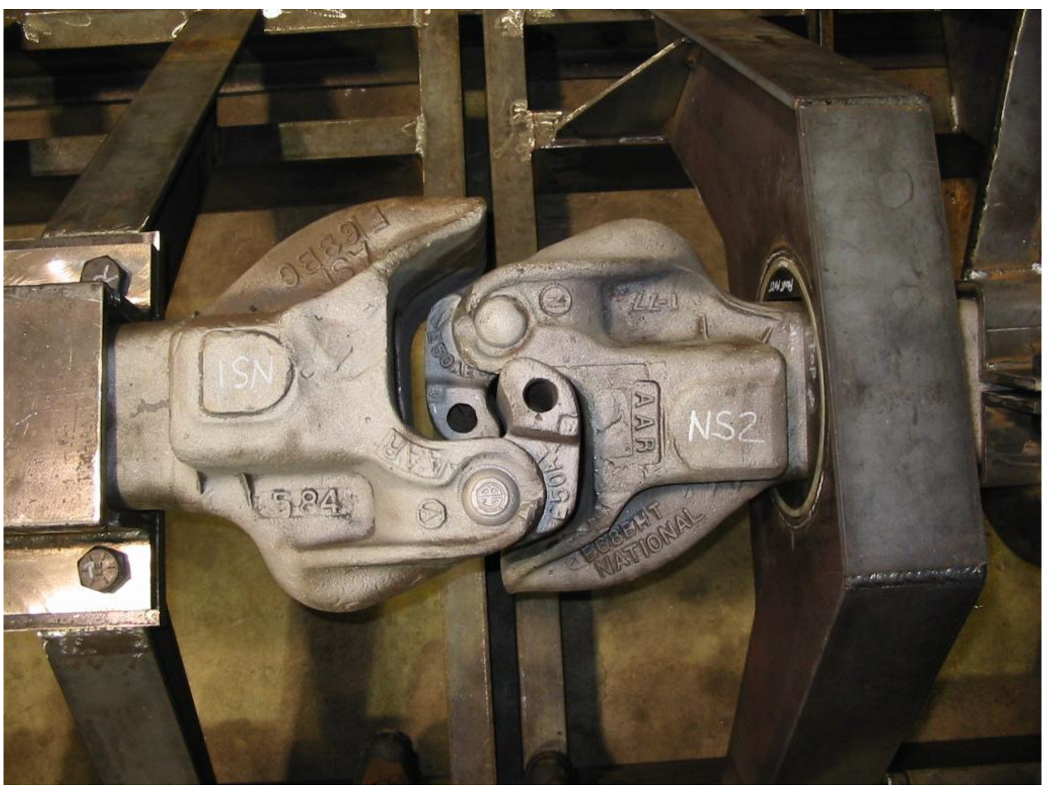


Figure 17. Typical Contact (No-Shelf Mated to No-Shelf)

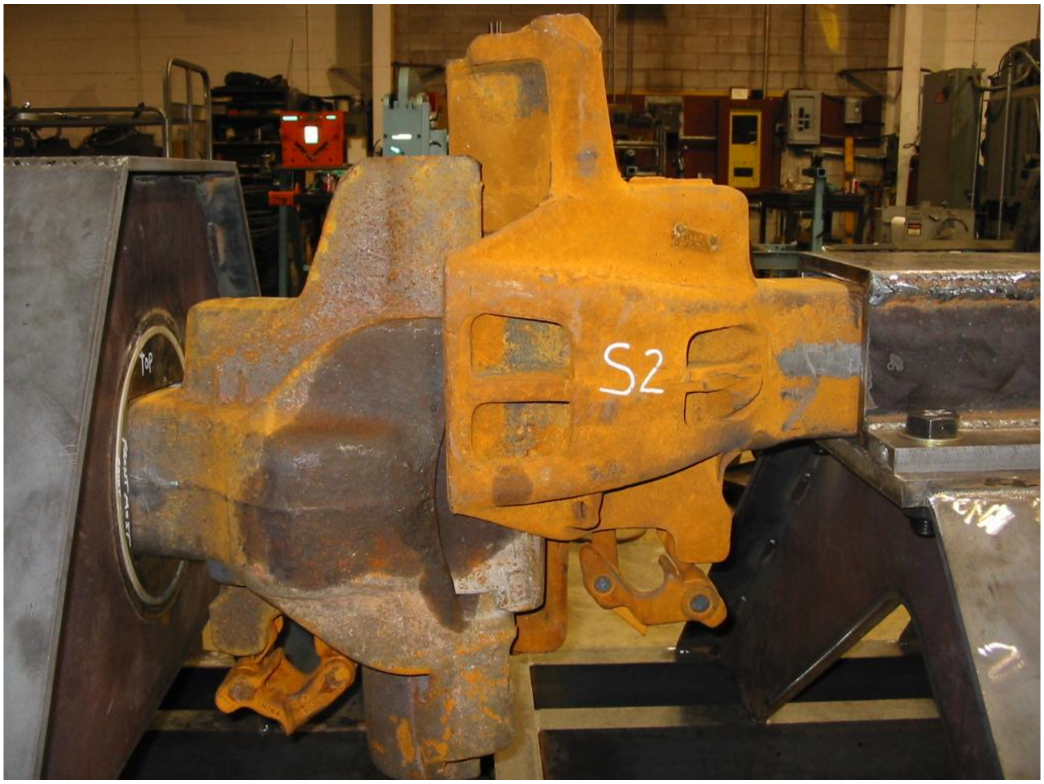


Figure 18. Shelf to Shelf Test Run

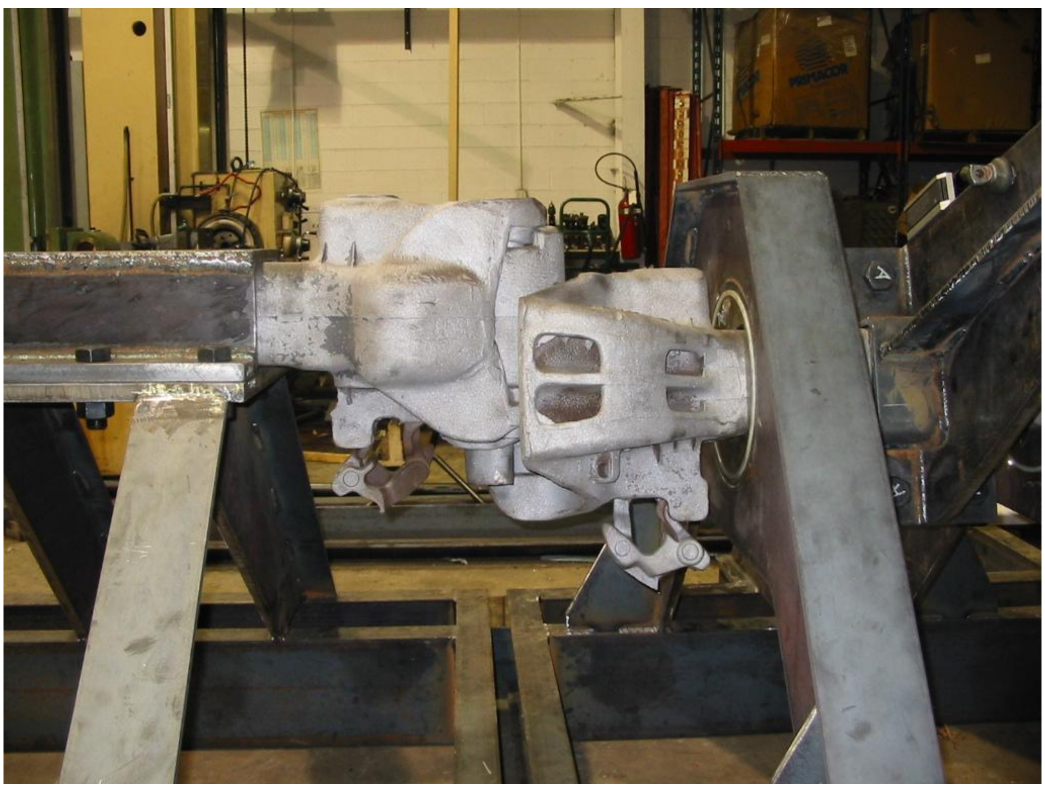


Figure 19. No-Shelf to No-Shelf Test Run

4. Conclusions

The results of this test clearly show less angular rotation for a given torque for couplers with shelves than those without. The combinations that included only one shelf coupler also showed less deflection than the combination without any shelves. This reveals that the presence of a shelf coupler or couplers will add to the torsional stiffness of the coupler connection. Therefore, it's possible that couplers with shelves are more prone to "domino" adjacent cars in a rollover scenario than those without. This propensity could be greater if the shelf-to-shelf couplers are in a generally buff condition when overturning because their upper shelves are more likely to be in contact in this position.

In contrast, in some scenarios, a mated shelf coupler may help to resist and inhibit an adjacent car from rolling over. Possibly, for every "domino" effect type of multicar rollover, some potential rollovers are eliminated because of the shelf type coupler.

5. Recommendations

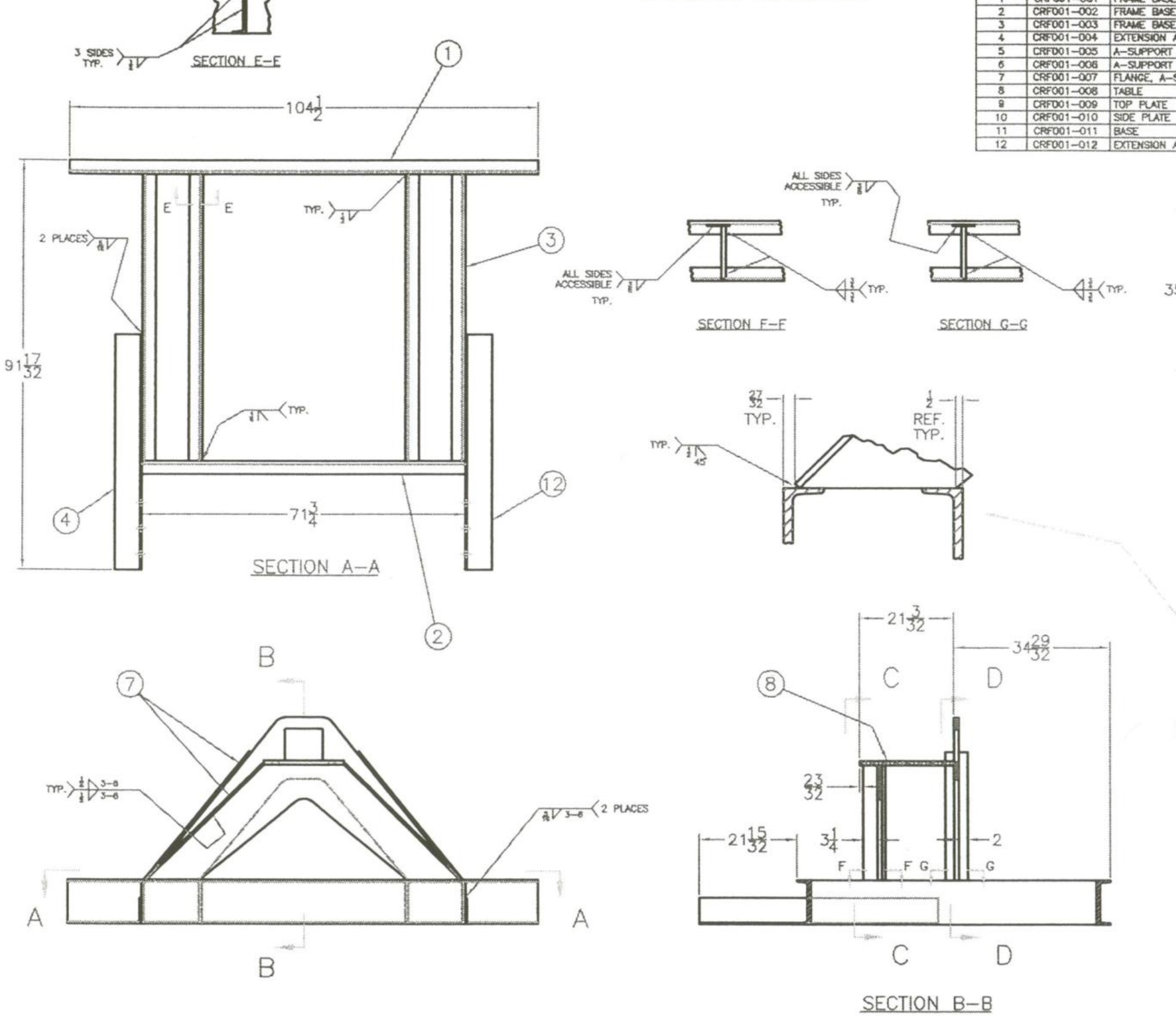
The question as to whether shelf couplers add to the torsional stiffness of coupled connections has been answered in this study. What is still not clear is whether this stiffness will contribute to, or help prevent, rollovers of adjacent coupled cars. Therefore, we recommend further study into the characteristics of the coupler connection and its propensity of propagating car rollovers. Through the use of the torque-displacement relationships established in this study, a vehicle dynamics simulation should be performed using a software application such as Vampire®. This type of analysis model can account for more variables such as those of the car suspension and car structure not possible in this study. A virtual model of a rail car with a coupler connection or multiple rail cars coupled can be constructed and analyzed to determine the actual propensity of one car to rollover its coupled neighbor based on the types of couplers used in the connection.

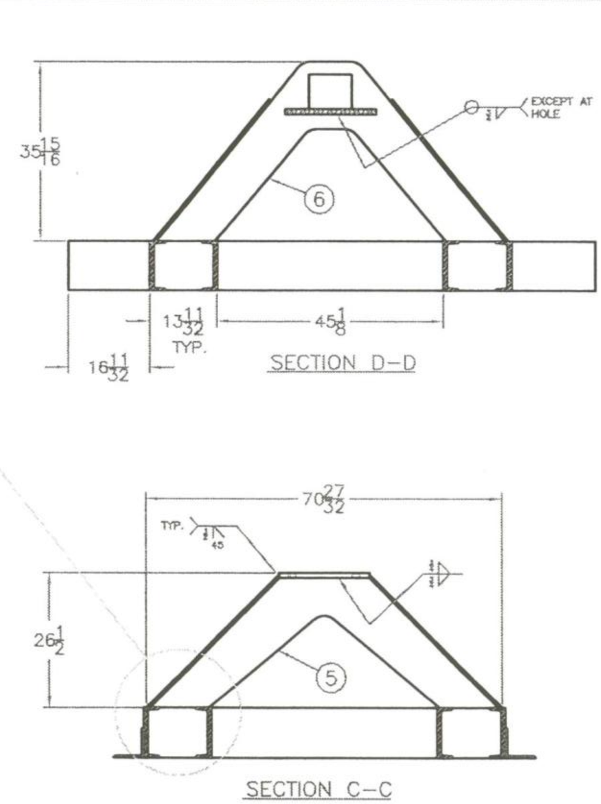
Appendix A – Fixture Assembly Drawings

NOTES:
SEE DRAWING SHEET 2 FOR GENERAL NOTES.

LIST OF MATERIALS

ITEM No.	PART No.	DESCRIPTION	MATERIAL / SPECIFICATION	QTY.
1	CRF001-001	FRAME BASE MEMBER	C10 X 30 A-36 STEEL CHANNEL - 105" LONG	1
2	CRF001-002	FRAME BASE MEMBER	C10 X 30 A-36 STEEL CHANNEL - 71 3/4" LONG	1
3	CRF001-003	FRAME BASE MEMBER	C10 X 30 A-36 STEEL CHANNEL - 64" LONG	4
4	CRF001-004	EXTENSION ANGLE	L6 x 6 x 3/8 A36 STEEL ANGLE - 53" LONG	1
5	CRF001-005	A-SUPPORT FRONT	A36 STEEL 1" THICK (BURNOUT) see dwg.	1
6	CRF001-006	A-SUPPORT REAR	A36 STEEL 1" THICK (BURNOUT) see dwg.	1
7	CRF001-007	FLANGE, A-SUPPORTS	A36 STEEL 0.50" THICK , 36 3/4" X 5"	4
8	CRF001-008	TABLE	A36 STEEL 1" THICK , 21 3/32" X 18"	1
9	CRF001-009	TOP PLATE	A36 STEEL 0.75" THICK , 9 1/2" X 21 3/32"	1
10	CRF001-010	SIDE PLATE	A36 STEEL 0.75" THICK , 6 7/16" X 21 3/32"	2
11	CRF001-011	BASE	A36 STEEL 1" THICK , 3 15/32" X 21 3/32"	2
12	CRF001-012	EXTENSION ANGLE	L6 x 6 x 3/8 A36 STEEL ANGLE - 53" LONG	1





DIMENSIONS IN INCHES
TOLERANCES UNLESS NOTED
FRACTIONS ± 1/16
TWO PLACES DEC. ± .03
THREE PLACES DEC. ± .016
ANGLES ± 1°

SHARMA & ASSOCIATES, INC.

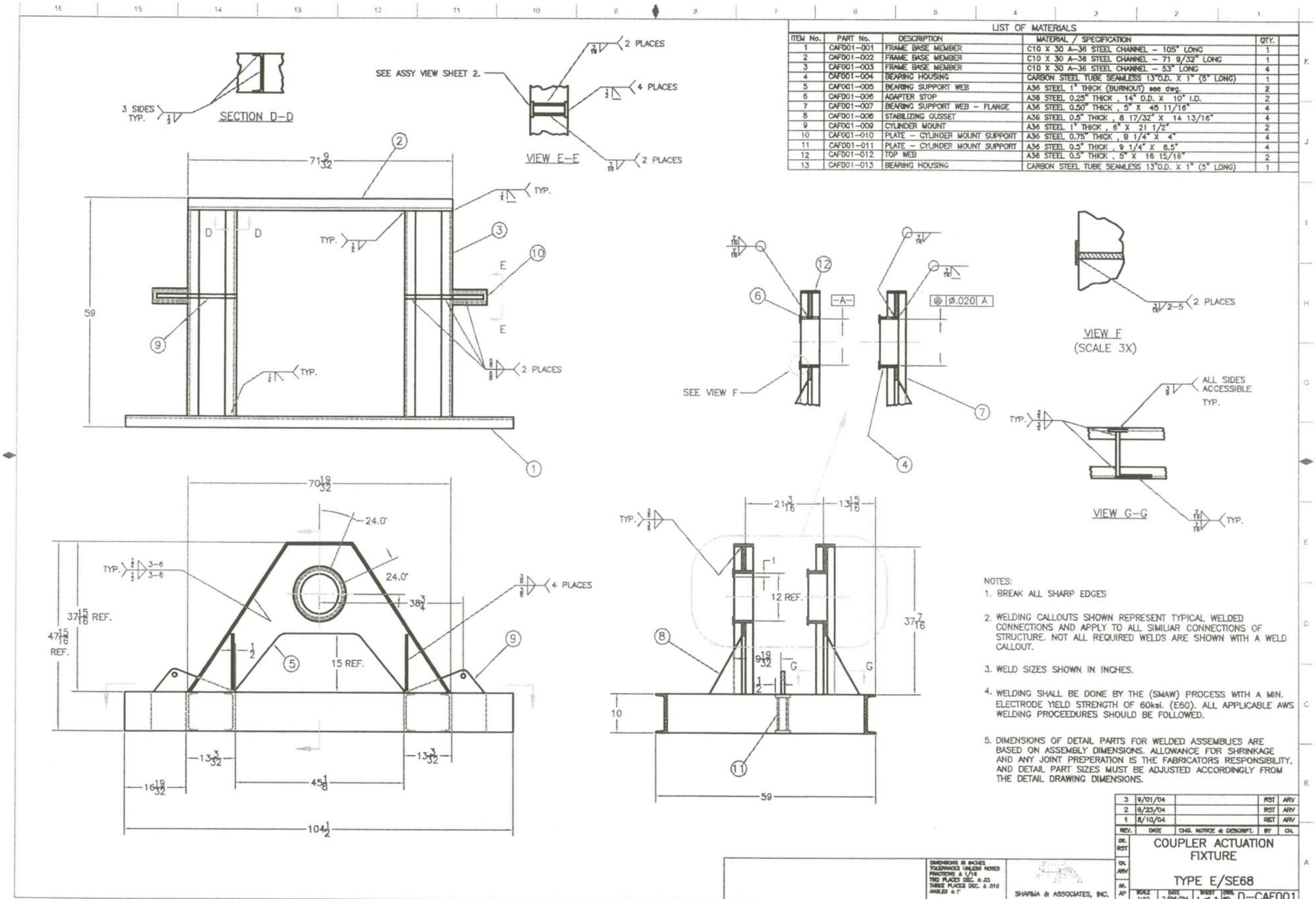
REV.	DATE	CHG. NOTICE & DESCRIPT.	BY	CH.
2	9/8/04		RST	ARV
1	8/25/04		RST	ARV

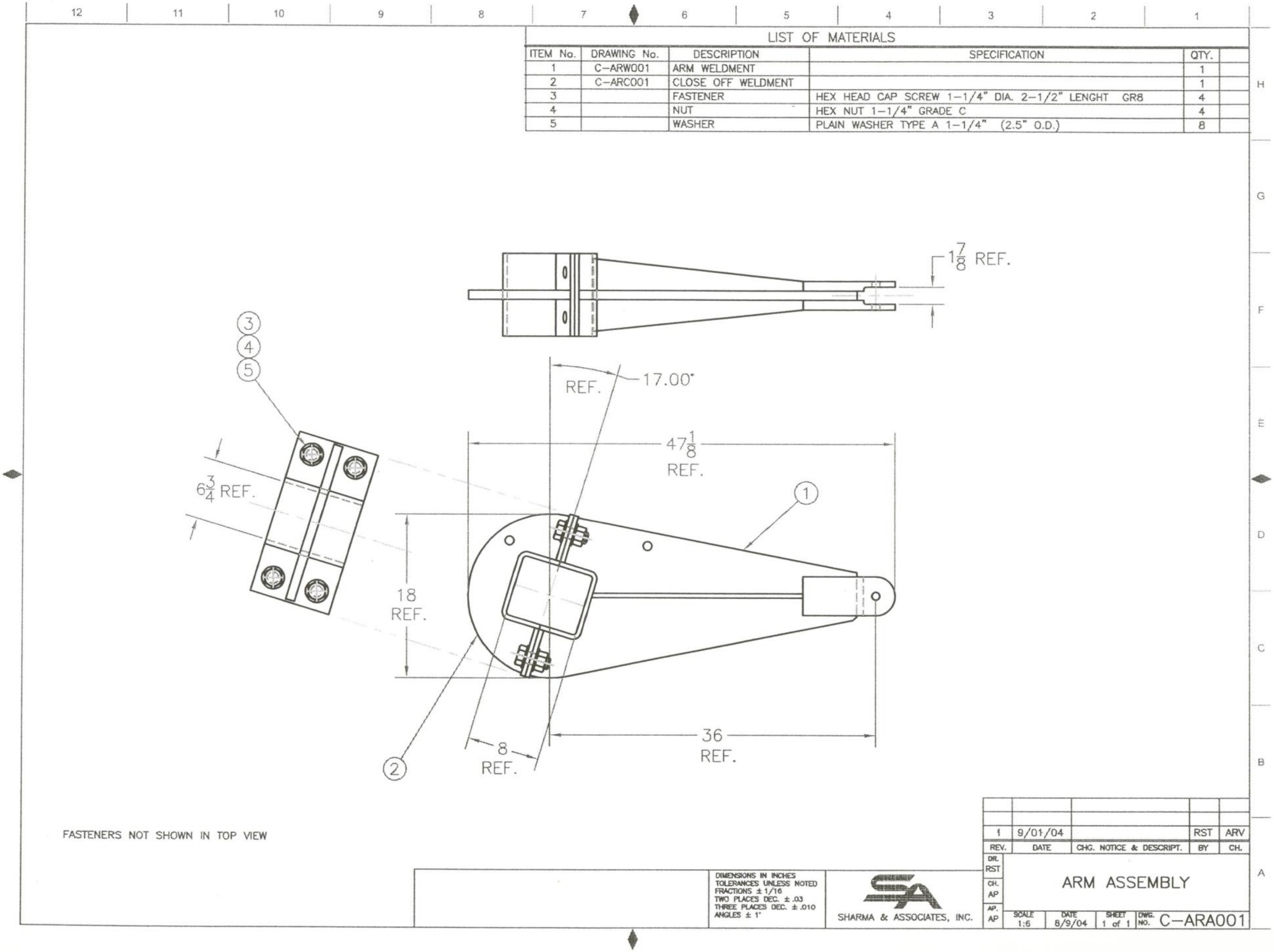
COUPLER REACTION
TEST FIXTURE

TYPE E/SE68CE

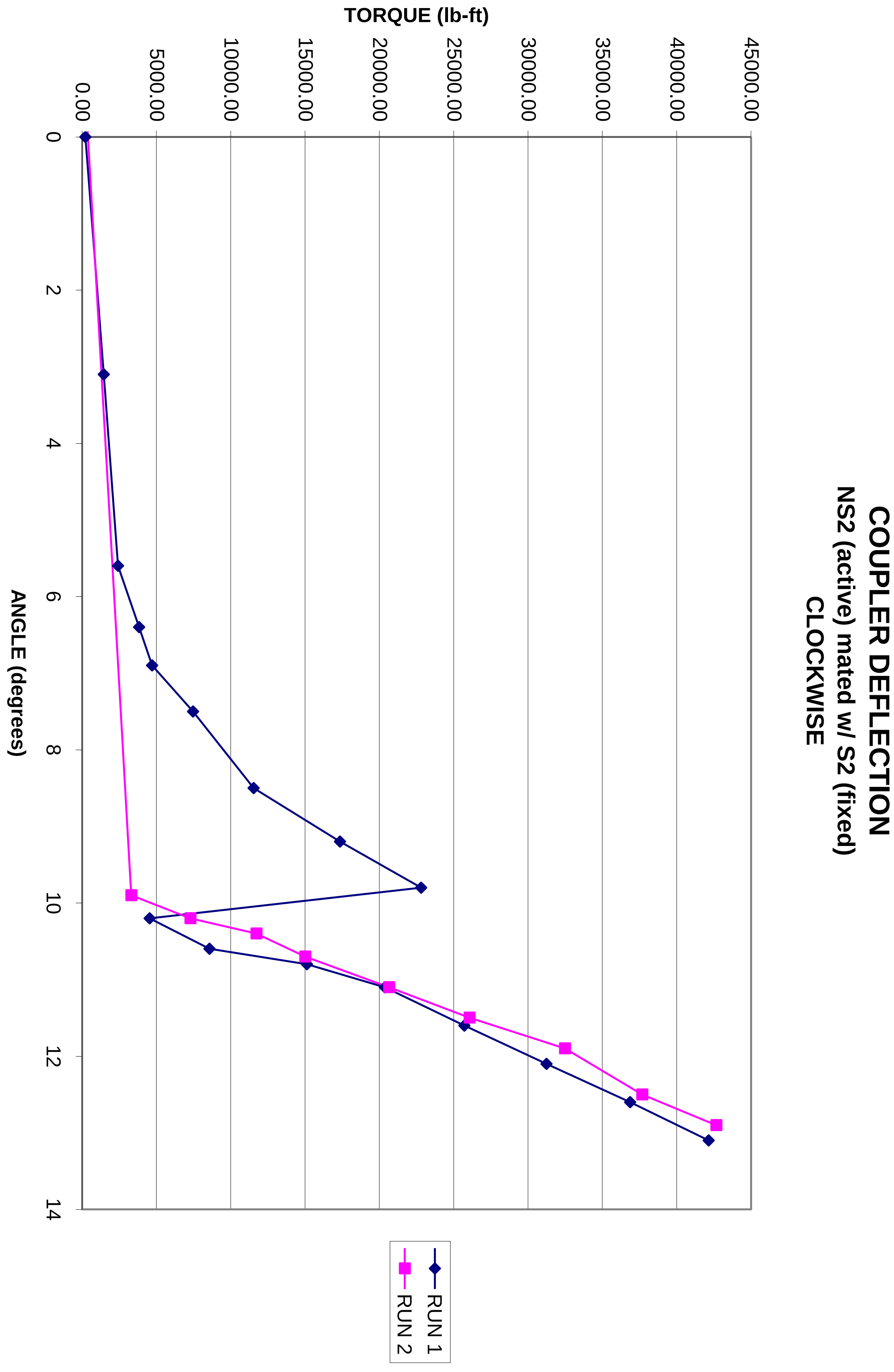
DR. RST · CH. ARV · AP. AP

SCALE 1:11 · DATE 7/29/04 · SHEET 1 of 2 · DWG. NO. D-CRF001

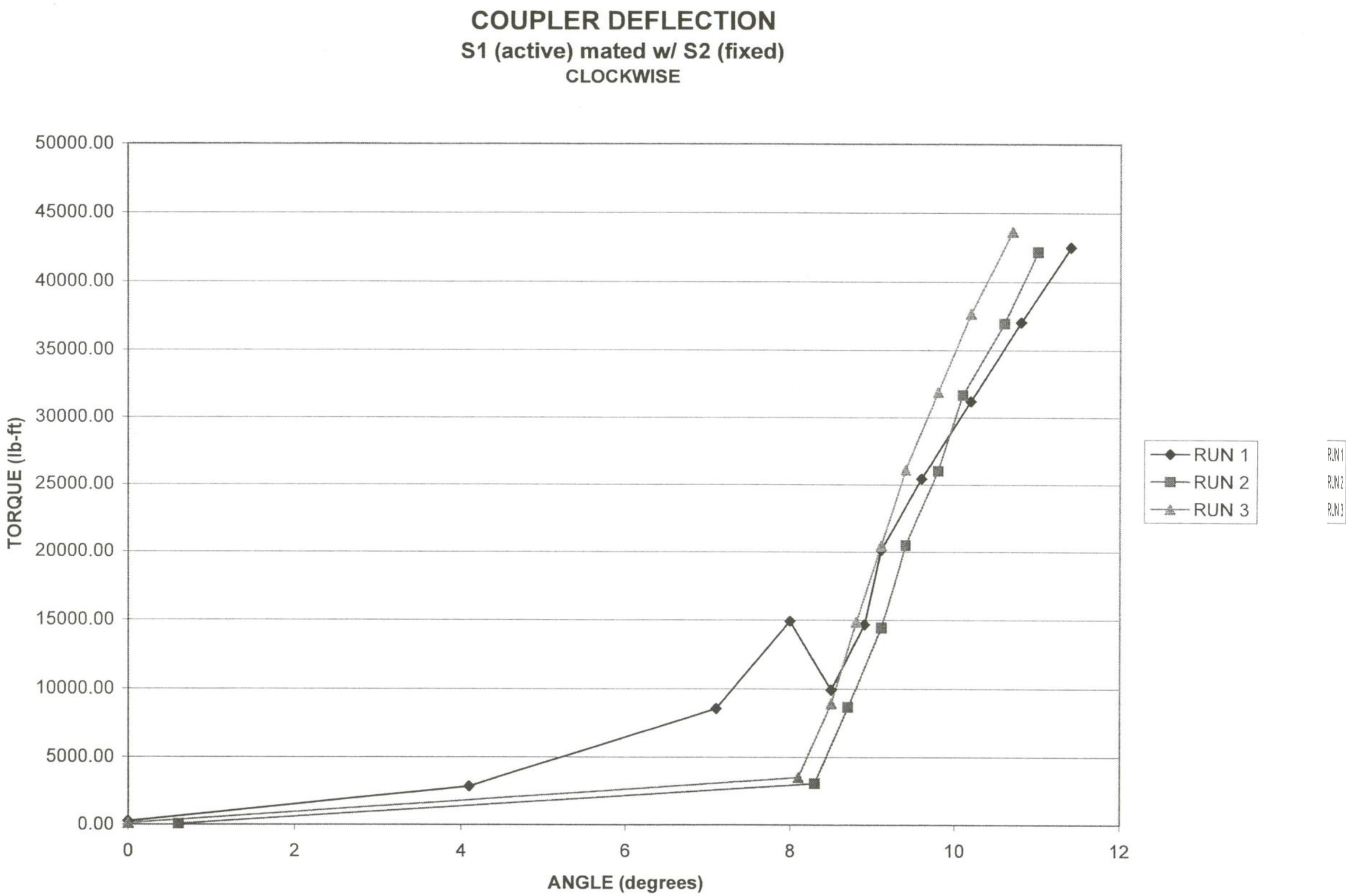

LIST OF MATERIALS
ITEM No. | PART No. | DESCRIPTION | MATERIAL / SPECIFICATION | QTY.
1 | CAFD01-001 | FRAME BASE MEMBER | C10 X 30 A-36 STEEL CHANNEL - 105" LONG | 1
2 | CAFD01-002 | FRAME BASE MEMBER | C10 X 30 A-36 STEEL CHANNEL - 71 9/32" LONG | 1
3 | CAFD01-003 | FRAME BASE MEMBER | C10 X 30 A-36 STEEL CHANNEL - 53" LONG | 4
4 | CAFD01-004 | BEARING HOUSING | CARBON STEEL TUBE SEAMLESS 13"O.D. X 1" (5" LONG) | 1
5 | CAFD01-005 | BEARING SUPPORT WEB | A36 STEEL 1" THICK (BURNOUT) see dwg. | 2
6 | CAFD01-006 | ADAPTER STOP | A36 STEEL 0.25" THICK , 14" O.D. X 10" I.D. | 2
7 | CAFD01-007 | BEARING SUPPORT WEB - FLANGE | A36 STEEL 0.50" THICK , 5" X 45 11/16" | 4
8 | CAFD01-008 | STABILIZING GUSSET | A36 STEEL 0.5" THICK , 8 17/32" X 14 13/16" | 4
9 | CAFD01-009 | CYLINDER MOUNT | A36 STEEL 1" THICK , 6" X 21 1/2" | 2
10 | CAFD01-010 | PLATE - CYLINDER MOUNT SUPPORT | A36 STEEL 0.75" THICK , 9 1/4" X 4" | 4
11 | CAFD01-011 | PLATE - CYLINDER MOUNT SUPPORT | A36 STEEL 0.5" THICK , 9 1/4" X 8.5" | 4
12 | CAFD01-012 | TOP WEB | A36 STEEL 0.5" THICK , 5" X 16 15/16" | 2
13 | CAFD01-013 | BEARING HOUSING | CARBON STEEL TUBE SEAMLESS 13"O.D. X 1" (5" LONG) | 1
SECTION D-D
3 SIDES TYP.
SEE ASSY VIEW SHEET 2.
VIEW E-E
2 PLACES
4 PLACES
TYP.
SEE VIEW F
VIEW F
(SCALE 3X)
VIEW G-G
ALL SIDES ACCESSIBLE TYP.
24.0°
12 REF.
15 REF.
REF.
NOTES:
1. BREAK ALL SHARP EDGES
2. WELDING CALLOUTS SHOWN REPRESENT TYPICAL WELDED CONNECTIONS AND APPLY TO ALL SIMILIAR CONNECTIONS OF STRUCTURE. NOT ALL REQUIRED WELDS ARE SHOWN WITH A WELD CALLOUT.
3. WELD SIZES SHOWN IN INCHES.
4. WELDING SHALL BE DONE BY THE (SMAW) PROCESS WITH A MIN. ELECTRODE YIELD STRENGTH OF 60ksi. (E60). ALL APPLICABLE AWS WELDING PROCEEDURES SHOULD BE FOLLOWED.
5. DIMENSIONS OF DETAIL PARTS FOR WELDED ASSEMBLIES ARE BASED ON ASSEMBLY DIMENSIONS. ALLOWANCE FOR SHRINKAGE AND ANY JOINT PREPERATION IS THE FABRICATORS RESPONSIBILITY, AND DETAIL PART SIZES MUST BE ADJUSTED ACCORDINGLY FROM THE DETAIL DRAWING DIMENSIONS.
3 9/01/04 RST ARV
2 8/23/04 RST ARV
1 8/10/04 RST ARV
REV. DATE CHG. NOTICE & DESCRIPT. BY CK.
COUPLER ACTUATION FIXTURE
TYPE E/SE68
SHARMA & ASSOCIATES, INC.
SCALE 1:10 DATE 7/06/04 SHEET 1 of 2 DWG. NO. D-CAF001


LIST OF MATERIALS
ITEM No. | DRAWING No. | DESCRIPTION | SPECIFICATION | QTY.
1 | C-ARW001 | ARM WELDMENT | | 1
2 | C-ARC001 | CLOSE OFF WELDMENT | | 1
3 | | FASTENER | HEX HEAD CAP SCREW 1-1/4" DIA. 2-1/2" LENGHT GR8 | 4
4 | | NUT | HEX NUT 1-1/4" GRADE C | 4
5 | | WASHER | PLAIN WASHER TYPE A 1-1/4" (2.5" O.D.) | 8
1 7/8 REF.
17.00° REF.
47 1/8 REF.
6 3/4 REF.
18 REF.
8 REF.
36 REF.
FASTENERS NOT SHOWN IN TOP VIEW
DIMENSIONS IN INCHES
TOLERANCES UNLESS NOTED
FRACTIONS ± 1/16
TWO PLACES DEC. ± .03
THREE PLACES DEC. ± .010
ANGLES ± 1°
SHARMA & ASSOCIATES, INC.
1 | 9/01/04 | | RST | ARV
REV. | DATE | CHG. NOTICE & DESCRIPT. | BY | CH.
DR. RST
CH. AP
AP. AP
ARM ASSEMBLY
SCALE 1:6
DATE 8/9/04
SHEET 1 of 1
DWG. NO. C-ARA001

Appendix B – Test Data



COUPLER DEFLECTION
NS2 (active) mated w/ S2 (fixed)
CLOCKWISE
TORQUE (lb-ft)
0.00
5000.00
10000.00
15000.00
20000.00
25000.00
30000.00
35000.00
40000.00
45000.00
ANGLE (degrees)
0
2
4
6
8
10
12
14
RUN 1
RUN 2


COUPLER DEFLECTION
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TORQUE (lb-ft)
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45000.00
40000.00
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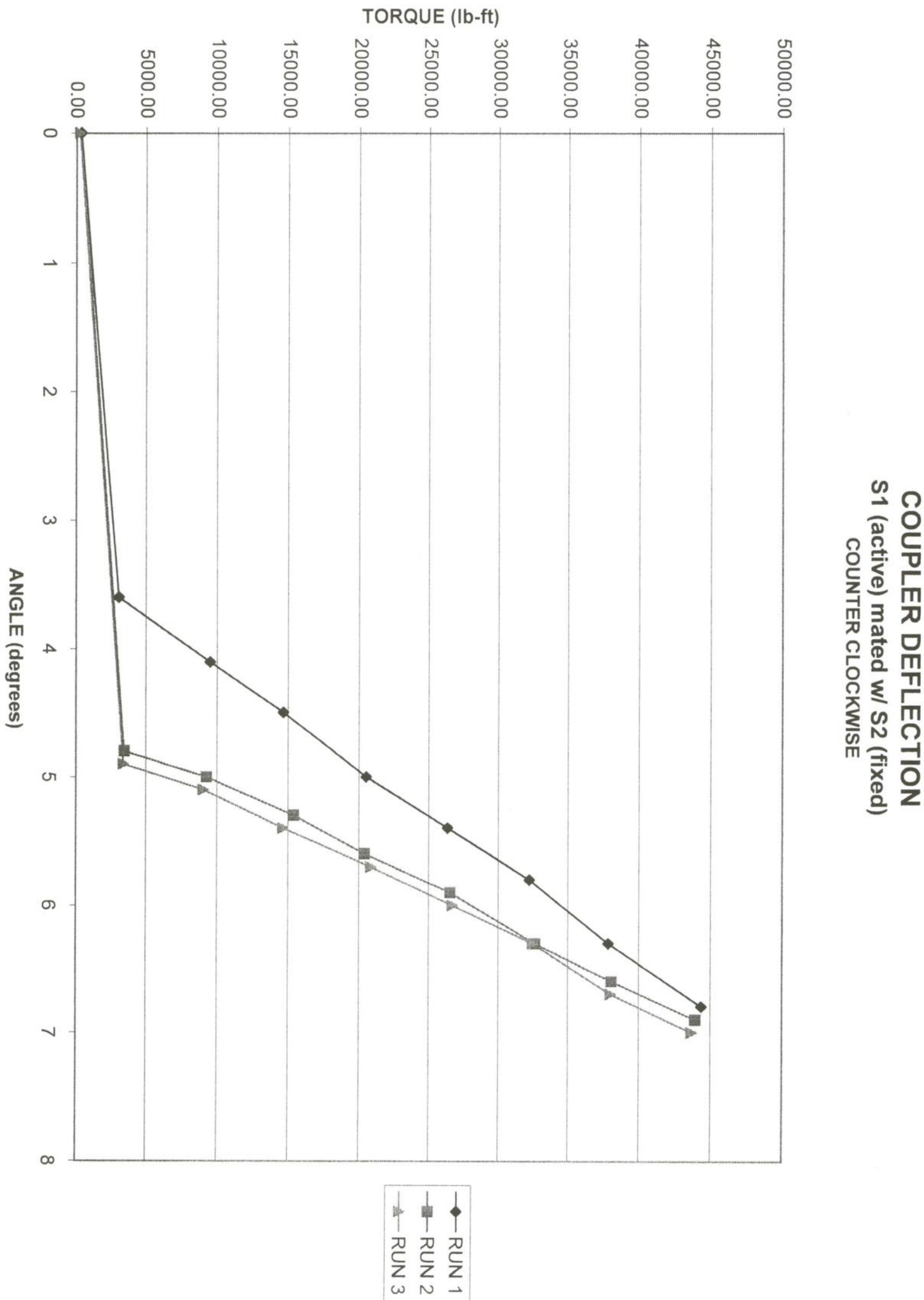
25000.00
20000.00
15000.00
10000.00
5000.00
0.00
0
2
4
6
8
10
12
ANGLE (degrees)
RUN 1
RUN 2
RUN 3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TORQUE (lb-ft)
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10000.00
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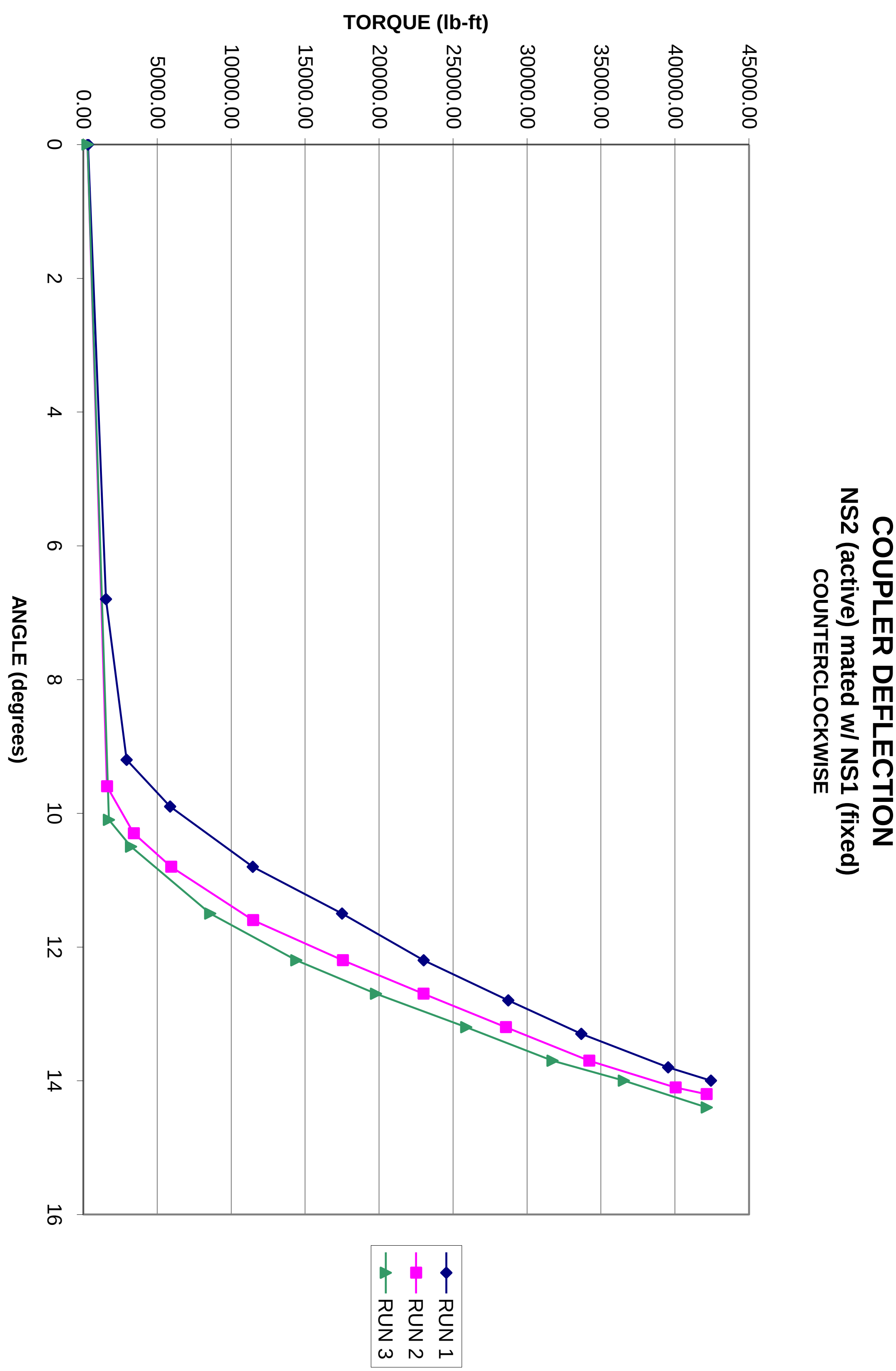
40000.00
45000.00
50000.00
0
1
2
3
4
5
6
7
8
ANGLE (degrees)
RUN 1
RUN 2
RUN 3


COUPLER DEFLECTION
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TORQUE (lb-ft)
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10000.00
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25000.00
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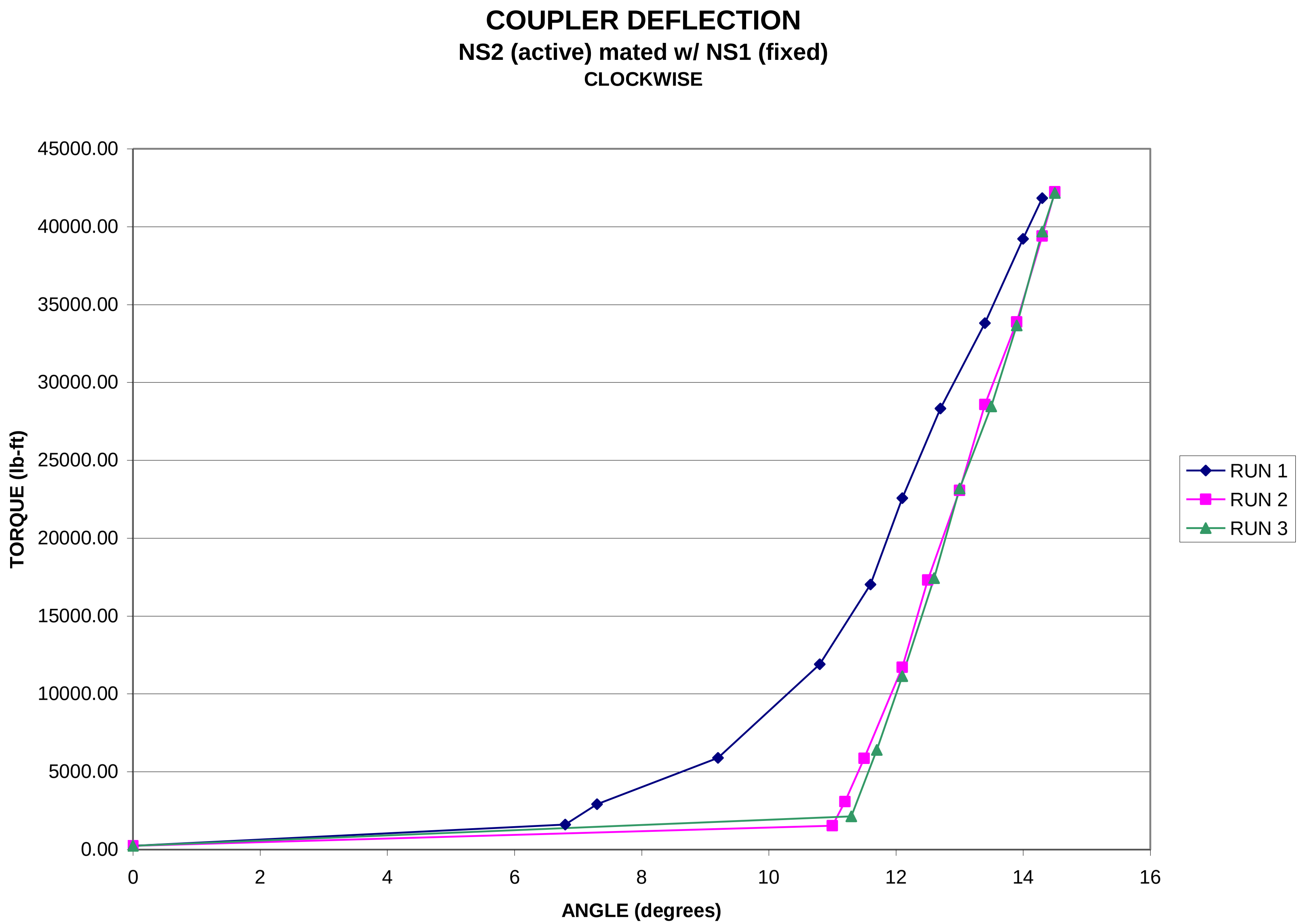
35000.00
40000.00
45000.00
ANGLE (degrees)
0
2
4
6
8
10
12
14
16
RUN 1
RUN 2
RUN 3


COUPLER DEFLECTION
NS2 (active) mated w/ NS1 (fixed)
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TORQUE (lb-ft)
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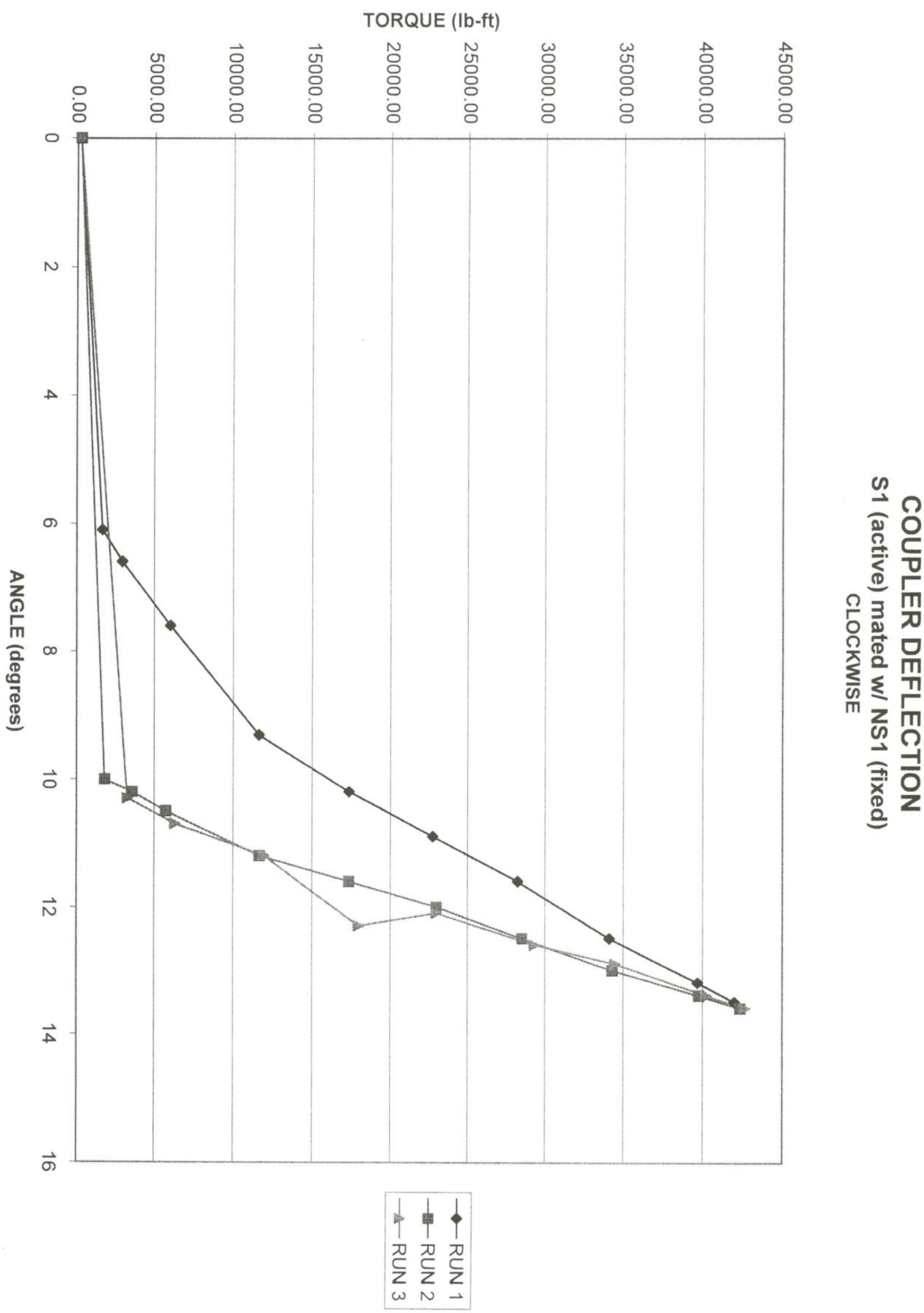
40000.00
35000.00
30000.00
25000.00
20000.00
15000.00
10000.00
5000.00
0.00
0
2
4
6
8
10
12
14
16
ANGLE (degrees)
RUN 1
RUN 2
RUN 3


COUPLER DEFLECTION
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TORQUE (lb-ft)
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10000.00
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30000.00
35000.00
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ANGLE (degrees)
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2
4
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10
12
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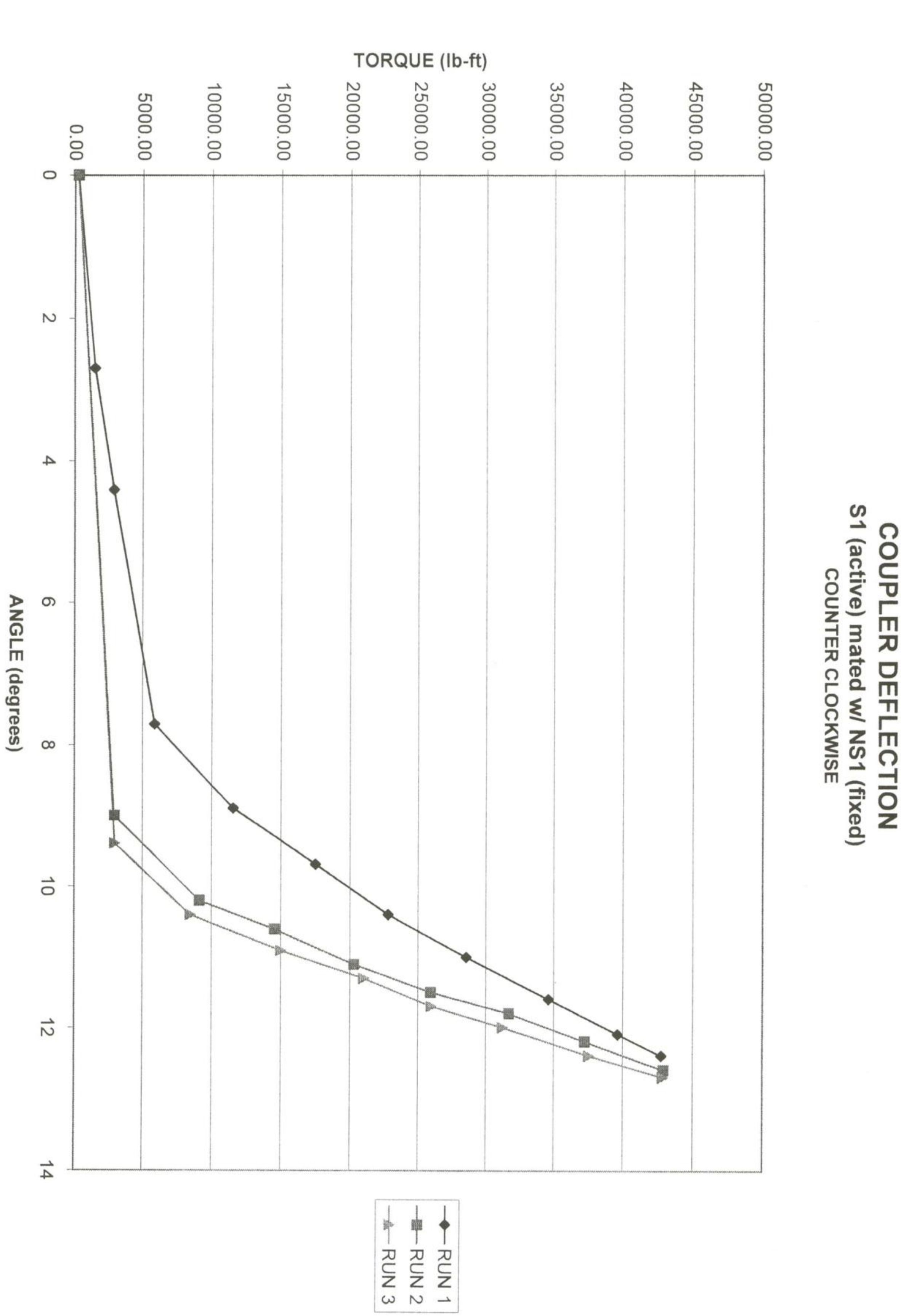
RUN 1
RUN 2
RUN 3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35000.00
40000.00
45000.00
5000.00
0
2
4
6
8
10
12
14
ANGLE (degrees)
RUN 1
RUN 2
RUN 3