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DESIGN OF THE M1 LIGHTWEIGHT STEEL TOW BAR

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MECHANICS AND STRUCTURES BRANCH

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ABSTRACT

The increased weight of today's M1A1 and M1A2 Main Battle Tanks has introduced a tow bar failure problem encountered only during field recovery operations. This problem is one of insufficient strength as the tow bar system currently used in the field was not designed for the recovery of these heavier vehicles. The direct result has been an increasing number of tow bar failures.

In a joint program between the U.S. Army Tank-Automotive Command (TACOM), the U.S. Army Materials Technology Laboratory (MTL) and Foster Miller, Inc. (FMI), a new lightweight composite tow bar was developed. During the development of this new system, it became apparent to MTL that the cost of manufacturing and materials for this composite system would be relatively high when compared to the current steel tow bar system. Accordingly, MTL developed a new lightweight tubular steel tow bar at the same time the composite tow bar program was coming to a close.

The objective of this report is to address the design of this new steel tow bar system. It was constructed of a combination of 4130 and 4340 alloy steel and possesses several key advantages over the current system. These include a 30% increase in strength, a 23% weight reduction and interchangeable legs. In addition to these advantages, the cost of this new system when in production was estimated to be comparable to that of the current tow bar system.

CONTENTS

INTRODUCTION

The increased weight of today's modified M1A1 and M1A2 Main Battle Tanks presents a dilemma which is encountered during field recovery operations. The problem is the current tow bar system used in the field has become inadequate for the recovery of these heavier vehicles. This system (shown in Figure 1) was designed in the 1950's for use in the recovery of M60 battle tanks which weigh 58 tons. This tow bar system weighs 340 lbs and requires a minimum of four soldiers for installation. Its legs are constructed of SAE 4130 steel alloy tubing with the lunette fixed to one of the legs.

With the weight of the modified M1A1s and M1A2s approaching 70 tons, the current tow bar system is experiencing a significant number of failures due to insufficient strength.

In an effort to resolve this problem, the U.S. Army Tank-Automotive Command (TACOM), U.S. Army Materials Technology Laboratory (MTL), and Foster-Miller, Inc. (FMI) developed a new lightweight composite tow bar system.[1] However, during the development of this tow bar system, it became apparent to MTL that the cost of manufacturing and materials for this new system (estimated to be 3 to 4 times the cost of the current system in production) would be too high relative to that of the currently fielded tow bar.

As an alternative, MTL independently developed a new lightweight steel tow bar at the same time the composite tow bar program was coming to a close.[2] The goals of this effort were to develop a new system which was lighter, stronger, and reasonable in price compared to the current tow bar system.

DESIGN

Load Requirements

The load requirements placed on the design of the new tow bar system were the same as those utilized in the development of the composite tow bar. A sequence of quasi-static load analyses were performed for various maneuvers and are summarized in Table 1. In this table, a straight tow refers to the recovery vehicle (M88A1 or M1A1) pulling the disabled vehicle straight forward or pushing straight backwards while the angled tow refers to the maneuvering through turns of both vehicles (see Figure 2). Also, elevation differential refers to one vehicle being above or below the other (see Figure 3).

All loads shown in Table 1 are axial loads (quasi-static). There were no torsion and only a small magnitude of bending loads[3] applied to the tow bar because all tank to tow bar connections act as universal joints allowing for rotation (see

Figure 4). The bending load applied was due to friction between the lunette and pintle during towing (see Figure 5).

In addition to the quasi-static analysis, the dynamic analysis showed a 54 ton steady-state force acting on the tow bar during a 40 mph tow on level terrain conditions. This estimate was based on existing field test data from the current tow bar system.[3] With a dynamic magnification factor of 2.0 for impulse conditions,[1] maximum dynamic loads were estimated at approximately 108 tons.

Table 1. QUASI-STATIC LOAD ANALYSIS SUMMARY

Maneuver	Maximum Tow Bar Leg Load Tension/Compression
Level Terrain, Straight Tow	34/-44 tons
Level Terrain, Angled Tow	76/-98
Sloped Terrain (30% grade), Straight Tow	43/-54
Sloped Terrain (30% grade), Angled Tow	95/-120
Stationary Turn	33/-33
Elevation Differential, Straight Tow	52/-48
Elevation Differential, Angled Tow	115/-108

From the information tabulated in Table 1, the maximum load experienced by the tow bar was 115 tons in tension and 120 tons in compression. By using the worst case load of 120 tons and introducing a factor of safety of 1.5, the design load for each leg of the tow bar was 180 tons, axial tension or compression. This 180 tons represented the maximum load each individual leg was to sustain before material began to yield.

Material Selection

All components of the currently fielded tow bar, including the tubular legs, are constructed of SAE 4130 alloy steel. In the new design, this alloy was used only for the tube material due to its strength, toughness, and heat treatability. A nominal heat treatment would generate the 150 ksi ultimate strength and 122 ksi yield strength, σ_{y1}, which were necessary to provide the strength required to endure the severe tow bar service environment.

The remaining components of the new design (male and female end fittings, lunette, and pins) were constructed of SAE 4340 alloy steel due to the existence of higher stresses in these components. An ultimate strength of 180 ksi and yield strength, σ_{y2}, of 160 ksi was required for this material.

Tube Selection

A tube with sufficient strength was designed utilizing the load and material criteria from above. The calculated dimensions of this tube were exact and would be costly to fabricate on a prototype basis. Therefore, for this reason, the dimensions calculated were regarded as minimum values and tubing of the nearest larger standard size was used in the fabrication of all prototypes.

To be able to sustain the tensile and compressive loads of 180 tons, the minimum required cross sectional area of the tube, A_{min}, was determined using the following equation:

$$A_{min} = \frac{P}{\sigma_{yl}} \qquad (1)$$

where P = Applied Load
σ_{yl} = Yield Strength of 4130 Steel.

After substituting the known values of P and σ_{yl}, A_{min} was calculated to be 2.9508 square inches. This area must be distributed in such a manner to resist both column and shell buckling resulting from compression loading.

To design against column buckling (for a pinned-pinned column), Euler's Equation[4] was used and is shown below.

$$P_{cr} = \pi^2 EI/L^2 \qquad (2)$$

where P_{cr} is the critical buckling load
E is the Modulus of Elasticity
I is the Moment of Inertia
$= \pi(R_o^4 - R_i^4)/4$ for a tube
L is the length of the column.

Because the critical buckling load (180 tons), modulus of elasticity (30×10^6 psi), and length of the column (same as the current tow bar, 69 inches, see Figure 6) were known, the only unknown was the moment of inertia, I. Euler's Equation was then manipulated to determine the ideal inner and outer radii (R_i & R_o, respectively) of the tube. This equation is shown below.

$$R_i^4 = R_o^4 - (4P_{cr}L^2)/(\pi^3 E) \qquad (3)$$

Through an iterative process using Equation 3, Table 2 was generated by substituting values for R_o and calculating R_i.

Table 2. VALUES GENERATED FROM MANIPULATED EULER'S EQUATION

R_o(in.)	R_i(in.)	Thickness(in.)	Area(in.^{2})	Weight(lbs)
1.9	1.422	0.478	4.989	97.42
2.0	1.630	0.370	4.219	82.38
2.1	1.800	0.300	3.676	71.78
2.2	1.951	0.249	3.247	63.40
2.3	2.089	0.211	2.909	56.80

An initial value of 1.9 inches for R_o was selected because it was slightly less than the R_o of the current tow bar system. The final value substituted for R_o was 2.3 inches. Values greater than and including 2.3 resulted in a tube cross section which did not meet the required cross sectional area necessary to prevent yielding. Therefore, it was assumed the outer dimension of the tube falls between the R_o values of 2.2 and 2.3 inches.

Once the inner and outer radii were determined, the cross sectional area and weight of the tube was calculated. The weights of the various tube sizes were calculated by multiplying the cross sectional areas by the length of the tube (69 in.) and the density of steel (0.283 lb/in.^{3}).

With reference to Table 2, standard tubing with the dimensions listed in Table 3 was selected.

Table 3. DIMENSIONS OF STANDARD TUBE SELECTED

R_o(in.)	R_i(in.)	Thickness(in.)	Area(in.^{2})	Weight(lb)
2.25	2.000	0.250	3.338	65.17

With a cross sectional area of 3.338 in^2, the stress induced in the tube by the 180 tons (360,000 lbs) was calculated to be 108 ksi. (calculated from Equation 1, all calculations can be found in Appendix A). This stress is less than the yield strength of the material (σ_{Y1} = 122 ksi.), therefore, the geometric and material properties of this tube enable it to support the required axial loads.

The next step was to determine the critical buckling load for the tube. Using Equation 2, the critical buckling load P_{cr} was calculated to be approximately 194 tons (or 387,700 lbs). This load of 194 tons is greater than the required 180 tons, therefore, this tube surpasses this column buckling load requirement.

The final check was to calculate the critical shell buckling stress due to compressive loading. Shell buckling is a localized collapse of a thin-walled tube (shell) subjected to compression loading. The axial stress required to cause this type of failure was calculated from Equation 4, shown below.[4]

$$\sigma_{cr} = Et/(3^{1/2}R_{av}(1 - \mu^2)^{1/2}) \qquad (4)$$

where σ_{cr} is the critical shell buckling stress
E is the Modulus of Elasticity
t is the wall thickness of the tube
R_{av} is the average radius of the tube
μ is Poisson's Ratio.

Using Equation 4, the critical shell buckling stress was 2,131 ksi. The maximum axial stress of 108 ksi was well below this value. Therefore, the shell buckling requirement was easily satisfied.

Lunette

The design of the lunette was completed during the development of the composite tow bar[1] and is shown in Figure 7. This particular lunette design offered a 50% weight reduction (new lunette weighs 24 lbs versus 48 lbs for the old). Prior to the new lightweight steel tow bar design, the structural integrity of the lunette was previously determined using finite element analysis (by MTL and FMI) and successfully proven during field testing of the composite tow bar.[5]

Male End Fitting

Having selected the tube size, the base of the end fitting had to fit into the inner diameter of the tube and also be chamfered for welding to the tube. Therefore, the outer diameter of the immediate base was undersized by 0.03 inches and a 0.25 inch x 45° chamfer was machined along the top edge as shown in Figure 8.

The design of the male portion of the end fitting required that bearing, tensile and shear stresses be considered. Since a 1.5 inch diameter pin is used in the current system to attach the end fitting to the clevis, the new design must also use the same diameter pin. In addition, the maximum allowable thickness was fixed at 1.75 inches due to the slot width of the receiving clevis (see Figure 9).

Table 4 lists the maximum values of bearing, tensile and shear stresses experienced by the male end fitting at the maximum load of 180 tons. (See Appendix A for all calculations.)

Since the yield strength of the 4340 steel alloy was greater (σ_{y2} = 160 ksi.) than the bearing, tensile, and shear stresses, the design of the new male end fitting was adequate.

Table 4. STRESSES EXPERIENCED BY MALE END FITTING AT A MAXIMUM LOAD OF 180 TONS

Stress Type	Max. Value (ksi)
Bearing	137.2
Tensile	128.6
Shear	93.5

Female End Fitting

The design of the female end fitting was similar to that of the male end fitting. The base geometries of the two end fittings are identical and the same stresses (bearing, tensile and shear) had to be calculated. A major difference between the end fittings was that the female end fitting had two holes for different size pins (2-inch and 1.41-inch diameter pins) used to attach the legs to the lunette. The small pin prevents rotation of the lunette in the latter's plane during recovery operations (see Figure 10). In the worst case situation, only the larger pin would be carrying the entire load because the smaller pin would not be installed in the same end fitting at all times. Therefore, the design was based on the fact that the larger pin would carry the majority of load while the smaller pin would lock one of the tow bar legs in place after installation.

For investigative purposes, under ideal loading conditions, the assumption was made that both pins begin loading simultaneously. Therefore, calculations in Appendix A showed the stresses within that section of the female end fitting based on the above assumption.

Calculations were made to determine the stresses within the female end fitting for the case where all loading was sustained only by the larger pin. Table 5 lists the values of bearing, tensile, and shear stresses at a maximum axial load of 180 tons. (See Appendix A for all calculations.)

The yield strength of the 4340 steel alloy was greater (σ_{y2} = 160 ksi) than the bearing, tensile, and shear stresses calculated for this female end fitting design. Therefore, this end fitting would adequately support all loading conditions and is shown in Figure 11.

Table 5. STRESSES EXPERIENCED BY FEMALE END FITTING AT A MAXIMUM LOAD OF 180 TONS.

Stress Type	Max. Value (ksi)
Bearing	120.0
Tensile	96.0
Shear	120.0

Welds

The end fittings were attached to the tubes with welds. The effective weld area was considered to be the same as the cross sectional area of the tube - 3.338 in.^{2} (see Figure 12). Utilizing this area, the axial stress in the weld was determined to be 107.8 ksi. Therefore, with the stress at this level, a high strength welding rod possessing a yield strength in the 120 to 130 ksi range was mandatory.

For each tow bar leg there were two welds, one connecting each end fitting to the ends of the tube. The welds were single V butt welds with 90° bevels (accomplished by machining a 45° chamfer on the end fittings and the ends of the tube) along the circumference of the tube ends with no root opening (see Figure 13). It was strongly recommended that two passes be made for each weld to ensure full penetration.

Prior to welding the second end fitting into place, care was taken to ensure that the faces of each end fitting were parallel to each other. This was necessary since the end fittings must lie in the same plane for proper alignment during installation of the tow bar (see Figure 14).

It was also highly recommended that all heat treating of the tubes and end fittings be performed after welding had been completed. This ensured that there were no localized heat effects as a result of welding and guaranteed that the proper heat treatment was homogeneous throughout the structure. The properties of the 4130 and 4340 steel alloys were selected so that they may be obtained at the same tempering temperature during heat treatment.

Pins

Attachment of the tow bar legs to the lunette required three pins. These incluue two 2-inch diameter and one 1.4-inch diameter pins as shown in Figure 15. All pins were fabricated from 4340 alloy steel. As previously discussed, the larger pins transferred the majority of the load while the single smaller pin acted to prevent rotation of the lunette. Table 6 summarizes the maximum stresses experienced by the pins during towing.

Table 6. MAXIMUM STRESSES EXPERIENCED BY THE PINS DURING TOWING OPERATIONS

Pin Diameter(in.)	Bearing Stress(ksi)	Shear Stress(ksi)
2	120.0	57.3
1.4	46.6	31.8

As shown in Table 6, the stresses induced in the pins during towing do not approach the yield strength of the material (σ_{y2} = 160 ksi), therefore, these pins (which are larger than the current pins) were adequately designed for the given load conditions.

CONCLUSIONS

This effort resulted in a new steel tow bar design (final drawings in Appendix B) which was lighter, stronger, and offered interchangeability of identical legs at a reasonable cost.

By using the combination of 4130 and 4340 high strength alloys, the weight was reduced from 340 lbs for the current tow bar to 268 lbs for the new tow bar system, a 23% weight reduction.

A 30% strength increase was accomplished in two steps. The first was an expansion of the tube diameter (both inner and outer) while maintaining the minimum cross-sectional area. This change resulted in increased buckling strength to resist the compressive loads experienced by a tow bar in service. The second step was the selection of the proper heat treatment. From the correct heat treatment, the materials obtain the strength required to successfully resist the tensile loads applied.

Added features of the new design are the leg interchangeability and separate lunette. These offer the soldier simplified assembly and transportability of the tow bar whereas today's system is less manageable (ie., requires additional soldiers for installation). In contrast to the current tow bar system's replacement procedure, if there were a component failure of the new system, only the failed component would have to be replaced rather than the entire system.

In addition to the advantages mentioned above, the estimated cost of this new system in production ($750) is comparable to that of the current tow bar system ($500).

Following the fabrication of several new prototype steel tow bar systems, laboratory and field tests were successfully conducted at MTL and Aberdeen Proving Ground, Aberdeen, MD, respectively.[6]

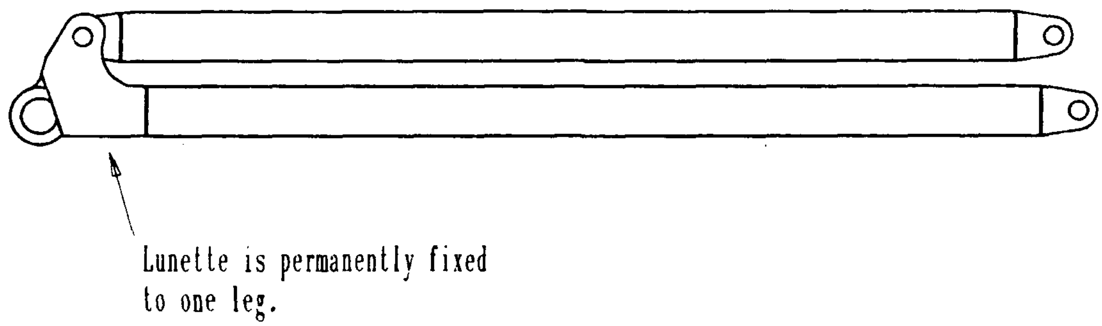


Figure 1. Current tow bar system.

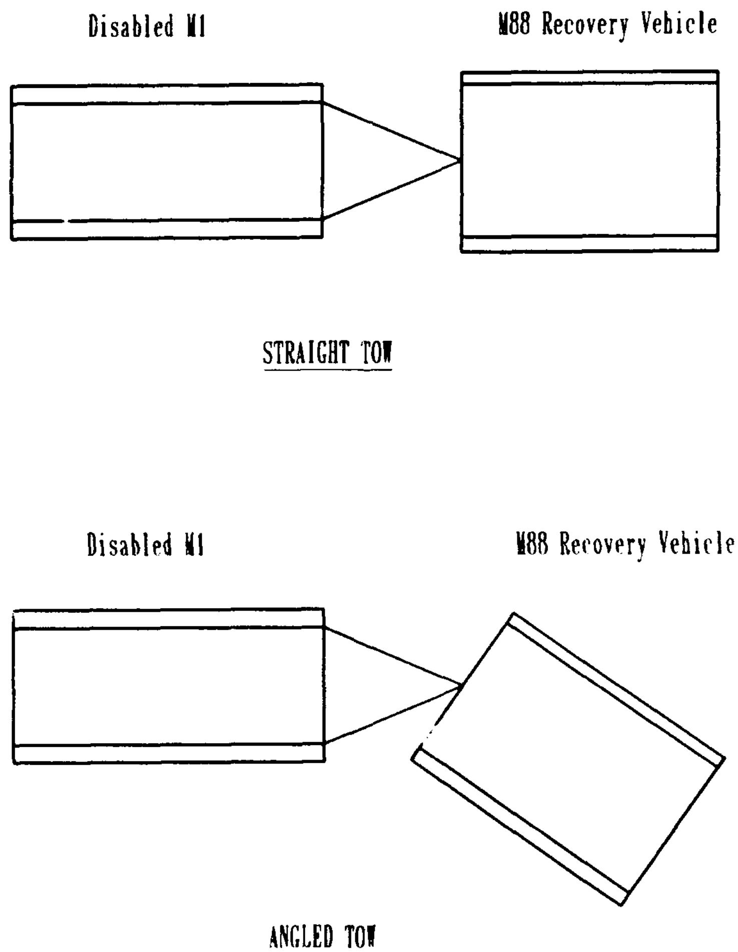


Figure 2. Illustration of straight and angled tows.

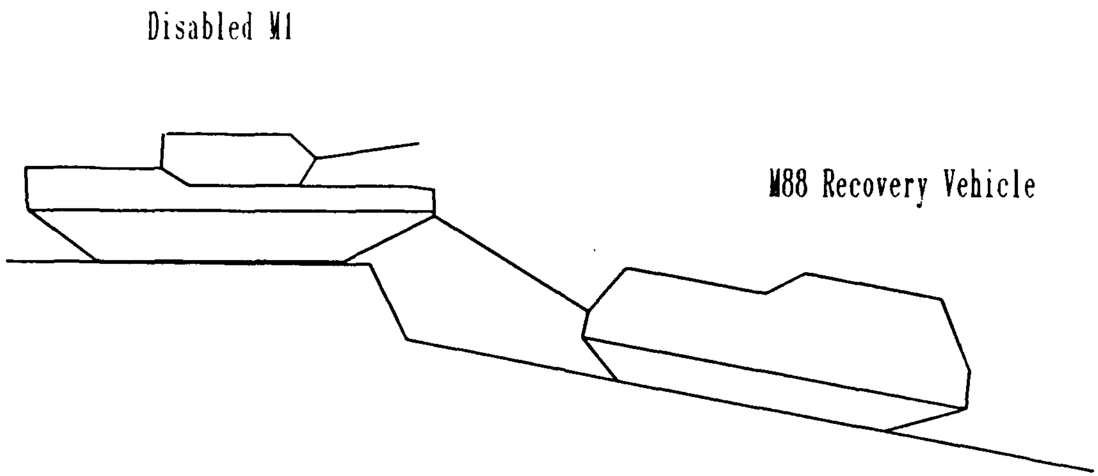
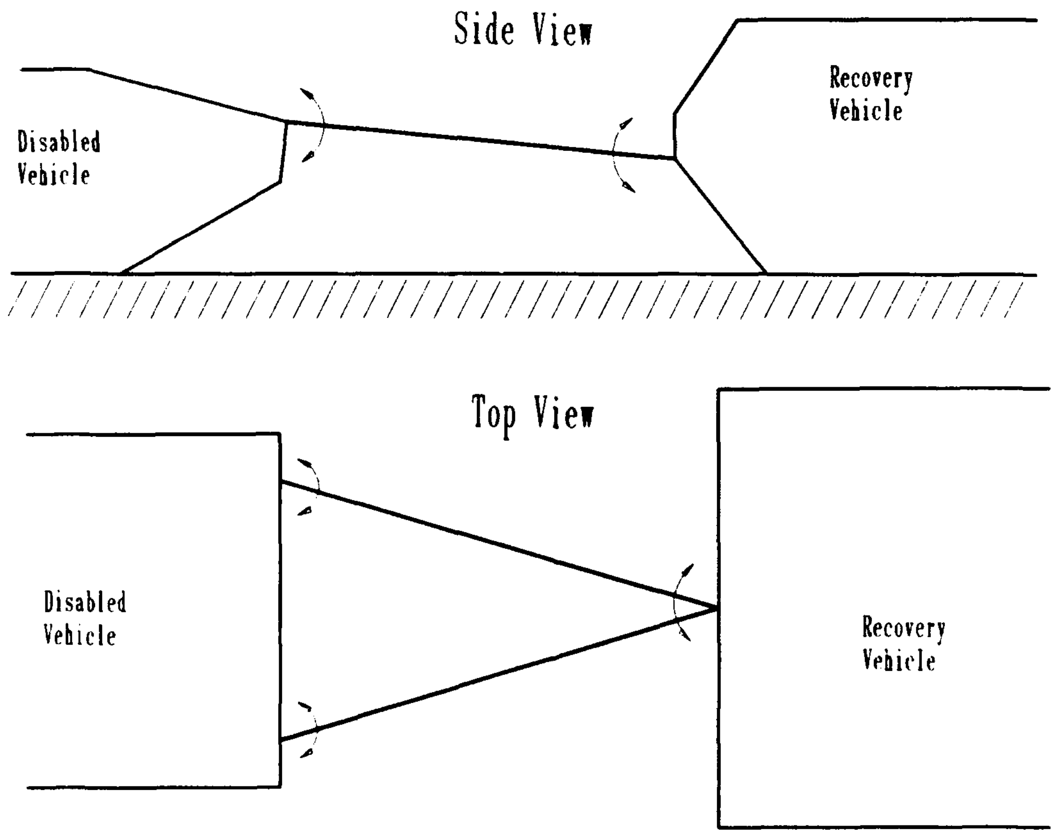


Figure 3. Elevation differential between vehicles.



The arrows indicate areas of rotation.

The pintle to which the tow bar attaches on the recovering vehicle also allows for 360 degrees rotation.

Figure 4. Tow bar connections allow for rotation to avoid torsion and reduce bending.

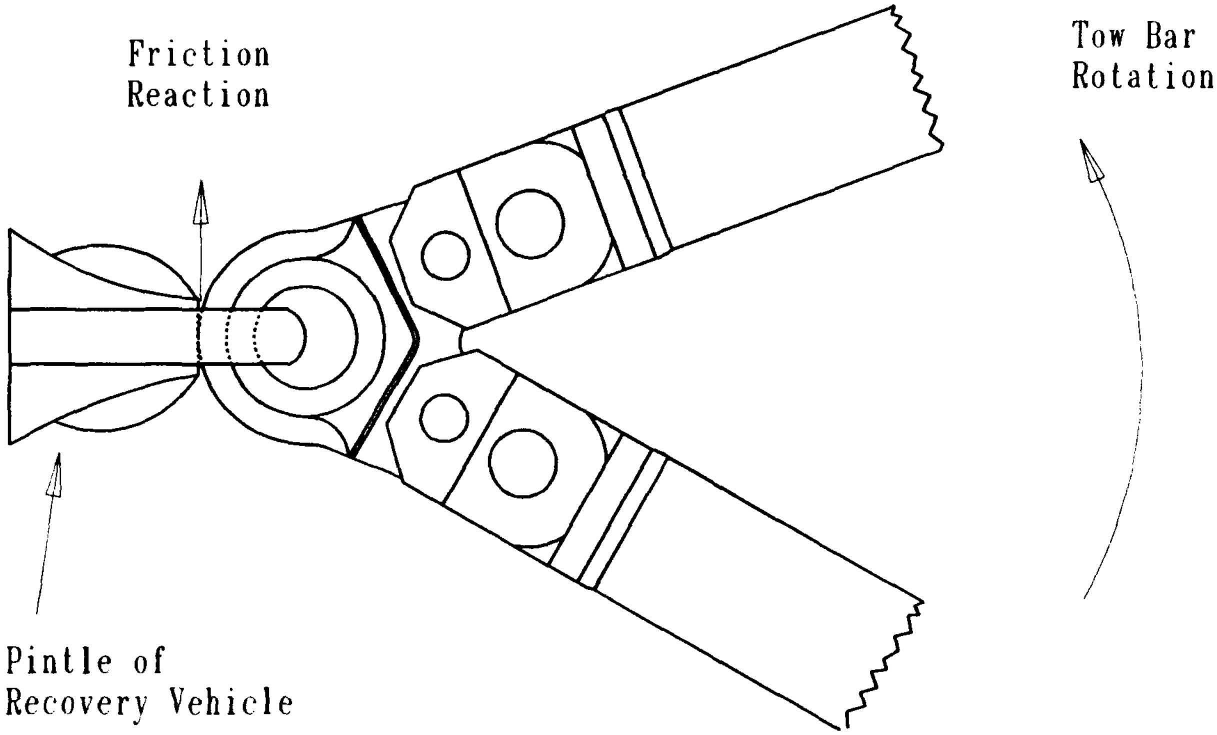
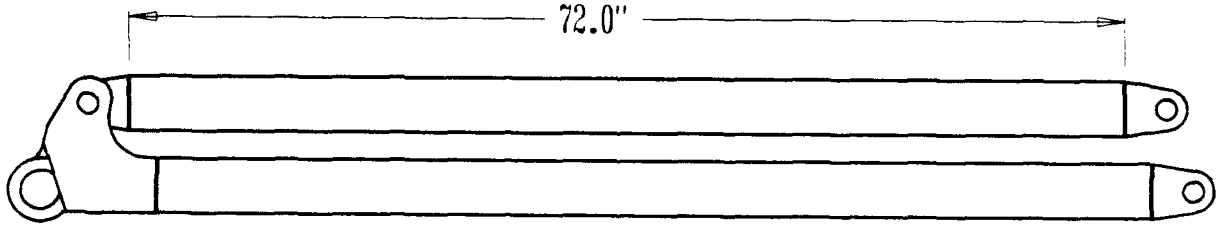


Figure 5. Some bending is induced during towing due to friction between the lunette and pintle.



Note:
The actual tube length of the new tube was 69.0 inches.
This reduction of 3 inches from the current tow bar leg is due
to an increase in the length of the new end fittings.

Figure 6. Length of current tow bar tube.

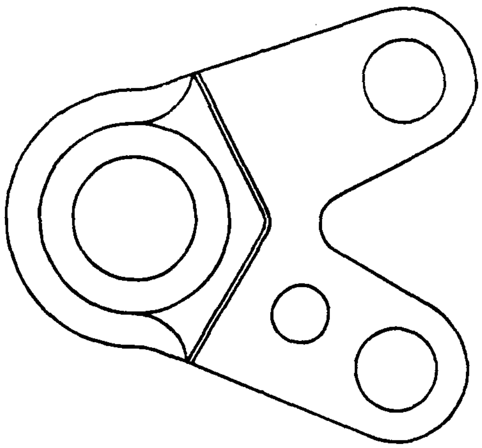


Figure 7. New independent lunette design.

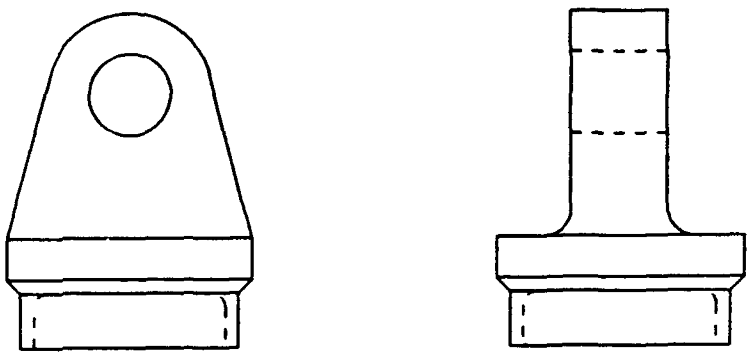


Figure 8 New male end fitting.

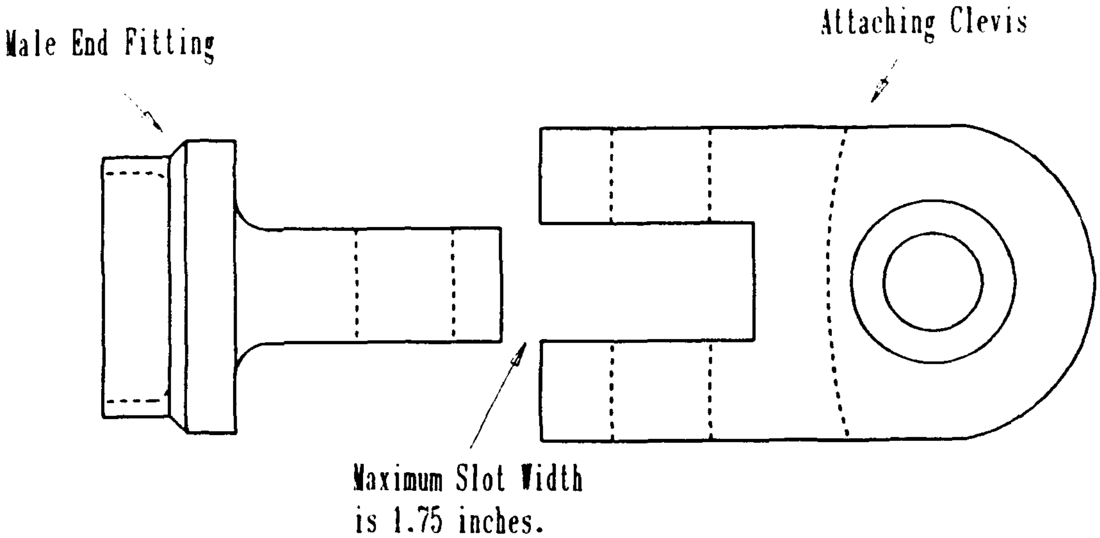


Figure 9 Receiving clevis shows slot width.

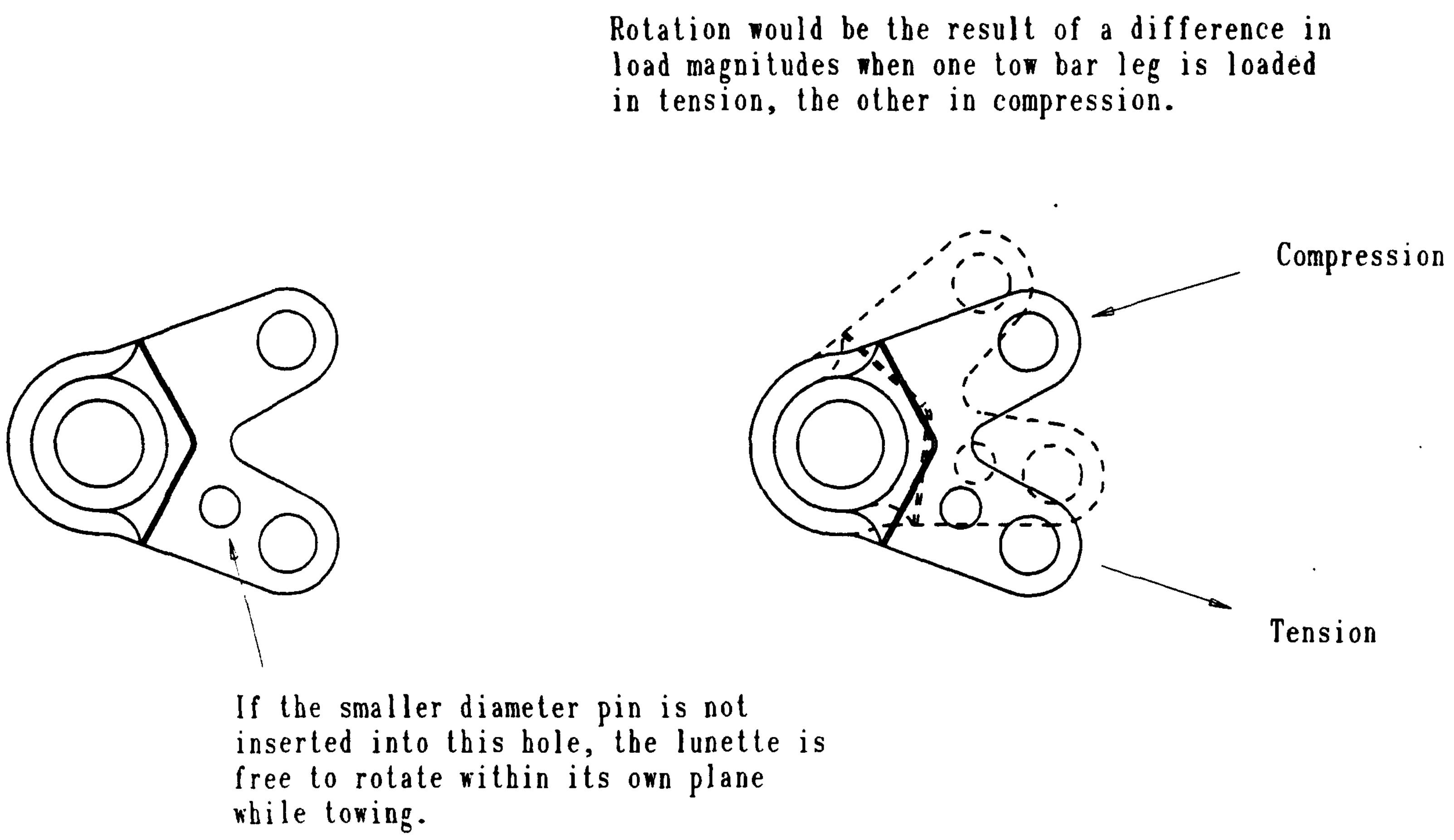


Figure 10. Three pin installation.

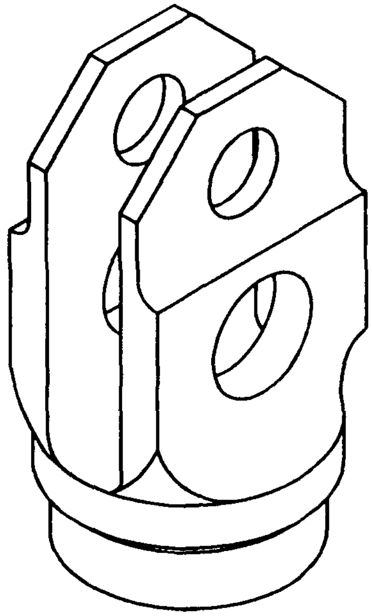


Figure 11. New female end fitting.

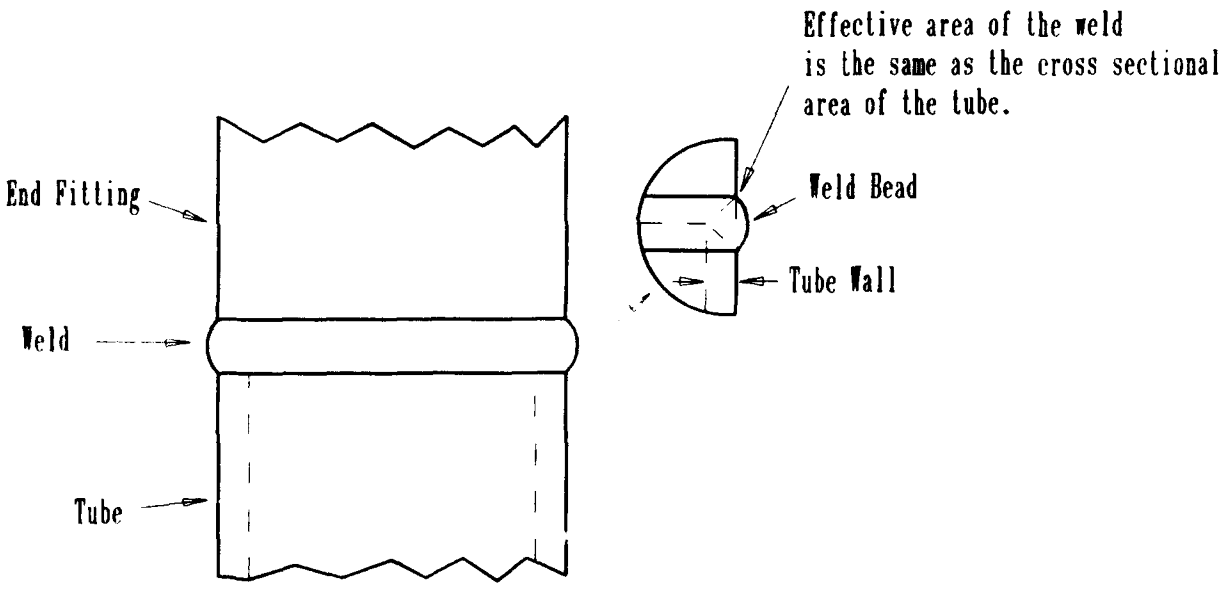


Figure 12. Weld area at tube and end fitting interface.

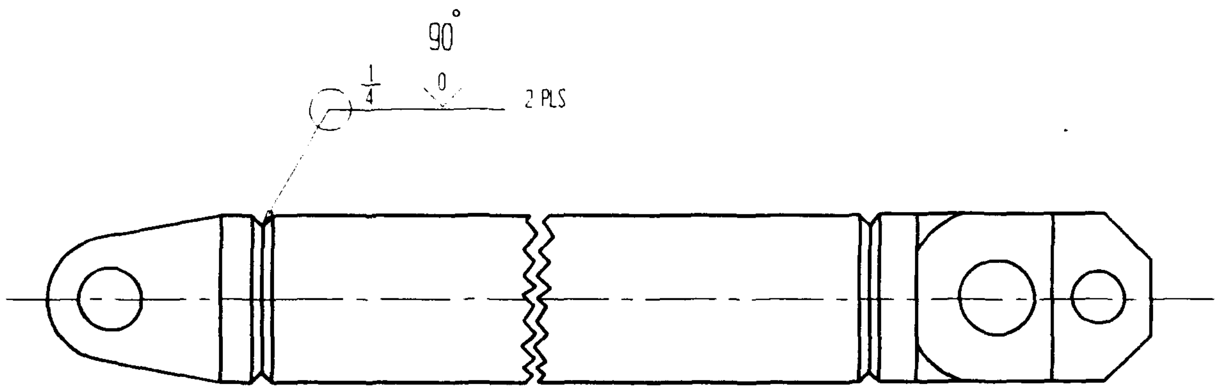


Figure 13. Weldment drawing.

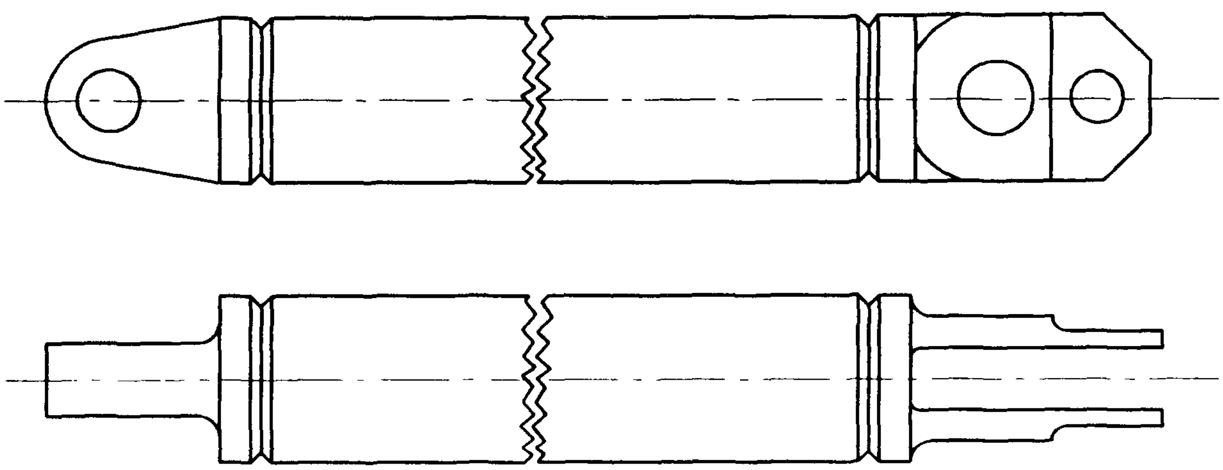


Figure 14. End fittings must lie in the same plane for proper installation of the tow bar.

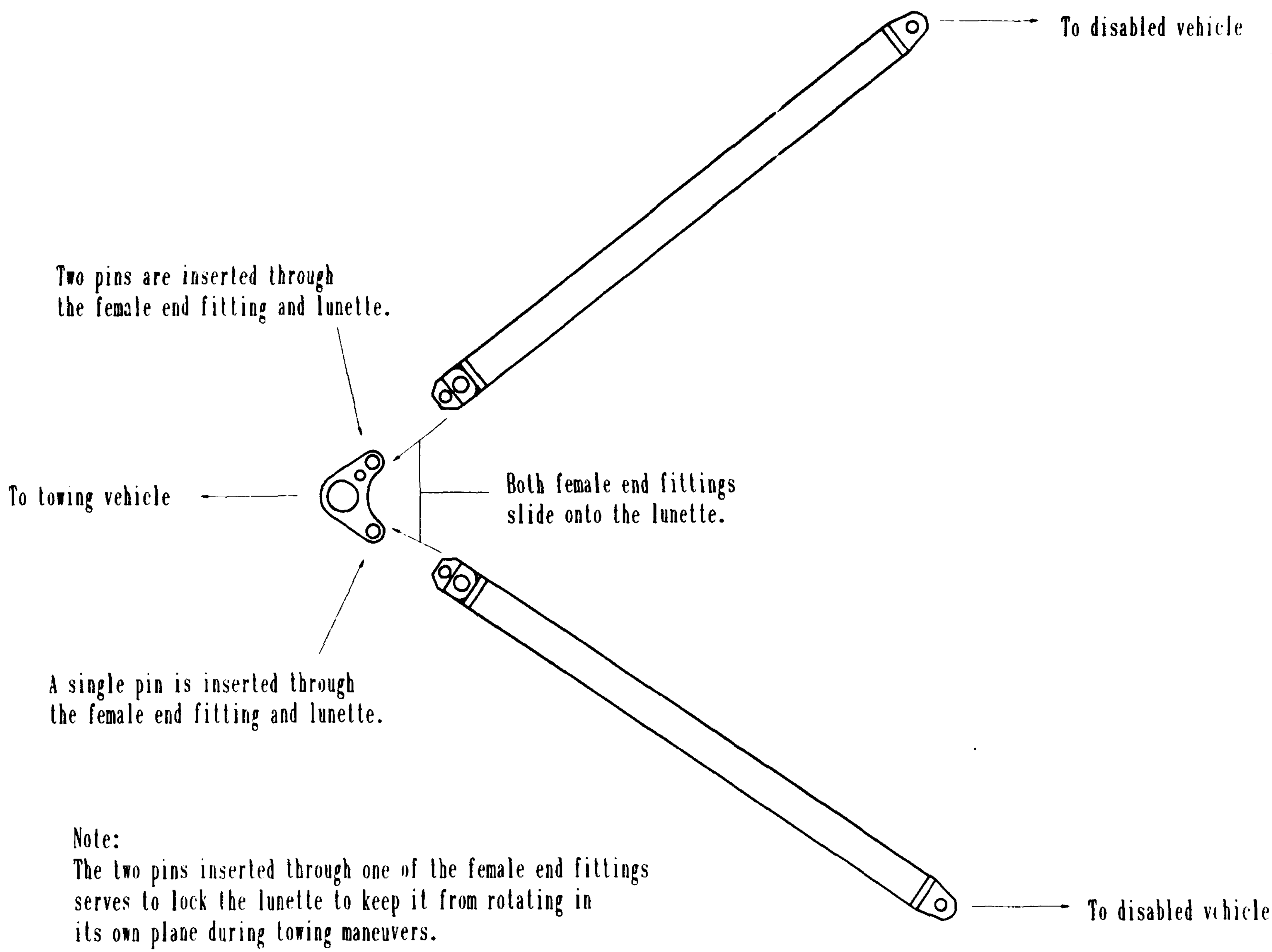


Figure 15. Assembly of new tow bar system.

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APPENDIX A

CALCULATIONS

NOMENCLATURE

σ	-	Axial stress
σ_b	-	Bearing stress
σ_s	-	Shear stress
σ_{s1}	-	Shear strength of 4130 alloy steel
σ_{s2}	-	Shear strength of 4340 alloy steel
σ_{cr}	-	Critical shell buckling stress
σ_{y1}	-	Yield strength of 4130 alloy steel
σ_{y2}	-	Yield strength of 4340 alloy steel
P	-	Applied load
P_{cr}	-	Critical buckling load
A	-	Cross sectional area
A_{min}	-	Minimum cross sectional area
A_{tot}	-	Total cross sectional area at pin hole locations
E	-	Modulus of Elasticity
I	-	Moment of Inertia
μ	-	Poisson's Ratio
δ	-	Density of Steel
L	-	Length of column
l	-	Length of single cross section next to a hole
w	-	Slot width between prongs of female end fitting
t	-	Thickness of sections
t_p	-	Thickness of section about large pin
t_s	-	Thickness of section about small pin
h	-	Length of shear planes tangent to pin holes
R_i	-	Inner radius of the tube
R_o	-	Outer radius of the tube
R_{av}	-	Average radius of the tube
r_1	-	Radius of large pin
r_2	-	Radius of small pin
d	-	Standard pin diameter used in the field
d_l	-	Large pin diameter
d_s	-	Small pin diameter

1. MATERIAL PROPERTIES

Material: 4130

Ultimate Strength, $\sigma_{u1} = 150$ ksi

Yield Strength, $\sigma_{y1} = 122$ ksi

Shear Strength, $.75\sigma_{y1} = \sigma_{s1} = 91.5$ ksi

Material: 4340

Ultimate Strength, $\sigma_{u2} = 180$ ksi

Yield Strength, $\sigma_{y2} = 160$ ksi

Shear Strength, $.75\sigma_{y2} = \sigma_{s2} = 120$ ksi

2. TUBE — Material: 4130

a) Axial Stress

$$\sigma = P/A = 360{,}000/\pi(2.25^2 - 2^2)$$

$$\sigma = 107{,}851 \text{ psi}$$

b) Column Buckling

$$P_{cr} = \pi^2 EI/L^2$$

$$P_{cr} = \pi^2(30E6)[\pi(2.25^4 - 2^4)]/(4(76)^2)$$

$$P_{cr} = 387{,}685 \text{ lbs}$$

c) Shell Buckling

$$\sigma_{cr} = Et/[3^{1/2}R_{av}(1 - \mu^2)^{1/2}]$$

$$\sigma_{cr} = (30E6)0.25/[3^{1/2}(2.125(1 - 0.293^2)^{1/2})]$$

$$\sigma_{cr} = 2.1E6 \text{ psi}$$

d) Weight

$$W = AL\delta$$

$$W = \pi(2.25^2 - 2^2)(69)(0.283)$$

$$W = 65.2 \text{ lbs}$$

3. MALE END FITTING — Material: 4340, Weight : 11 lbs

a) Pin Bearing

$$A = dt = 1.5(1.75) = 2.625 \text{ in.}^2$$

$$\sigma_b = P/A = 360{,}000/2.625$$

$$\sigma_b = 137{,}143 \text{ psi}$$

b) Tensile Stress About Hole

$$A = 2tl = 2(1.75)(0.8) = 2.800 \text{ in.}^2$$

$$\sigma = P/A = 360{,}000/2.80 = 128{,}571 \text{ psi}$$

c) Shear Pull Out

$$A = 2th = 2(1.75)(1.1) = 3.850 \text{ in.}^2$$

$$\sigma_s = P/A = 360{,}000/3.850 = 93{,}507 \text{ psi}$$

4. FEMALE END FITTING

Material: 4340
Weight : 12 lbs

I. Worst Case - Large Pin Taking All Load

a) Pin Bearing

$$A = 2dt = 2(2)(0.75) = 3.00 \text{ in.}^2$$

$$\sigma_b = P/A = 360{,}000/3.00 = 120{,}000 \text{ psi}$$

b) Tensile Stress About Hole

$$A = 4lt = 4(1.25)(0.75) = 3.75 \text{ in.}^2$$

$$\sigma = P/A = 360{,}000/3.75 = 96{,}000 \text{ psi}$$

c) Shear Pull Out

$$A = 4th = 4(0.75)(1.43) + 4(0.4)(0.5) = 5.11 \text{ in.}^2$$

$$\sigma_s = P/A = 360{,}000/5.11 = 70{,}415 \text{ psi}$$

II. Case Where Both Pins Begin Loading At Same Instant

a) Pin Bearing

$$A_{tot} = 2d_l t_l + 2d_s t_s = 2(2)(0.75) + 2(1.4)(0.4)$$

$$A_{tot} = 4.120 \text{ in.}^2$$

$$\sigma_b = P/A = 360{,}000/4.12 = 87{,}379 \text{ psi}$$

b) Axial Stress About Holes

$$A_{tot} = 4(tl)_l + 4(tl)_s$$

$$A_{tot} = 4(0.75)(1.25) + 4(0.4)(1.55)$$

$$A_{tot} = 6.23 \text{ in.}^2$$

$$\sigma = P/A = 360{,}000/6.23$$

$$\sigma = 57{,}784 \text{ psi}$$

c) Shear Pull Out

$$A_{tot} = 4(th)_l + 4(th)_s$$

$$A_{tot} = 4(0.75)(1) + 4(0.4)(1.4)$$

$$A_{tot} = 5.24 \text{ in.}^2$$

$$\sigma_s = P/A = 360{,}000/5.24$$

$$\sigma_s = 68{,}702 \text{ psi}$$

5. WELDS

$$A = \pi(R_o^2 - R_i^2)$$

$$A = \pi(2.25^2 - 2^2)$$

$$A = 3.338 \text{ in.}^2$$

$$\sigma = P/A = 360{,}000/3.338$$

$$\sigma = 107{,}852 \text{ psi}$$

With the utilization of a high strength weld rod, this stress is acceptable.

6. TWO INCH DIAMETER PIN Weight: 3.75 lbs

a) Shear Stress

$$A_s = 2\pi r_1^2 = 2\pi(1)^2$$

$$A_s = 6.283 \text{ in.}^2$$

$$\sigma_s = P/A = 360{,}000/6.283$$

$$\sigma_s = 57{,}300 \text{ psi}$$

b) Bearing Stress

$$A = 2w = 2(1.5)$$

$$A = 3.0 \text{ in.}^2$$

$$\sigma_b = P/A = 360{,}000/3.0$$

$$\sigma_b = 120{,}000 \text{ psi}$$

7. 1.4 INCH DIAMETER PIN Weight: 1.5 lbs

From 3.II.a above, the maximum load seen by this pin would be:

$$P = \sigma A = 87,379(1.4)(0.4)(2)$$

$$P = 97,865 \text{ lbs}$$

a) Bearing Stress

$$A = d_s w = (1.4)(1.5)$$

$$A = 2.1 \text{ in.}^2$$

$$\sigma_b = P/A = 97,865/2.1$$

$$\sigma_b = 46,602 \text{ psi}$$

b) Shear Stress

$$A = 2\pi r_2{}^2 = 2\pi(0.7)^2$$

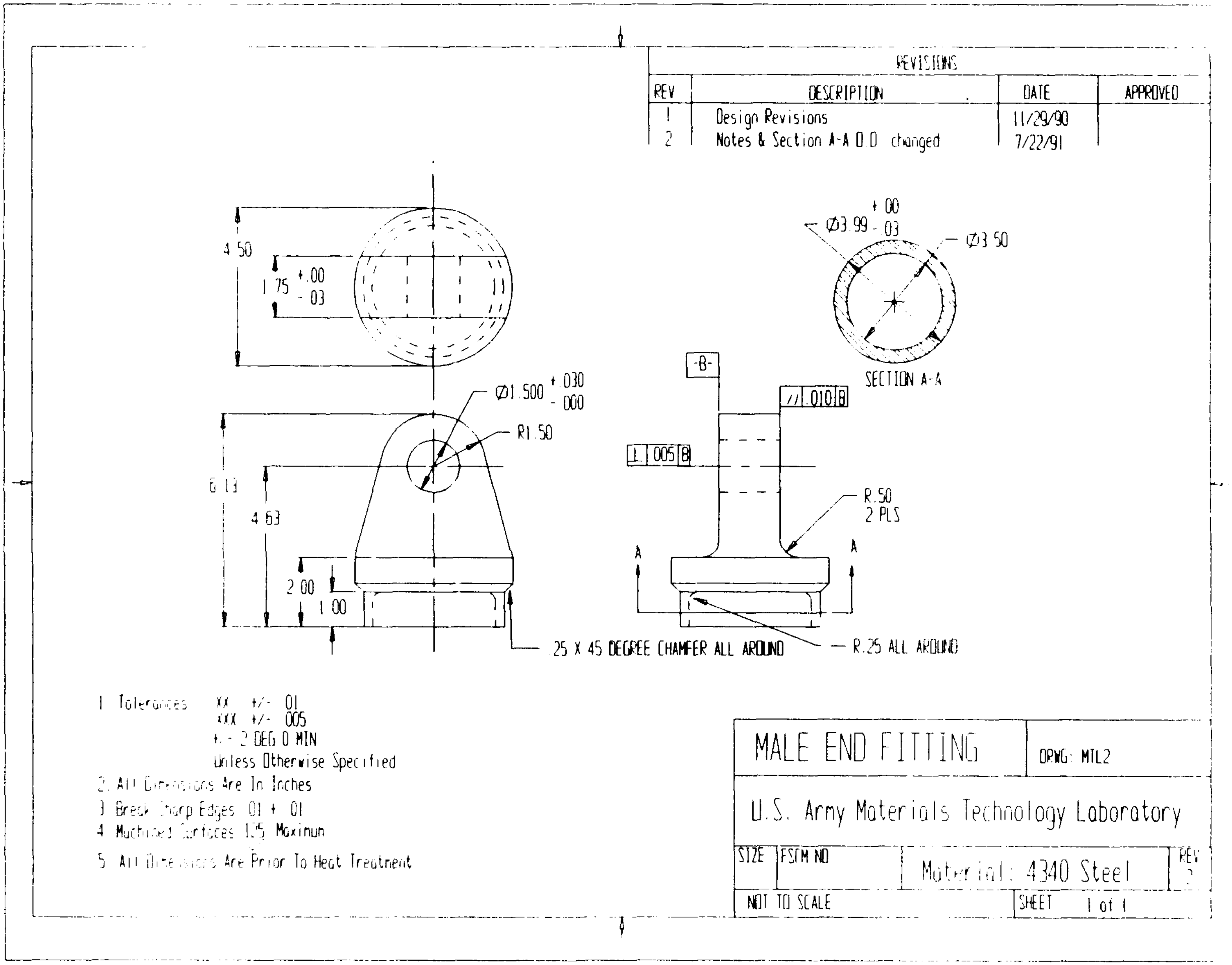
$$A = 3.078 \text{ in.}^2$$

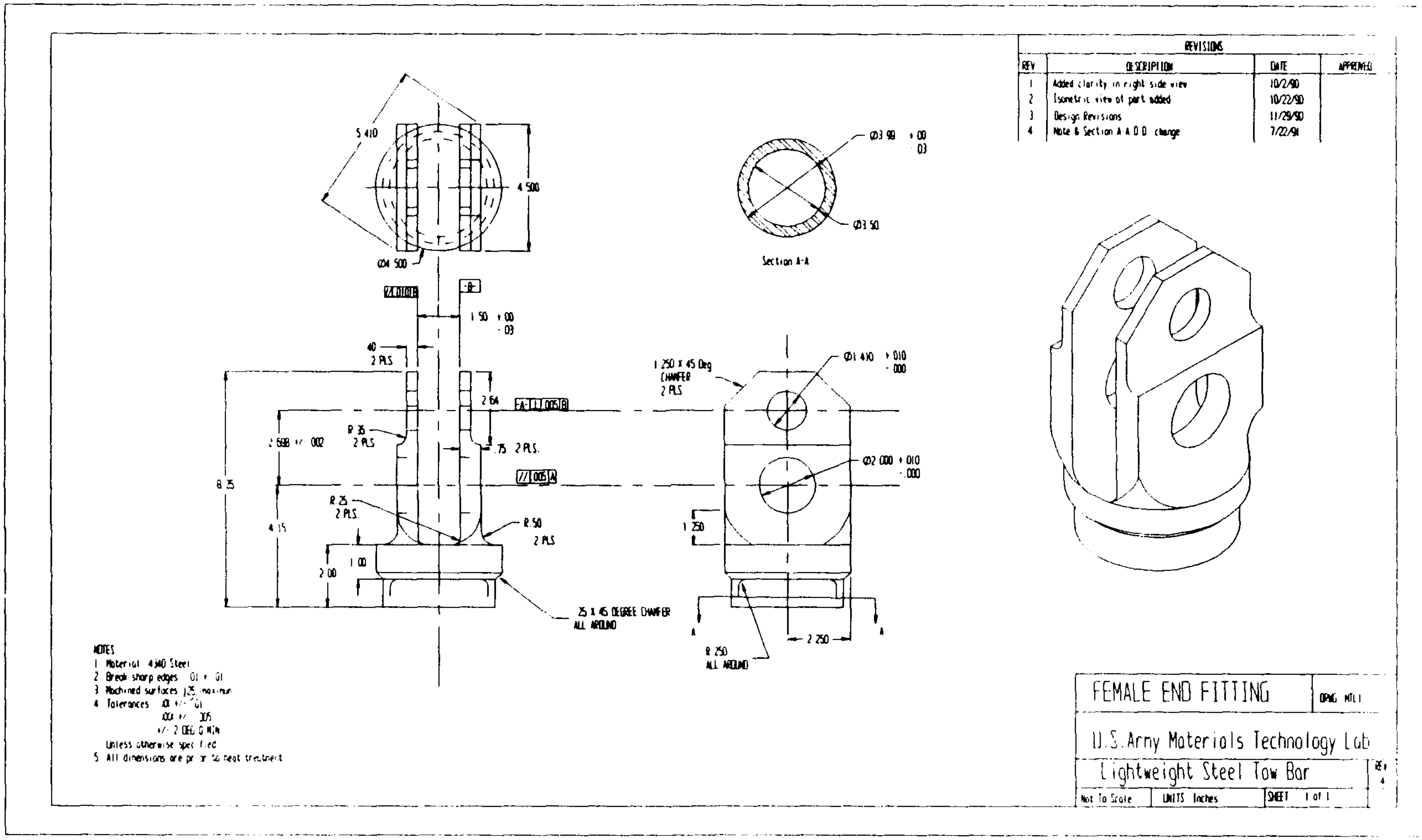
$$\sigma_s = P/A = 97,865/3.078$$

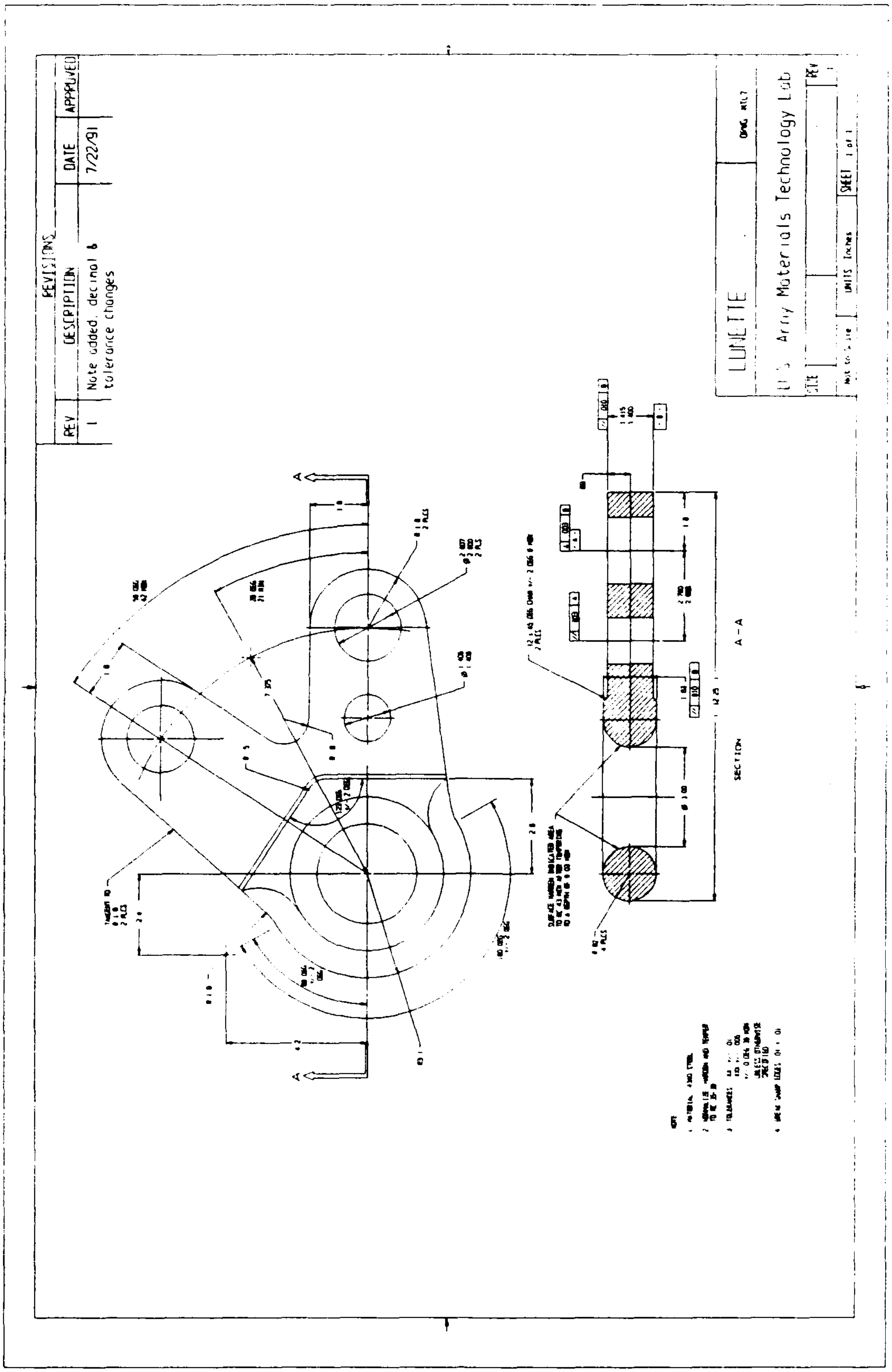
$$\sigma_s = 31,795 \text{ psi}$$

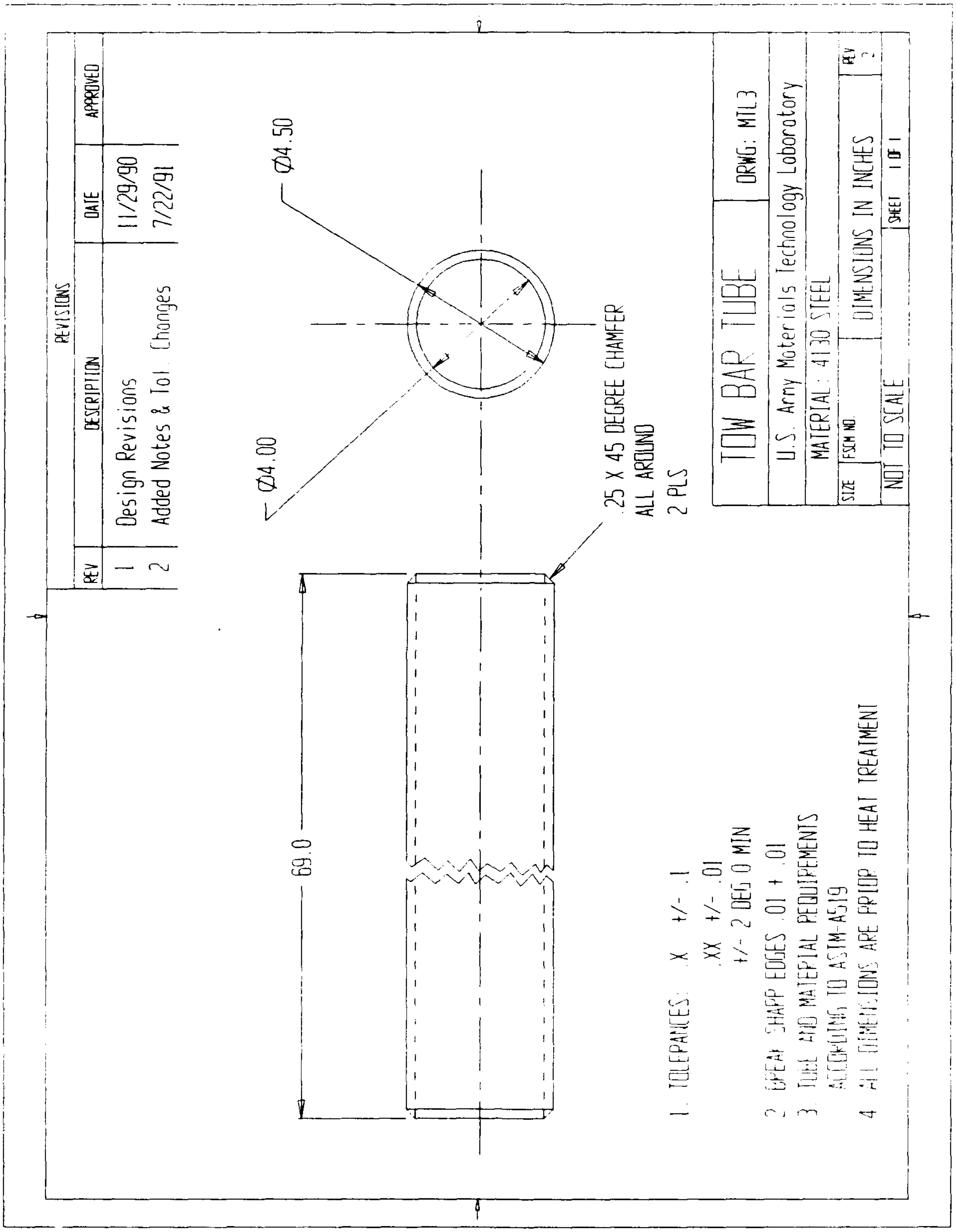
APPENDIX B

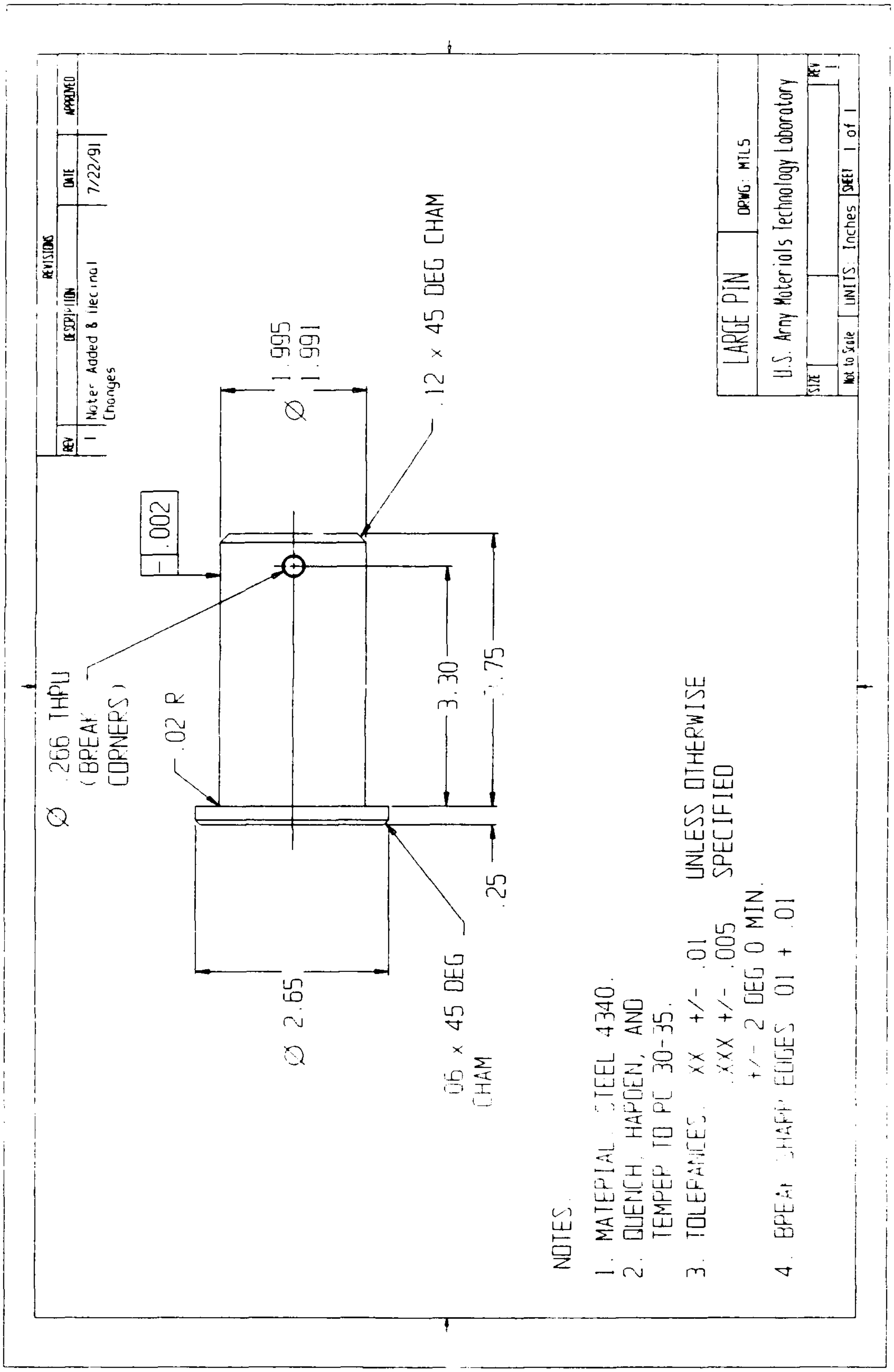
ENGINEERING DRAWINGS

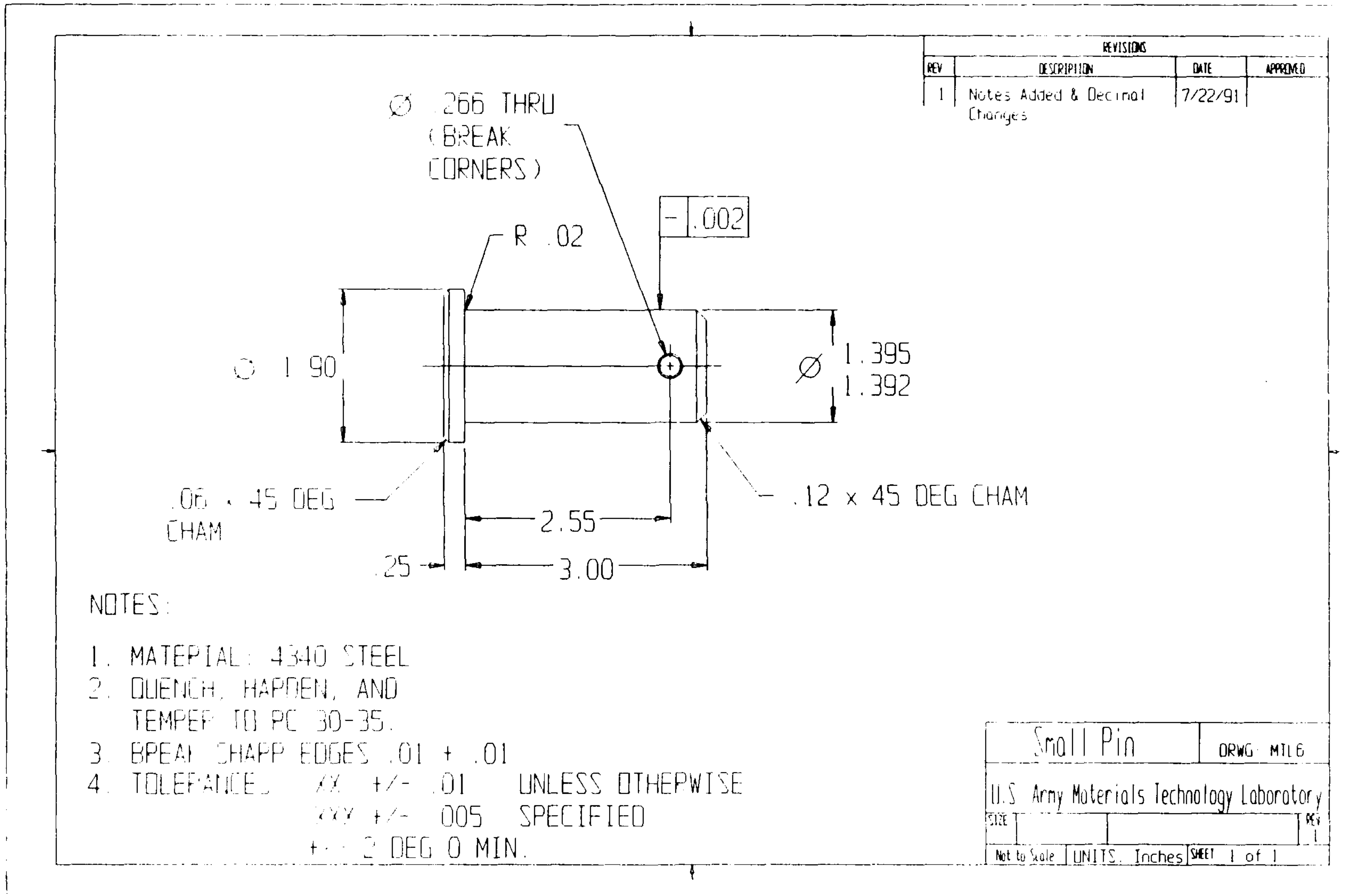

REVISIONS
REV | DESCRIPTION | DATE | APPROVED
1 | Design Revisions | 11/29/90
2 | Notes & Section A-A O.D. changed | 7/22/91
4.50
1.75 +.00 -.03
Ø1.500 +.030 -.000
R1.50
6.13
4.63
2.00
1.00
.25 X 45 DEGREE CHAMFER ALL AROUND
-B-
.005 B
// .010 B
R.50
2 PLS
A
R.25 ALL AROUND
Ø3.99 +.00 -.03
Ø3.50
SECTION A-A
1 Tolerances .XX +/- .01
.XXX +/- .005
+/- 2 DEG 0 MIN
Unless Otherwise Specified
2. All Dimensions Are In Inches
3 Break Sharp Edges .01 + .01
4 Machined Surfaces 125 Maximum
5 All Dimensions Are Prior To Heat Treatment
MALE END FITTING
DRWG: MTL2
U.S. Army Materials Technology Laboratory
SIZE
FSCM NO
Material: 4340 Steel
REV
NOT TO SCALE
SHEET 1 of 1

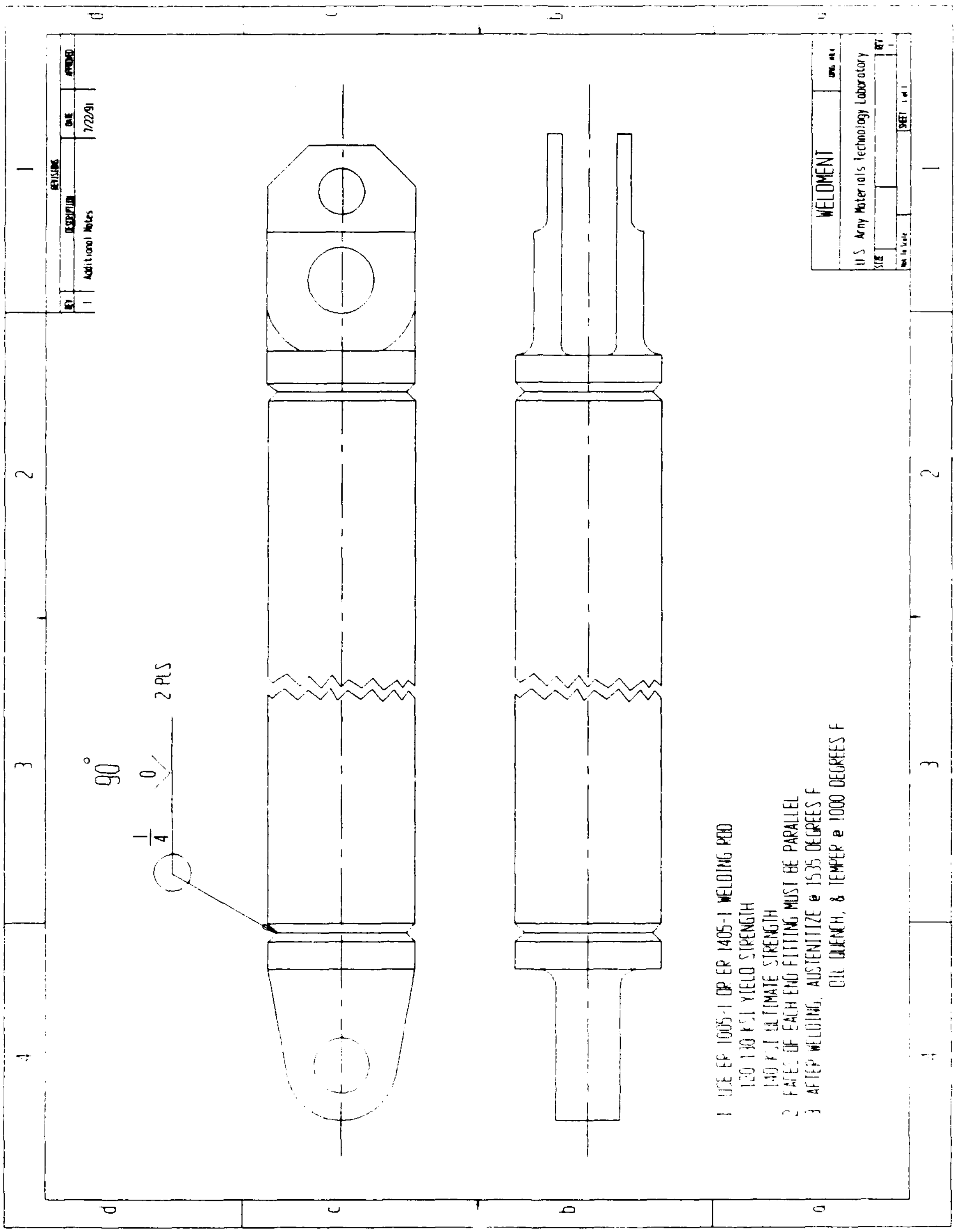

REVISIONS
REV | DESCRIPTION | DATE | APPROVED
1 Added clarity in right side view 10/2/90
2 Isometric view of part added 10/22/90
3 Design Revisions 11/29/90
4 Note & Section A A O D change 7/22/91
Section A-A
NOTES
1 Material 4340 Steel
2 Break sharp edges
3 Machined surfaces
4 Tolerances
Unless otherwise specified
5 All dimensions are prior to heat treatment
25 X 45 DEGREE CHAMFER ALL AROUND
1 250 X 45 Deg CHAMFER 2 PLS
R 250 ALL AROUND
FEMALE END FITTING
U.S. Army Materials Technology Lab
Lightweight Steel Tow Bar
Not To Scale
UNITS Inches
SHEET 1 of 1


REVISIONS
REV
DESCRIPTION
DATE
APPROVED
1
Note added, decimal & tolerance changes
7/22/91
LUNETTE
U.S. Army Materials Technology Lab
SECTION
A - A
UNITS Inches
SHEET 1 of 1
SIZE
REV

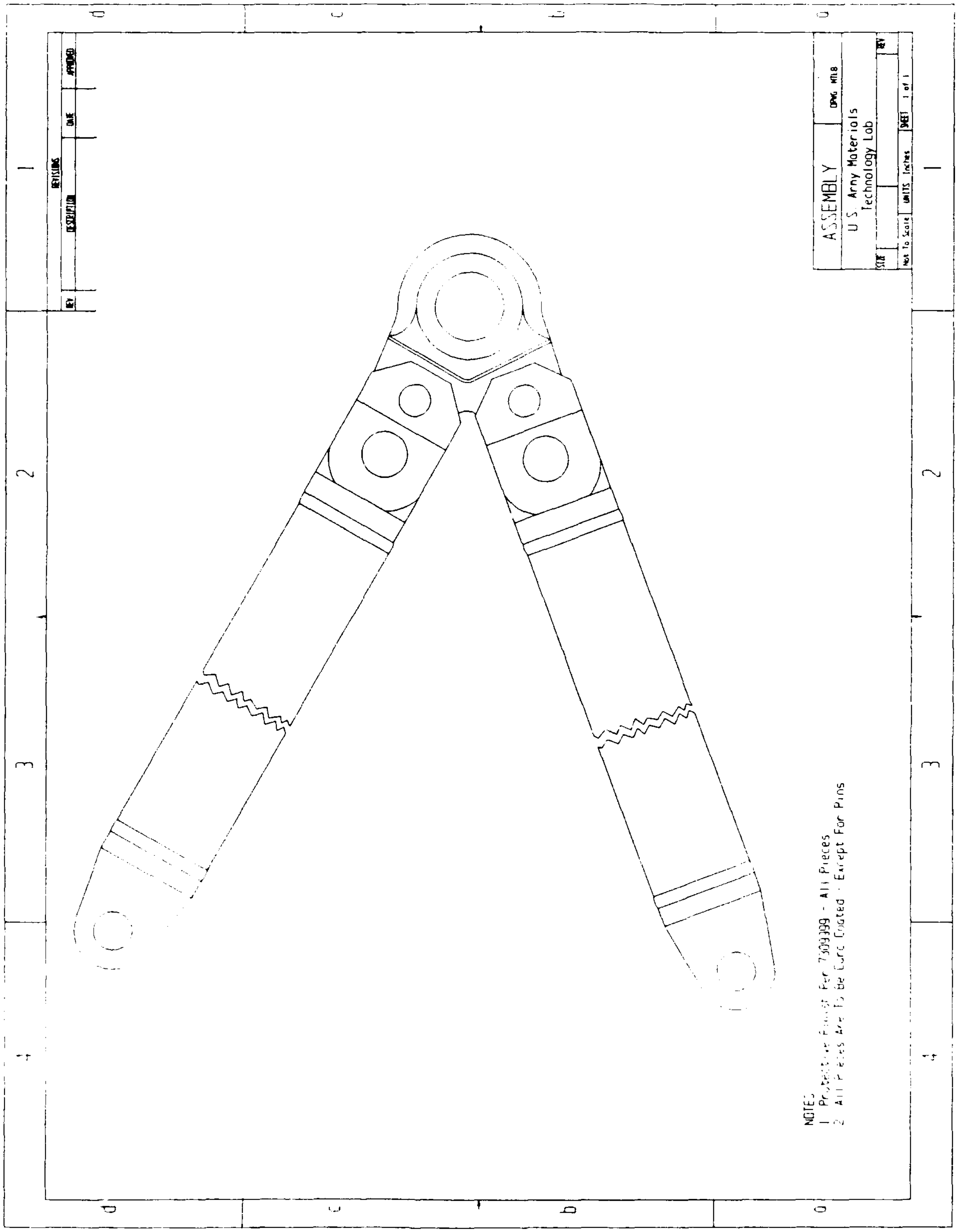

REVISIONS
REV
DESCRIPTION
DATE
APPROVED
1
Design Revisions
11/29/90
2
Added Notes & Tol. Changes
7/22/91
69.0
Ø4.00
Ø4.50
.25 X 45 DEGREE CHAMFER
ALL AROUND
2 PLS
1. TOLERANCES: .X +/- .1
.XX +/- .01
+/- 2 DEG 0 MIN
2. BREAK SHARP EDGES .01 + .01
3. TUBE AND MATERIAL REQUIREMENTS
ACCORDING TO ASTM-A519
4. ALL DIMENSIONS ARE PRIOR TO HEAT TREATMENT
TOW BAR TUBE
DRWG: MTL3
U.S. Army Materials Technology Laboratory
MATERIAL: 4130 STEEL
SIZE
FSCM NO.
DIMENSIONS IN INCHES
REV
2
NOT TO SCALE
SHEET 1 OF 1


REVISIONS
REV
DESCRIPTION
DATE
APPROVED
1
Notes Added & Decimal Changes
7/22/91
.002
Ø 1.995
1.991
.12 x 45 DEG CHAM
Ø .266 THRU
(BREAK CORNERS)
.02 R
3.30
3.75
.25
Ø 2.65
.06 x 45 DEG
CHAM
NOTES:
1. MATERIAL: STEEL 4340.
2. QUENCH, HARDEN, AND
TEMPER TO RC 30-35.
3. TOLERANCES: .XX +/- .01 UNLESS OTHERWISE
.XXX +/- .005 SPECIFIED
+/- 2 DEG 0 MIN.
4. BREAK SHARP EDGES .01 + .01
LARGE PIN
DRWG: MTL5
U.S. Army Materials Technology Laboratory
SIZE
REV
1
Not to Scale
UNITS: Inches
SHEET 1 of 1


REVISIONS
REV | DESCRIPTION | DATE | APPROVED
1 | Notes Added & Decimal Changes | 7/22/91
∅ .266 THRU (BREAK CORNERS)
R .02
- | .002
∅ 1.90
∅ 1.395 1.392
.06 x 45 DEG CHAM
.12 x 45 DEG CHAM
2.55
.25
3.00
NOTES:
1. MATERIAL: 4340 STEEL
2. QUENCH, HARDEN, AND TEMPER TO RC 30-35.
3. BREAK SHARP EDGES .01 + .01
4. TOLERANCE: .XX +/- .01 UNLESS OTHERWISE
.XXX +/- .005 SPECIFIED
+/- 2 DEG 0 MIN.
Small Pin
DRWG: MTL6
U.S. Army Materials Technology Laboratory
SIZE
REV 1
Not to Scale
UNITS: Inches
SHEET 1 of 1



REV
DESCRIPTION
DATE
APPROVED
REVISIONS
1
Additional Notes
7/22/91
1/4
90°
2 PLS
1 USE ER 100S-1 OR ER 140S-1 WELDING ROD
120-130 KSI YIELD STRENGTH
140 KSI ULTIMATE STRENGTH
2 FACES OF EACH END FITTING MUST BE PARALLEL
3 AFTER WELDING, AUSTENITIZE @ 1535 DEGREES F
OIL QUENCH, & TEMPER @ 1000 DEGREES F
WELDMENT
U S Army Materials Technology Laboratory
SIZE
REV
SHEET 1 of 1


REVISIONS
REV
DESCRIPTION
DATE
APPROVED
ASSEMBLY
DWG MTL8
U.S. Army Materials
Technology Lab
SIZE
REV
Not To Scale
UNITS Inches
SHEET 1 of 1
NOTES
1 Per 7309399 - All Pieces
2 All Pieces Are To Be Coated · Except For Pins

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	Director, U.S. Army Materials Technology Laboratory, Watertown, MA 02172-0001
2	ATTN: SLCMT-TML
3	Authors

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Watertown, Massachusetts 02172-0001
DESIGN OF THE M1 LIGHTWEIGHT STEEL TOW BAR -
Christopher Cavallaro, Robert B. Dooley, and
Paul V. Cavallaro

Technical Report MTL TR 92-60, September 1992, 36 pp -
illus-tables, D/A Project 1L263100D71

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Key Words

Tanks
Tow bar
Lightweight steel

The increased weight of today's M1A1 and M1A2 Main Battle Tanks has introduced a tow bar failure problem encountered only during field recovery operations. This problem is one of insufficient strength as the tow bar system currently used in the field was not designed for the recovery of these heavier vehicles. The direct result has been an increasing number of tow bar failures. In a joint program between the U.S. Army Tank-Automotive Command (TACOM), the U.S. Army Materials Technology Laboratory (MTL) and Foster Miller, Inc. (FMI), a new lightweight composite tow bar was developed. During the development of this new system, it became apparent to MTL that the cost of manufacturing and materials for this composite system would be relatively high when compared to the current steel tow bar system. Accordingly, MTL developed a new lightweight tubular steel tow bar at the same time the composite tow bar program was coming to a close. The objective of this report is to address the design of this new steel tow bar system. It was constructed of a combination of 4130 and 4340 alloy steel and possesses several key advantages over the current system. These include a 30% increase in strength, a 23% weight reduction and interchangeable legs. In addition to these advantages, the cost of this new system when in production was estimated to be comparable to that of the current tow bar system.

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