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John D. Landes

Dr. John Landes, Major Professor

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Thomas C Carley

J A

Accepted for the Council:

Lew Minkel

Associate Vice Chancellor and Dean of The Graduate School

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Date April 27, 1994

THE INTEGRATED BELT-IN-SEAT DESIGN BY NATIONAL SEATING, INC.

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Alvaro Eduardo Urrutia

May, 1994

DEDICATION

This thesis is dedicated to my parents

Sr. Luis Juan Urrutia Nicolini

and

Sra. Blanca Rosa Castro de Urrutia

who have provided me with all the ingredients

required to achieve my goals.

ACKNOWLEDGMENTS

I would like to thank National Seating, especially Mr. Larry Blankenship, for supporting this work from beginning to end. I would also like to express my thanks to Mr. Rodney Smith at National Seating for laying out the details of a functional recline mechanism which most likely will become part of the final product. I would also like to thank my major professor, Dr. J.D. Landes, and the other committee members, Dr. T.G. Carley and Dr. J.A.M. Boulet, for their comments. I would like to express my thanks to my friend, Mrs. Helen Roberts, for helping me improve the form in which this document was written. Finally, I would like to thank my wife, Deyna Zaragoza Urrutia, for encouraging me to place this work as a top priority until it was completed.

ABSTRACT

This research attempted to develop a prototype heavy-truck seat with integrated seat belts, requiring no attachment of the seat belt to the vehicle's body or tether belts attached to the floor.

The design process consisted of iterations between theoretical analysis and actual model testing. The Mechanics of Materials method was selected as the analytical tool throughout the design process. In addition, a number of physical models were built and tested at National Seating's facilities in Vonore, Tennessee, in order to supplement the theoretical analysis.

A final prototype capable of sustaining the required loads was built and tested. The test results confirmed that the belt-in-seat concept is definitely feasible from the structural standpoint alone.

The integrated belt-in-seat design concept may be offered as an alternative to the conventional seat design in the heavy-truck industry. Despite the fact that this design concept demands a heavier structure, the weight added is small relative to the overall weight of a truck. Furthermore, this minor drawback may be offset to a large degree by the added safety and comfort.

Before the belt-in-seat design concept is fully implemented, the prototype may need to be optimized in order

to meet National Seating's current tooling. Additional manufacturing capabilities may be required to reduce costs in order to meet the market's needs.

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CHAPTER 1

INTRODUCTION

1.1. PROBLEM DEFINITION

It is standard practice in the automobile industry to attach one end of the upper torso seat belt to the vehicle's body, which ensures that the vehicle's body ultimately absorbs the loads during a crash. Truck seats typically have an additional belt, the tether belt, which secures the seat base to the vehicle's floor. Attaching the upper torso belt to the upper frame of the seat (back rest) instead of to the vehicle's body would provide additional comfort to the seat occupant. However, the seat structure would have to be substantially stronger than it is now in order to absorb the loads during a crash.

This research attempted to develop a heavy-truck seat prototype that would require no attachment of the seat belt to the vehicle's body or tether belts attached to the floor.

1.2. ADVANTAGES OF INTEGRATED SEAT BELTS

An integrated belt-in-seat, tetherless design offers the following advantages over the conventional three-belt anchor point design:

1. *Constant belt-to-body angle in any seat position:*
   The anchor points move with the seat structure when the seat is adjusted forward.

2. *Minimization of forward head excursion:*
   A shorter seat belt is required for this design. Therefore, a reduction of belt stretch during a crash results in less head excursion.

3. *Ease of mounting seat onto body of vehicle:*
   Since tether belts are eliminated, the chances of occupant injury due to inadequate tether belt installation are non existant.

4. *Added quality assurance:*
   Since the liability for proper seat belt installation rests with the seat manufacturer, the quality control of the overall assembly process is improved.

5. *Additional comfort:*
   The seat belt anchor points are mounted on the seat rather than on the vehicle body. Elimination of motion of the occupant relative to the seat belt anchor points minimizes discomfort due to friction of the belt on the occupant's body.

1.3. DESIGN REQUIREMENTS

The seat structure must be designed to meet the requirements of paragraph S5.2 from Section 571.210 of the Code of Federal Regulations. The loading configuration during a static test is depicted on Fig. 1.3.1. These forces are intended to simulate the load that the seat occupant would exert on the seat structure in a crash situation.

F_1, the upper torso force, has a magnitude of 3,000 pounds and must be applied horizontally forward.

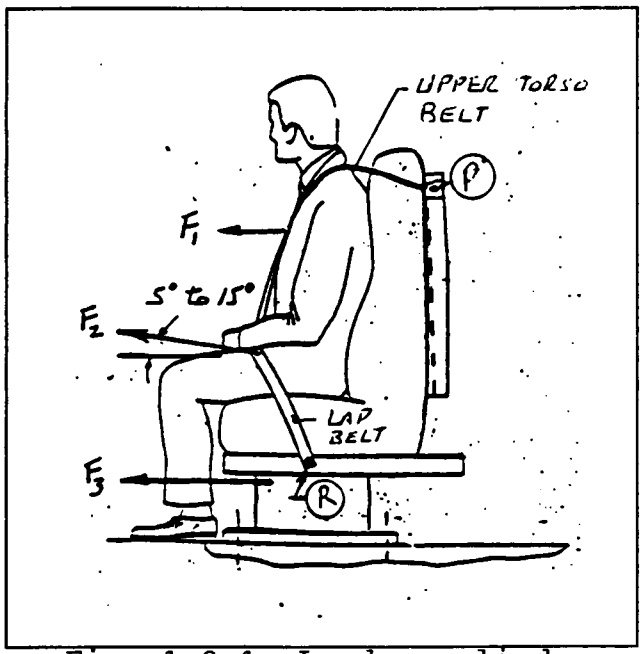


Fig. 1.3.1. Loads applied during a static test.

F_2, the pelvic or lap force, has a magnitude of 3,000 pounds and is applied forward at an angle of not less than 5 degrees or more than 15 degrees above the horizontal (an angle of 10 degrees will be used for design purposes).

F_3 is a horizontally forward force of magnitude twenty times the weight of the seat. This force is applied at the center of gravity of the seat assembly and it simulates the dynamic forces associated with the deceleration of the seat

during a crash.

The upper torso force F_1, the lap force F_2, and the force F_3 applied at the center of gravity must all be applied simultaneously. Fig. 1.3.2. shows the actual test setup used at National Seating's facilities (F_3 was omitted in that particular test).



Fig. 1.3.2. Test setup used at National Seating, Inc.

As required by the Code of Federal Regulations, the external forces must be sustained for at least 10 seconds. In order to be able to sustain the loads, the structure may not collapse. Therefore, the governing design criterion is plastic yield. The design safety factor is determined by the seat manufacturer. The calculations in this document do not include safety factor. Instead, the only consideration in our presentation is that no element in the structure collapses when the required loads are applied. Design safety factors would be considered while converting our prototype into a final design; however, this is beyond the scope of this work.

As far as design features, the back frame must be

capable of reclining at least 12 degrees from the vertical plane. In addition, the complete seat assembly must be able to slide backward and forthward at least 4 inches. The reclinement feature would be accomplished by means of a recline mechanism, whereas the sliding feature would be provided by tracks placed between the seat and the base. The specific details of these mechanisms are beyond the scope of this work.

CHAPTER 2

DESIGN

2.1. DESIGN APPROACH

In order to treat this problem analytically, a number of assumptions and simplifications need to be made. These simplifications introduce an unknown degree of deviation from the actual application.

Numerical methods, such as Finite Element Analysis (FEA), would be appropriate to minimize error due to oversimplification. However, the accuracy of the FEA model would rely on variables such as fitting clearances and manufacturing tolerances. Limit load analysis (collapse load), rather than elastic analysis, would be less sensitive to these manufacturing variables since ordinary construction uncertainties have little effect on plastic collapse load. However, this advantage may be largely offset by the added complexity of the FEA model, since non-linear analysis is required for collapse analysis. In spite of the effort required to develop a non-linear FEA model, the results would still require validation by actual test data.

Rather than developing a purely theoretical model before a prototype was built, a more practical approach was chosen. The design process consisted of a series of iterations between theoretical analysis and actual model

testing. The steps followed throughout the design process are detailed in section 2.3.

2.2. DESIGN CONSIDERATIONS

2.2.1. Free Body Diagram

Fig. 2.2.1. shows all the forces and reactions considered for calculation purposes.

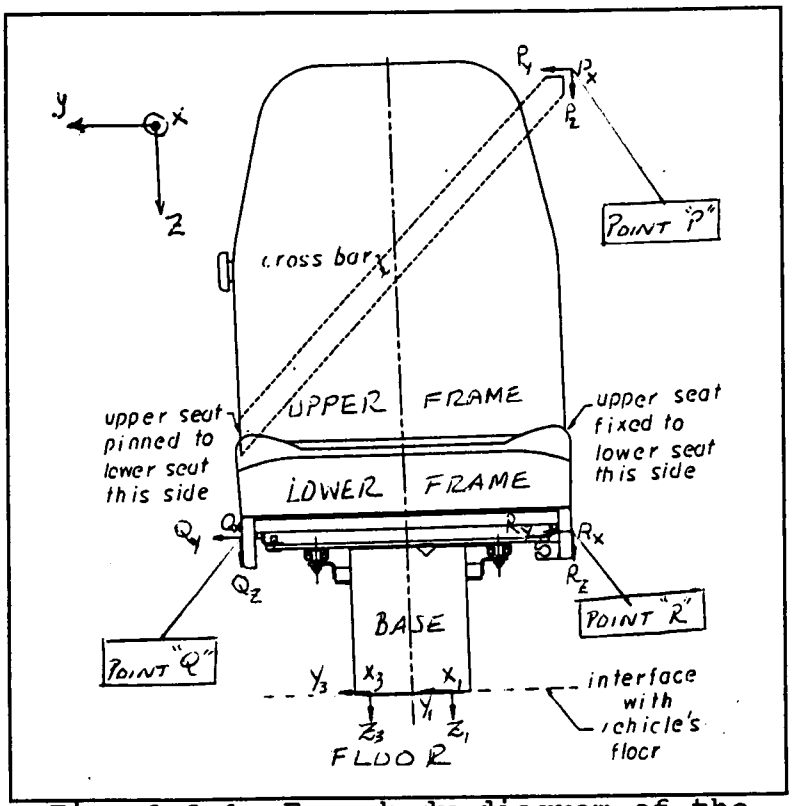


Fig. 2.2.1. Free body diagram of the seat.

F_1 and F_2, the upper torso and lap forces respectively, are transferred through the seat belts onto the seat structure at points P, Q, and R. Therefore, the components of forces P_x, P_y, P_z, Q_x, Q_y, Q_z, R_x, R_y, and R_z, in lieu of the applied forces F_1 and F_2, may be used to design the seat upper frame.

F_3, the force applied at the center of gravity, has no effect on the design of the upper frame or the lower frame since the center of gravity falls below points Q and R. It

only affects the seat base which is beyond the scope of this work. However, F_3 does affect the floor reactions which must be calculated herein. All forces and reactions shown in Fig. 2.2.1. have been defined in the positive sense of the chosen coordinate system.

2.2.2. Interaction Between the Upper Torso and Lap Belt Segments

Fig. 1.3.1. shows how the belt wraps around the occupant's body and how it attaches to the seat. Fig. 2.2.1. labels the three belt-to-seat attachment points as P, Q, and R.

The upper torso belt (segment connecting points P and Q) loops around a bracket at point Q to become the lap belt (segment connecting points Q and R). F_1 and F_2, the upper torso and lap forces, respectively, are applied to body blocks which are the size of a human body. The upper torso force F_1, produces a tension, T_1, on the upper torso belt. Similarly, the lap force F_2, produces a tension, T_2, on the lap belt.

Because the upper torso belt and the lap belt are made from one piece, part of the tension in the upper torso is transferred to the lap belt and vice versa. The amount of tension transferred from one belt segment to the other depends upon the friction at the bracket (point Q). The lower the friction, the greater the tension transferred between

belt segments. Thus, a frictionless loop would allow all of the lap belt tension produced by F_2 to be transferred to the upper torso belt, and all of the upper torso belt tension produced by F_1 to be transferred to the lap belt. The total tension for either belt segment would be the sum $T_1 + T_2$. Conversely, if the bracket allowed no slippage at Q, the only tension on the upper torso belt would be that produced by F_1, and the only force exerting tension on the lap belt would be F_2. The upper torso and lap belt tensions for the slippage-free bracket would be T_1 and T_2, respectively.

The belt tension in an actual application falls somewhere between the case of a frictionless bracket and the case of a slippage-free bracket. Rather than attempting to evaluate the friction at the loop analytically, it was chosen to conduct a test to accurately define the interaction between belt segments. The test consisted of measuring the component of force P_x while applying F_1 only, F_2 only, and F_1 and F_2 together. By doing so, it was possible to determine that F_2, the external force applied to the lap belt, had little effect on P_x (for more details, refer to Test Report No.7 in Appendix A). Based on the test data, P_x was approximately half the force F_1.

2.2.3. Evaluation of Support Reactions

The seat support reactions must be provided to the vehicle's manufacturer, since the floor must be designed to

hold the loaded seat in place. The loads applied to the seat are transferred to the vehicle's body at the interface between the seat base and the floor. For practical purposes, it is conservatively assumed that the floor-mounting bolts are the only points of contact between the seat base and the vehicle's floor. A minimum of four bolts is required to secure the seat to the floor and it will be the decision of the vehicle's manufacturer to specify additional bolts, if necessary.

Because the upper frame of the seat is fixed to the lower frame at one side and pinned at the other side, the structure is asymmetric. Because of the lack of symmetry, no two mounting bolts are expected to transfer equal loads to the floor. Rather than performing a cumbersome analysis to determine the load transferred by each bolt, we present design calculations so that the number of bolts is not a factor. Furthermore, given the critical roll that the floor-mounting bolts play in a crash, it would not be unreasonable to select a substantial design safety factor.

2.3. DESIGN PROCESS

The design process consisted of iterations between theoretical analysis and testing of physical models, which were built to provide design data. This section describes the specific steps followed throughout the design process, which led to the final prototype depicted in section 2.4.

A conventional seat was used as a starting point. It was first modified by welding a bar diagonally to the upper frame as shown in Fig. 2.2.1. The modified conventional seat was then tested with all seat belts anchored to the seat. This was done to observe the behavior of a conventional seat when exposed to the same loading conditions as those of an all-belts-to-seat design. During the test, the forces were progressively increased until the seat collapsed at its weakest point. By carefully analyzing the mode of failure at the weakest point, it was possible to identify which structural properties needed to be increased at that particular location. The pertinent structural properties at the weakest point were then increased to withstand the required loads. The strengthening criteria consisted of increasing the pertinent structural properties by the ratio of required load to load at which the structure collapsed. This strengthening criteria is just an approximation. The stress at any point in the structure increases linearly with the applied forces only until yield occurs at the weakest point. Once yield is reached at the weakest point, the stress no longer increases linearly with the external forces.

A second test was conducted at higher loads until the second weakest point collapsed. The strengthening process described above was repeated for the second weakest point, and so on. The first five failures are summarized in table 2.1.

TABLE 2.1.

- MODE OF FAILURE OF FIRST FIVE STATIC TESTS -	
Test	*Description of Mode of Failure*
1	*The inboard side panel began to deform when each of the applied forces, F_1 and F_2, equalled 600 lbs.*
2	*The rear bar of bearing support frame collapsed when each of the applied forces, F_1 and F_2, equalled 650 lbs.*
3	*The outboard side of the upper frame began to deform at the cross section located 18" below point P when each of the applied forces, F_1 and F_2, equalled 650 lbs.*
4	*The front bar of bearing support frame collapsed when each of the applied forces, F_1 and F_2, equalled 900 lbs.*
5	*The outboard side of the upper frame began to bend forward at the cross section located 13" below point P when each of the applied forces, F_1 and F_2, equalled 1500 lbs.*

After five consecutive iterations it was possible to predict, with reasonable approximation, the strength requirements of the upper frame. By analyzing the mode of failure of the upper frame during static test no. 5, it was possible to estimate the component of force P_x transferred by the upper torso belt to the upper frame. This estimated value was used to develop an upper frame that was capable of withstanding the required forces. The upper frame was validated during static test no. 6, since it sustained the required loads without collapsing.

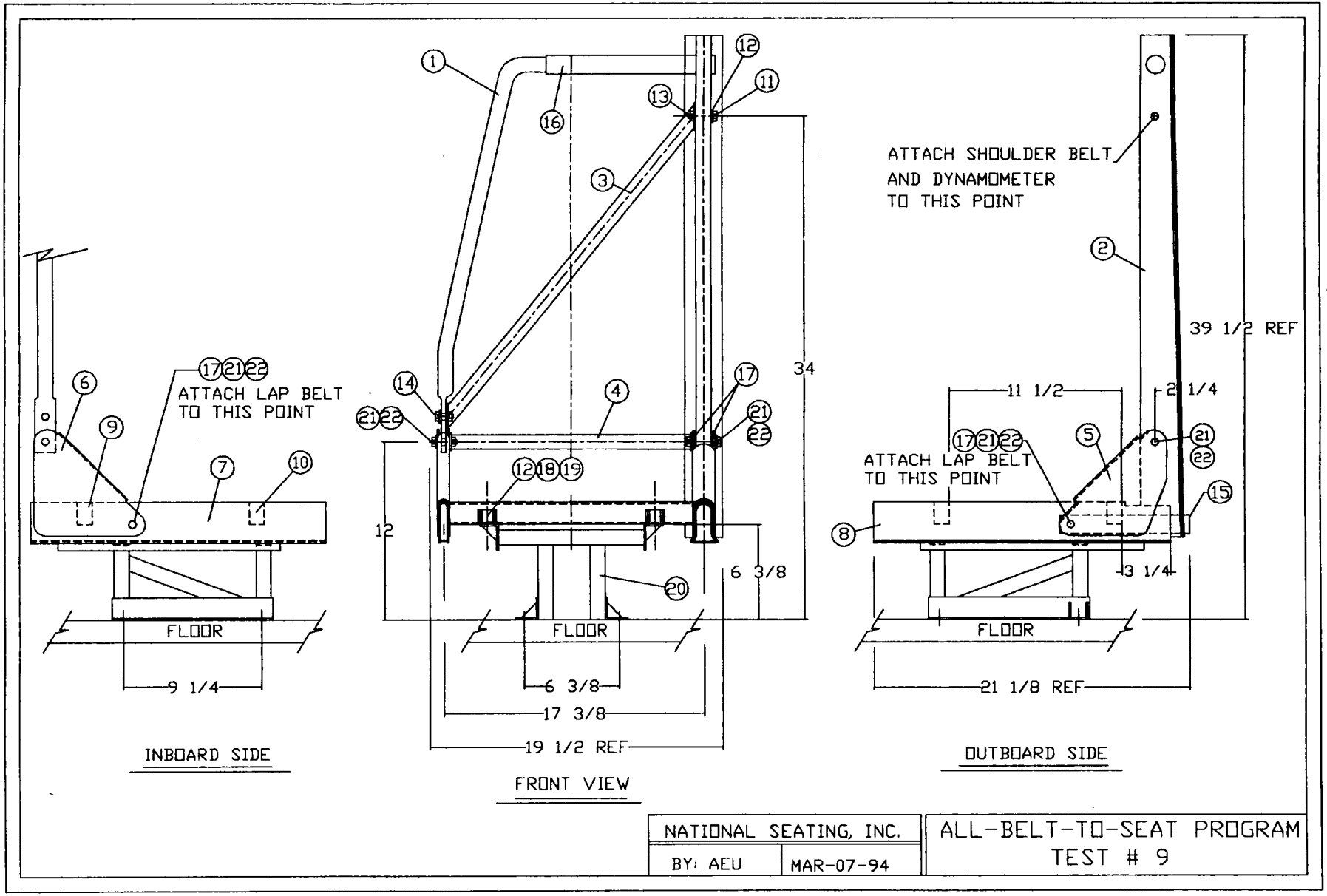
Once an upper frame that could sustain the required loads was available, it was possible to measure the forces transferred by the upper torso belt onto the upper frame while the required external forces, F_1 and F_2, were being applied. A dynamometer was used to record the component of force P_x during test no. 7. The data obtained during test no. 7 also defined the interaction between belt segments as it is discussed in section 2.2.2.

The components of force P_y and P_z were evaluated analytically. The forces at points Q and R may also be obtained from analysis. This task was omitted since points Q and R may be placed at the seat's base; the design of the base is beyond the scope of this work. The data obtained from test number 7 in conjunction with the results obtained analytically constituted the basis for the design calculations which led to the final prototype. Test no. 9 was conducted to validate the final prototype.

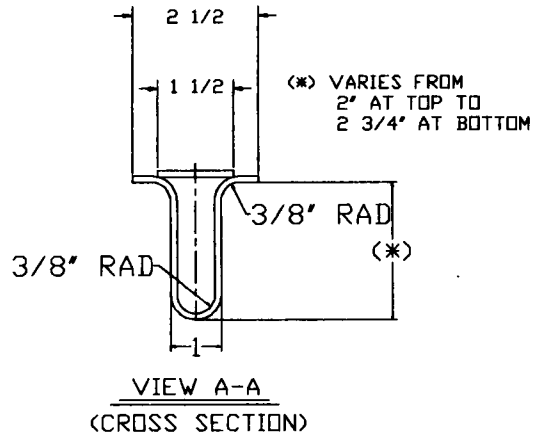
2.4. FINAL DRAWINGS

BILL OF MATERIALS

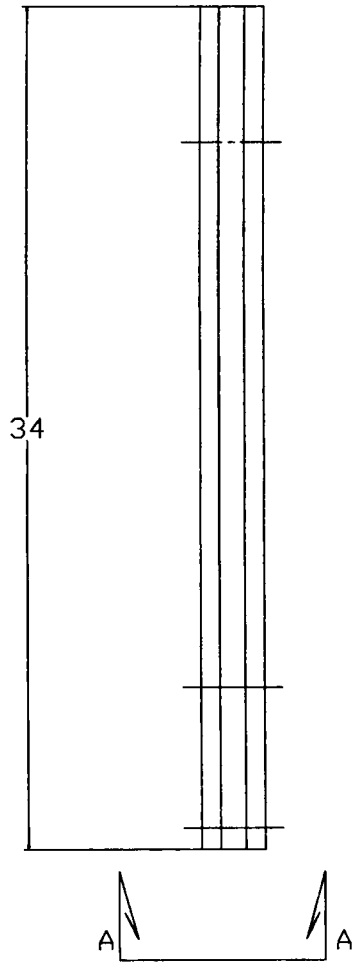
ITEM	QTY	DESCRIPTION	MATERIAL
1	1	FRAME: 0.083"T	C.STL.
2	1	OUTBOARD BEAM: 0.120" T	ASTM A606
3	1	DIAGONAL BRACE: 1"OD x 0.083"T	ASTM A513
4	1	HORIZONTAL BRACE: 1"OD x 0.083"T	ASTM 513
5	1	OUTBOARD GUSSET: 0.120" T	ASTM A606
6	1	INBOARD GUSSET: 0.120"T	ASTM A606
7	1	INBOARD SIDE PANEL: .075"T	C.STL.
8	1	OUTBOARD SIDE PANEL: 0.120"T	ASTM A606
9	1	REAR CROSS BAR: 1 1/2" x 1" x 0.120" T	ASTM A513 (40 KSI MIN.YIELD)
10	1	FRONT CROSS BAR: 1 1/2" x 1" x 0.120" T	ASTM A513 (40 KSI MIN.YIELD)
11	1	BOLT: 5/16" DIA	SAE J429 GR.5
12	10	WASHER: 5/16" NOM. ID	C.STL.
13	2	NUT: 5/16" HEX	C.STL.
14	1	BOLT: 5/16" DIA	SAE J429 GR.5
15	1	RECLINER BAR: 1 1/4"WD x 1/2"T	ASTM A36
16	1	ADAPTER: 1 1/4" OD x 0.120" T	C.STL.
17	8	WASHER: 1/2"NOM.ID x 1/8"T	C.STL.
18	8	BRACKET: 1/8"T	C.STL.
19	8	BOLT: 5/16"DIA	SAE J429 GR.5
20	1	BASE ASSEMBLY	C.STL.
21	4	BOLT: 1/2"DIA	SAE J429 GR.5
22	4	NUT: 1/2"DIA	C.STL.

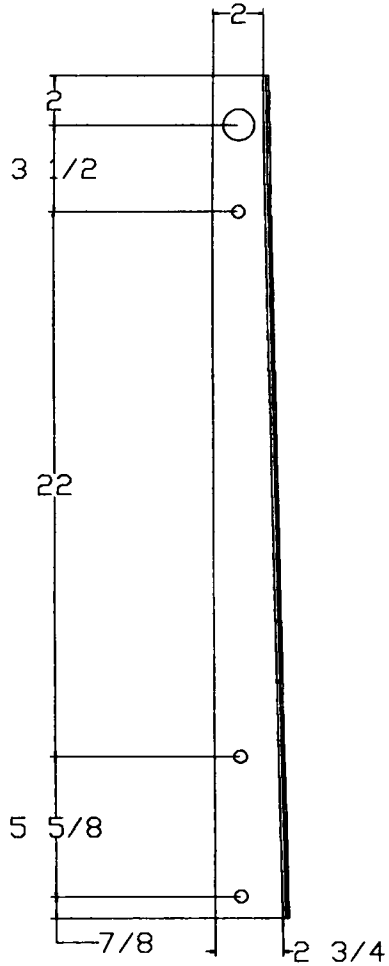


NATIONAL SEATING, INC.		ALL-BELT-TO-SEAT PROGRAM TEST # 9
BY: AEU	MAR-07-94	



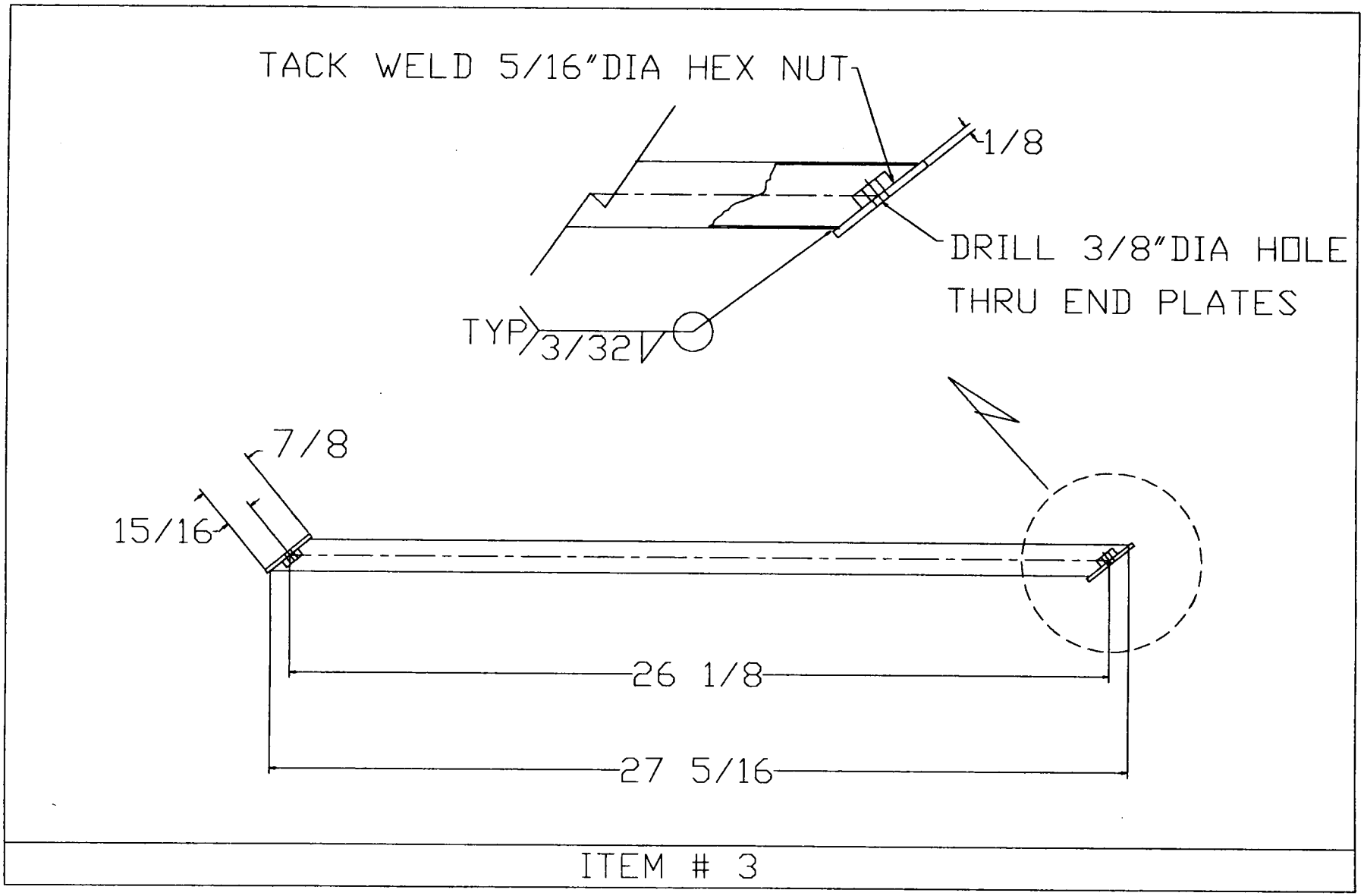
VIEW A-A
(CROSS SECTION)



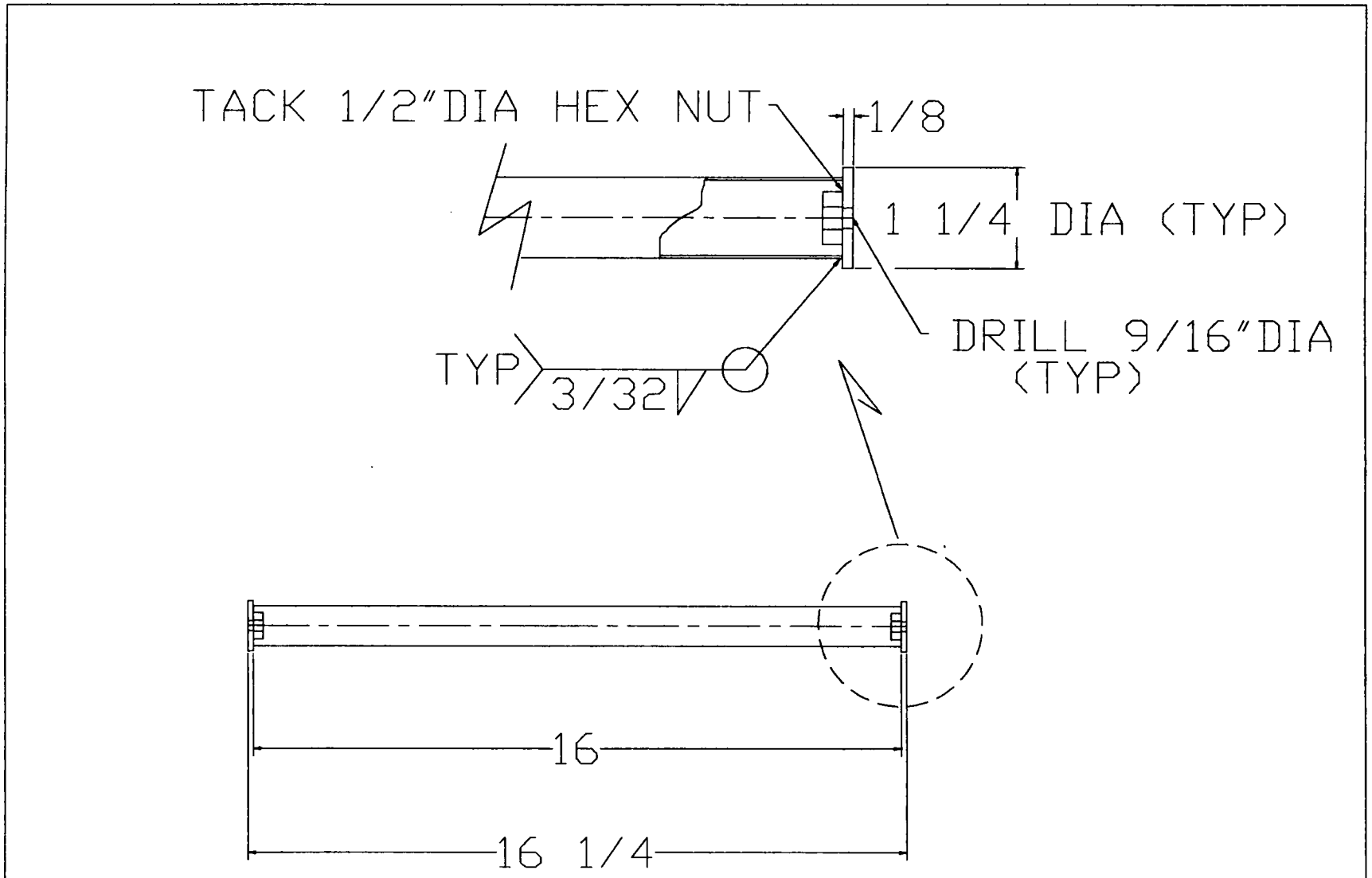


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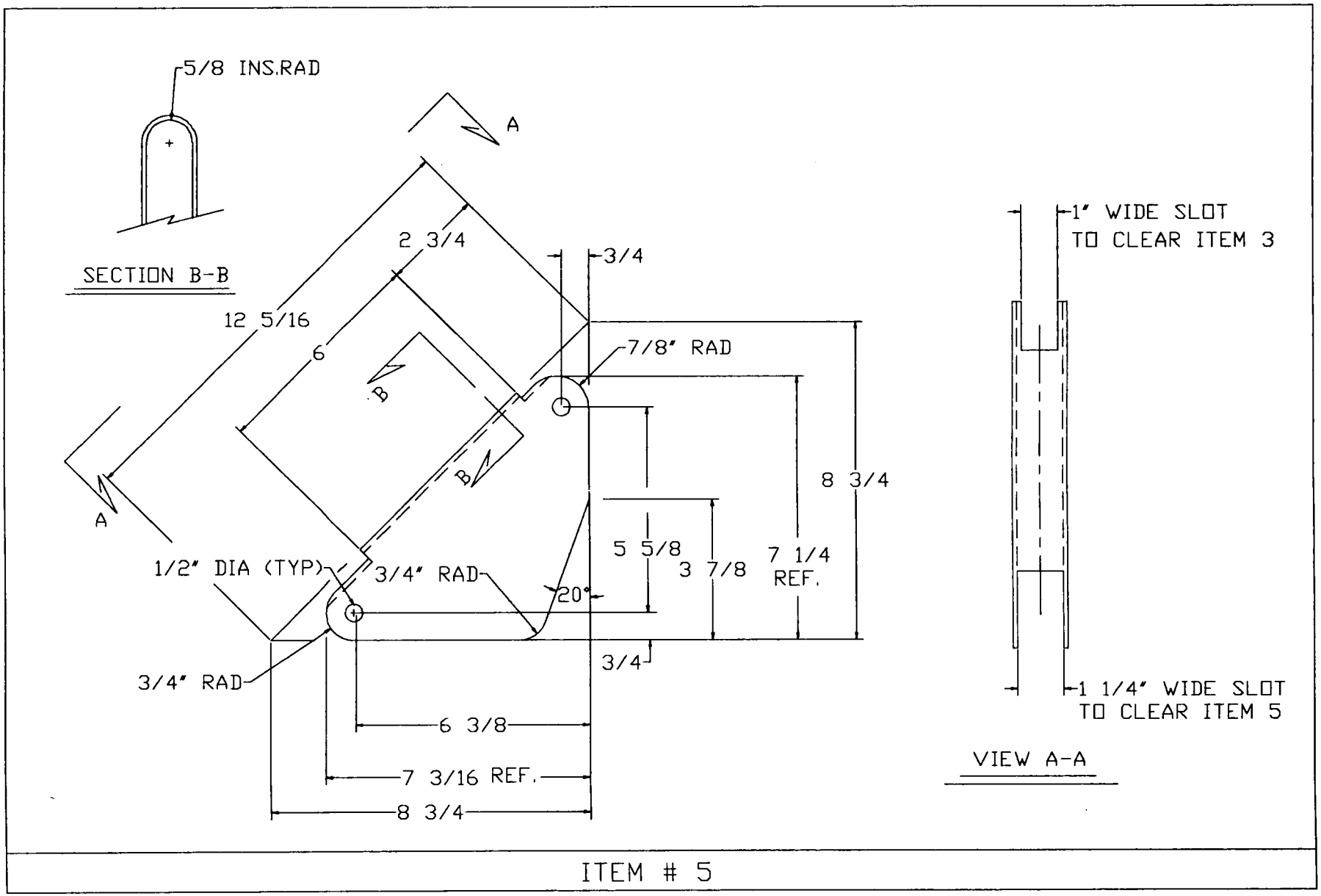
16


TACK WELD 5/16"DIA HEX NUT
1/8
DRILL 3/8"DIA HOLE
THRU END PLATES
TYP
3/32
7/8
15/16
26 1/8
27 5/16

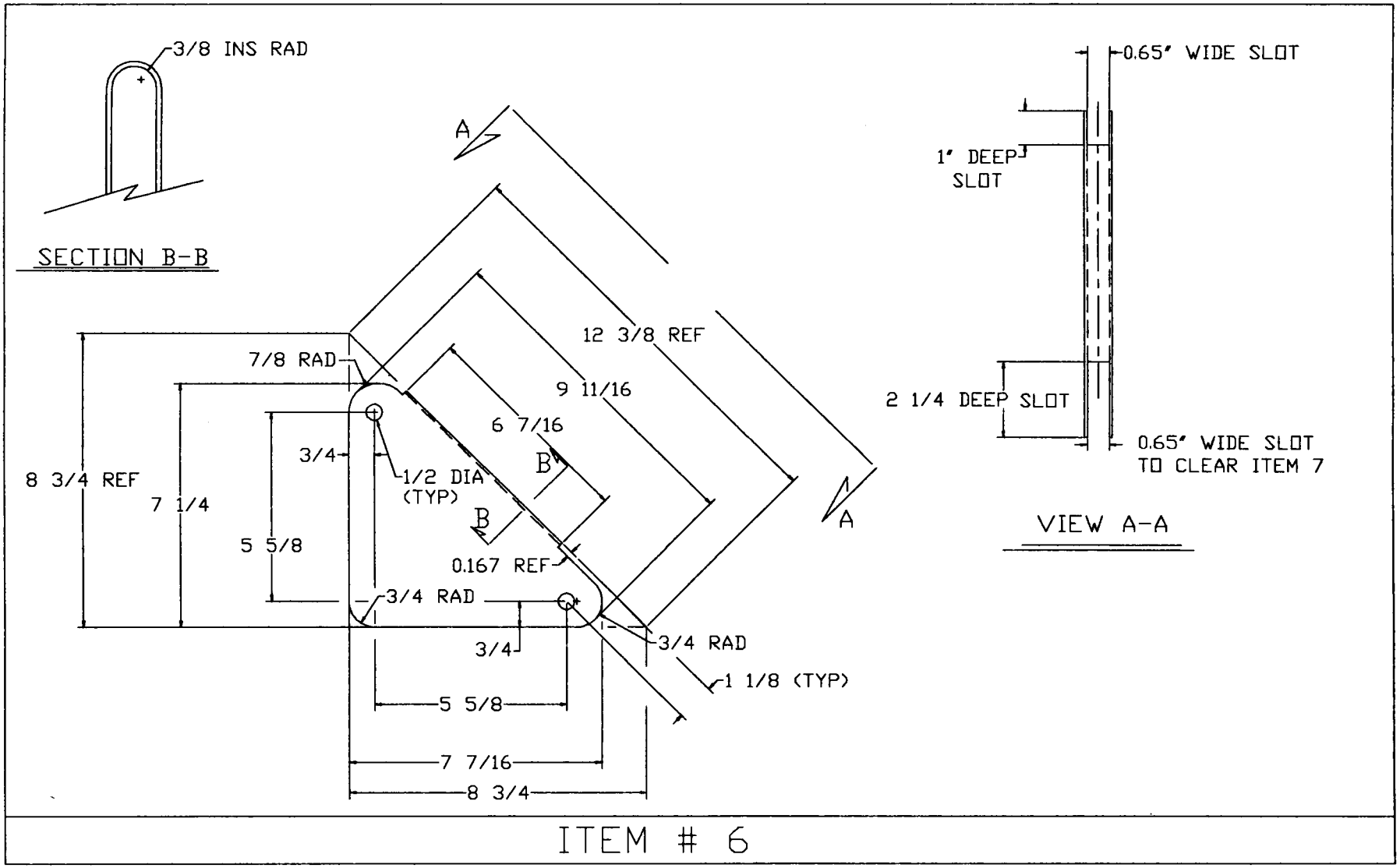
ITEM # 3


TACK 1/2"DIA HEX NUT
1/8
1 1/4 DIA (TYP)
DRILL 9/16"DIA
(TYP)
TYP
3/32
16
16 1/4

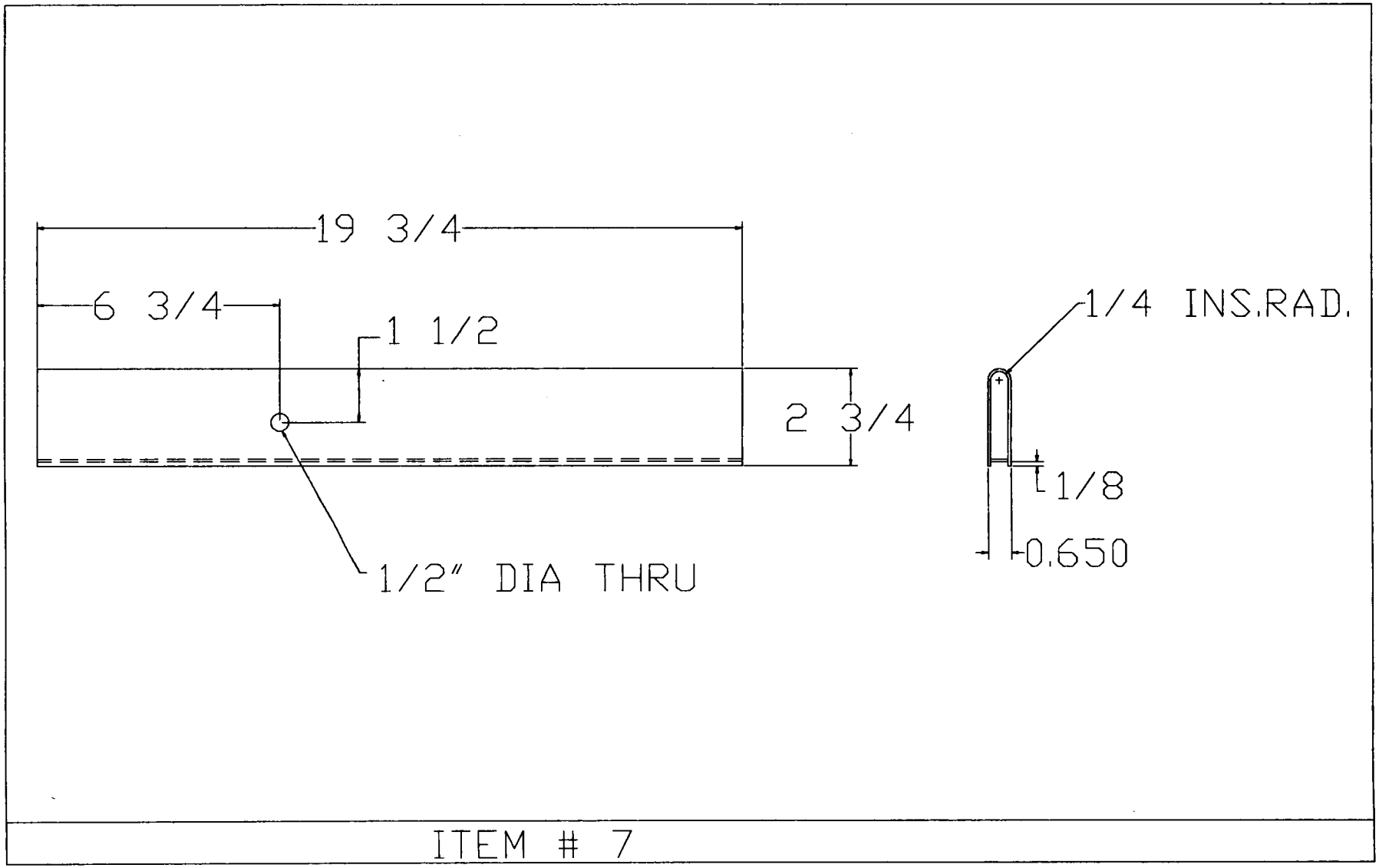
ITEM # 4


5/8 INS.RAD
SECTION B-B
A
B
12 5/16
6
2 3/4
3/4
7/8" RAD
8 3/4
5 5/8
3 7/8
7 1/4
REF.
1/2" DIA (TYP)
3/4" RAD
20°
3/4" RAD
3/4
6 3/8
7 3/16 REF.
8 3/4
1" WIDE SLOT
TO CLEAR ITEM 3
1 1/4" WIDE SLOT
TO CLEAR ITEM 5
VIEW A-A

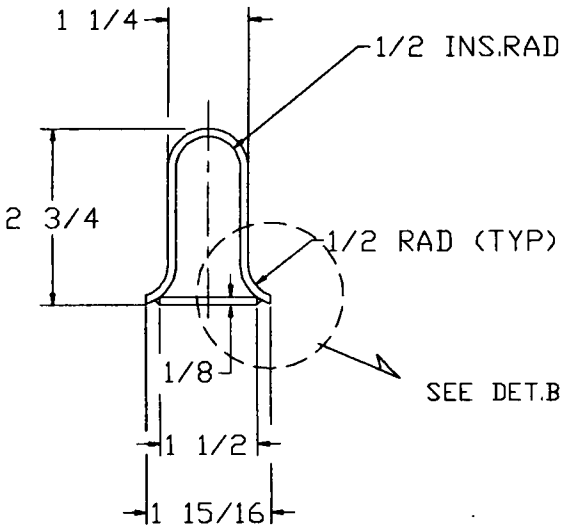
ITEM # 5


3/8 INS RAD
SECTION B-B
A
A
12 3/8 REF
9 11/16
6 7/16
7/8 RAD
3/4
1/2 DIA
(TYP)
B
B
8 3/4 REF
7 1/4
5 5/8
0.167 REF
3/4 RAD
3/4
3/4 RAD
1 1/8 (TYP)
5 5/8
7 7/16
8 3/4
0.65" WIDE SLOT
1" DEEP
SLOT
2 1/4 DEEP SLOT
0.65" WIDE SLOT
TO CLEAR ITEM 7
VIEW A-A

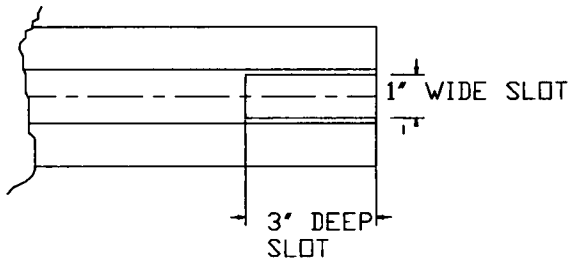
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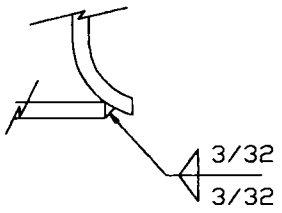
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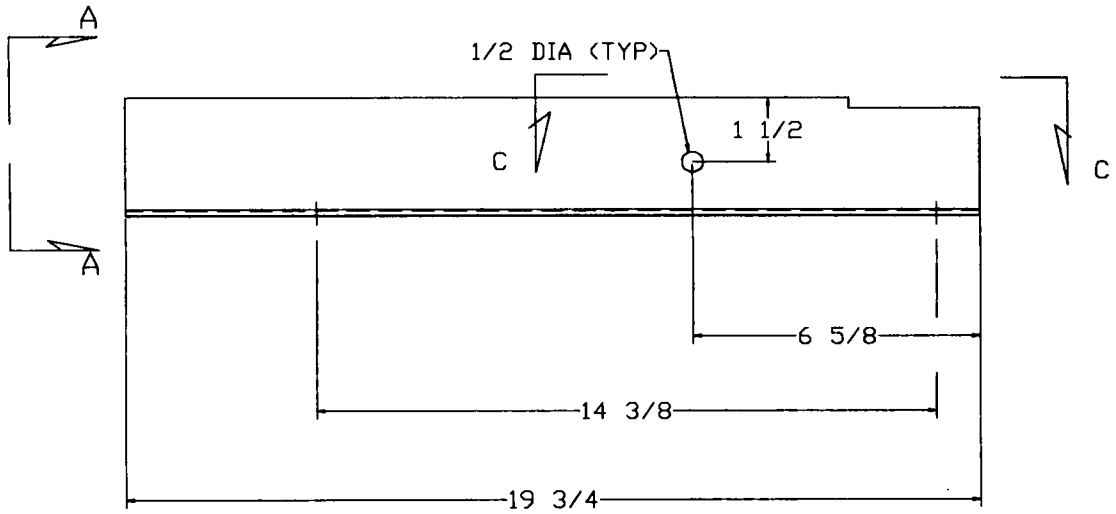
VIEW A-A



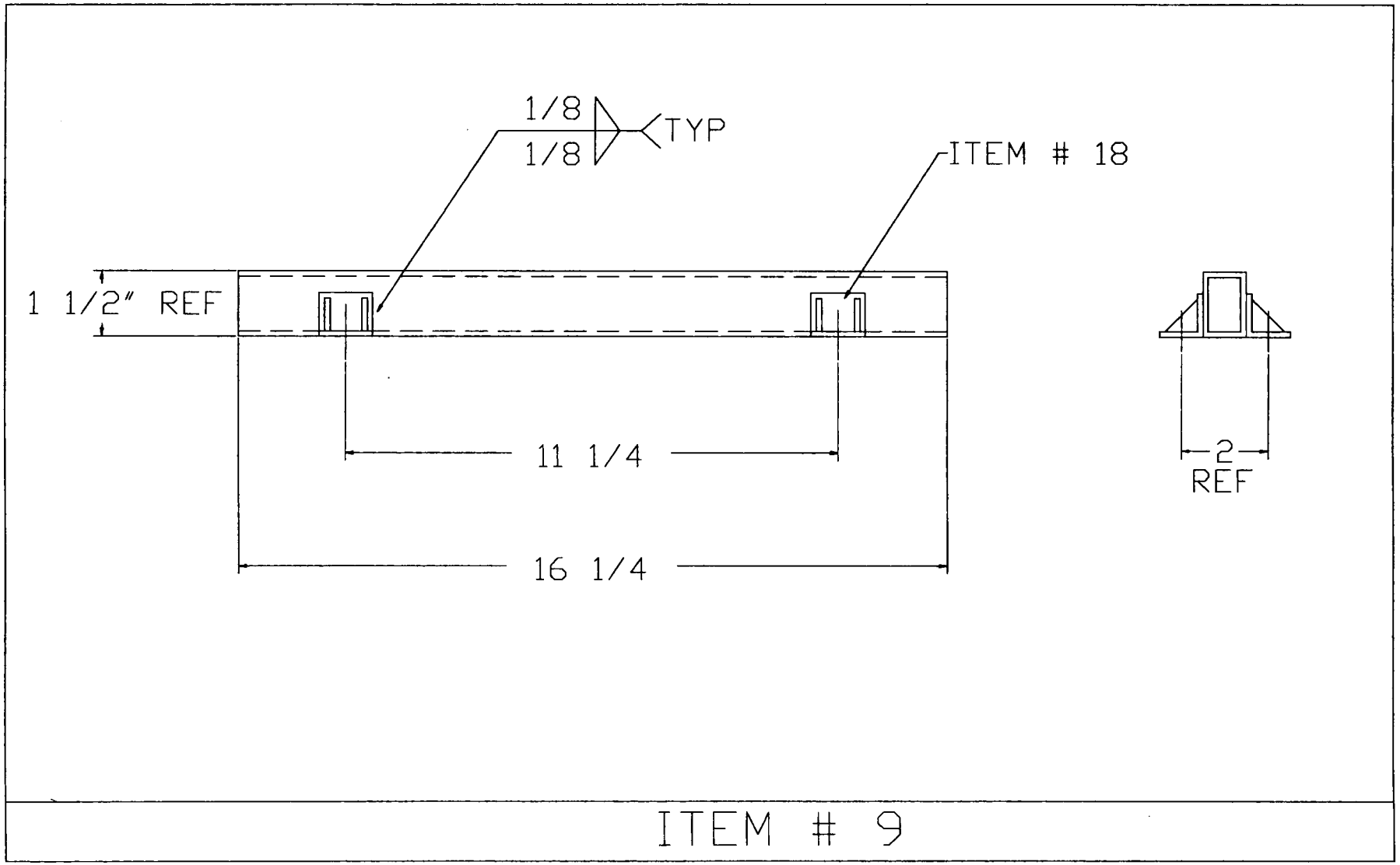
VIEW C-C



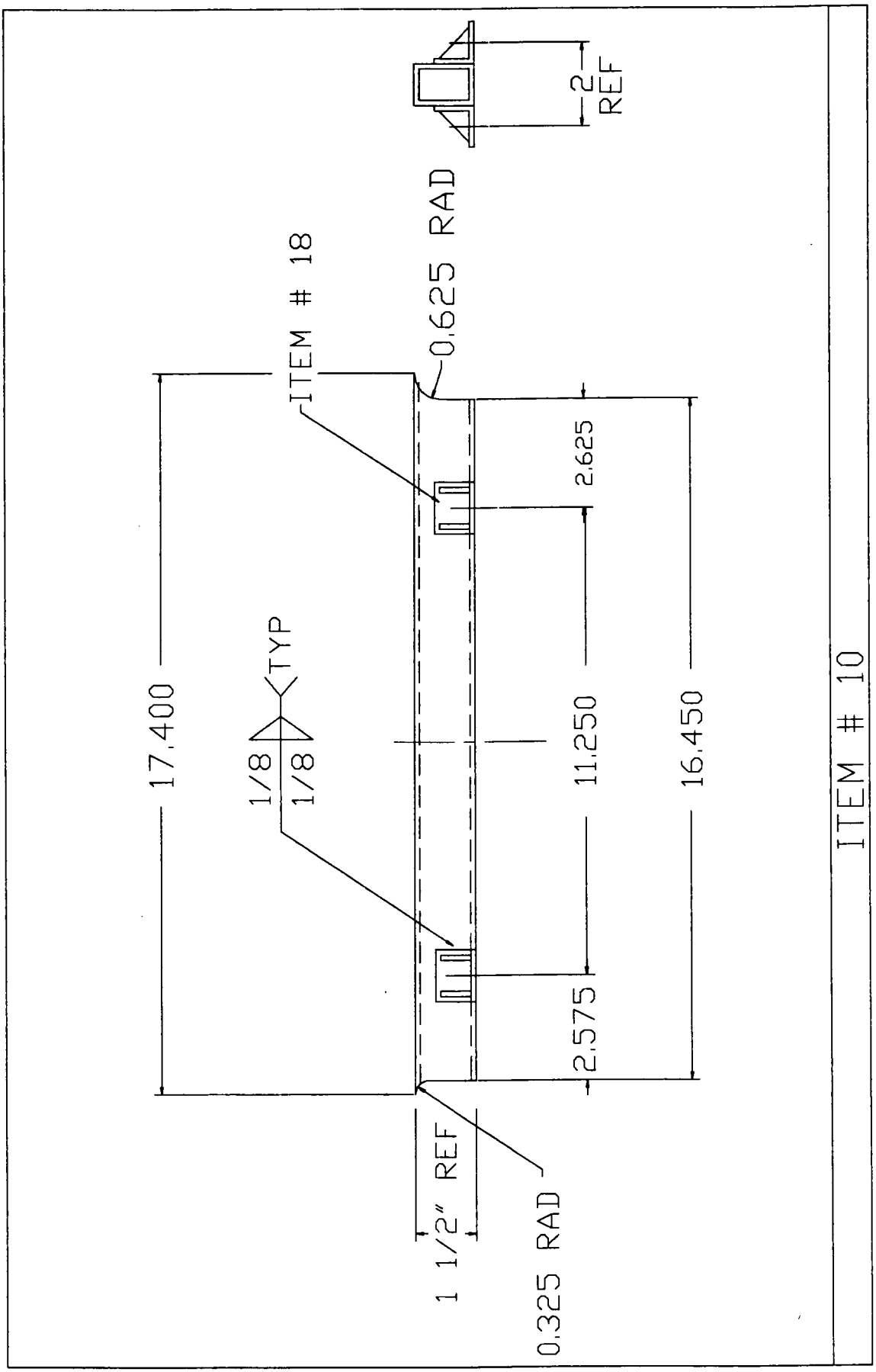
DETAIL B

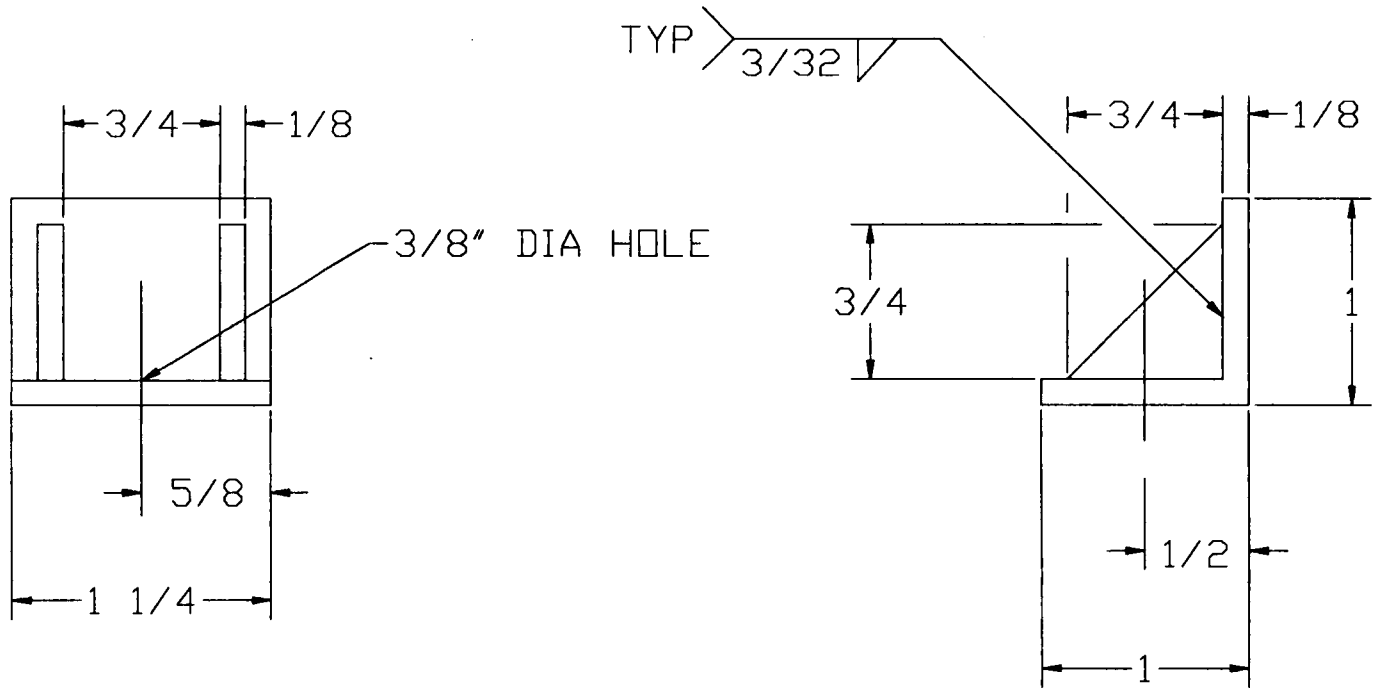


ITEM # 8

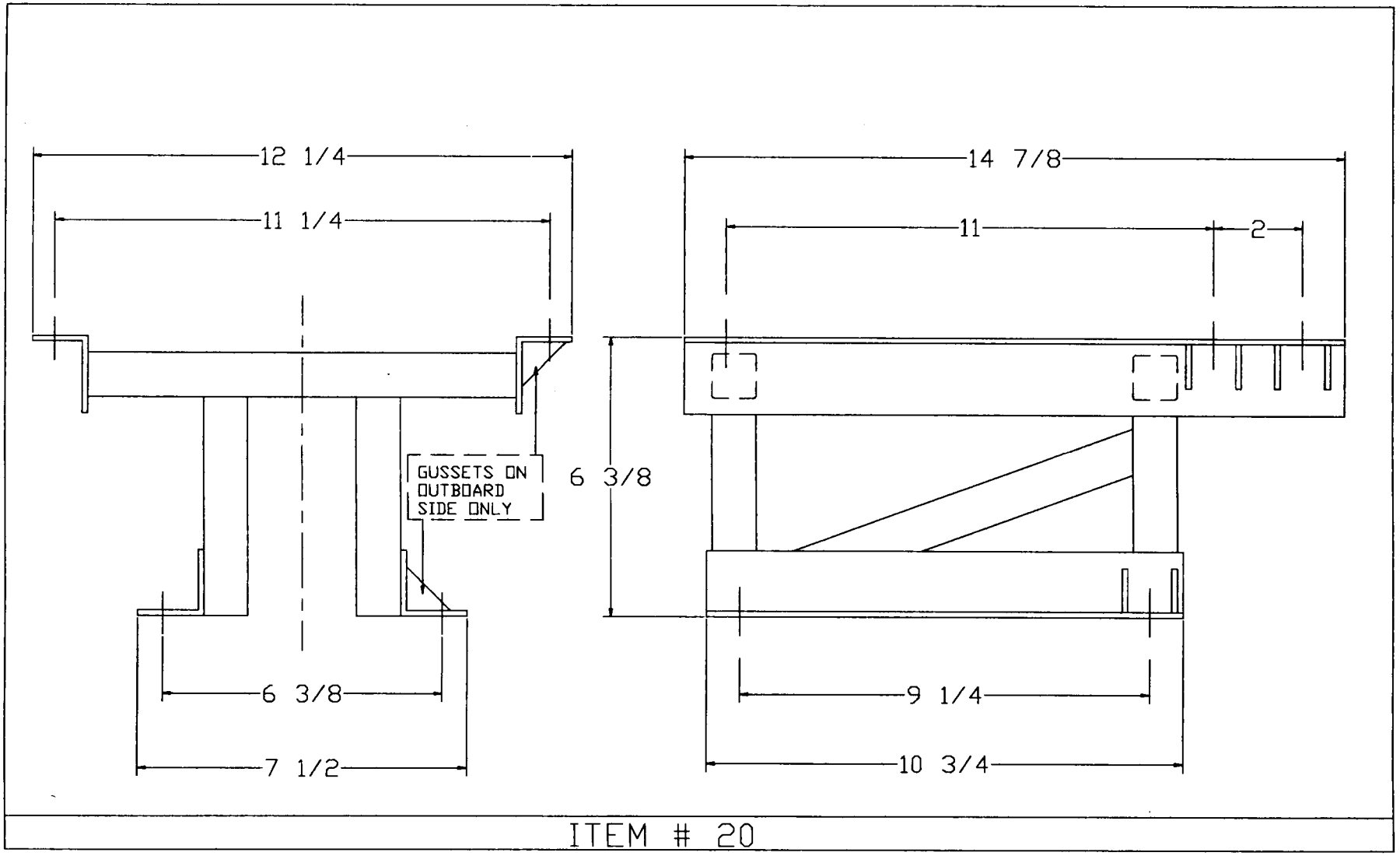


ITEM # 9


2
REF
0.625 RAD
ITEM # 18
2.625
17.400
1/8
1/8
TYP
11.250
16.450
2.575
1 1/2" REF
0.325 RAD
ITEM # 10



ITEM # 18


12 1/4
11 1/4
GUSSETS ON
OUTBOARD
SIDE ONLY
6 3/8
7 1/2
14 7/8
11
2
6 3/8
9 1/4
10 3/4

ITEM # 20

2.5. DESIGN CALCULATIONS

2.5.1. Upper Frame

2.5.1.1. COMPONENTS OF FORCE AT POINT P

Before the upper frame structure is analyzed, the components of force Px, Py, and Pz, as defined in Fig. 2-5-1, are evaluated.

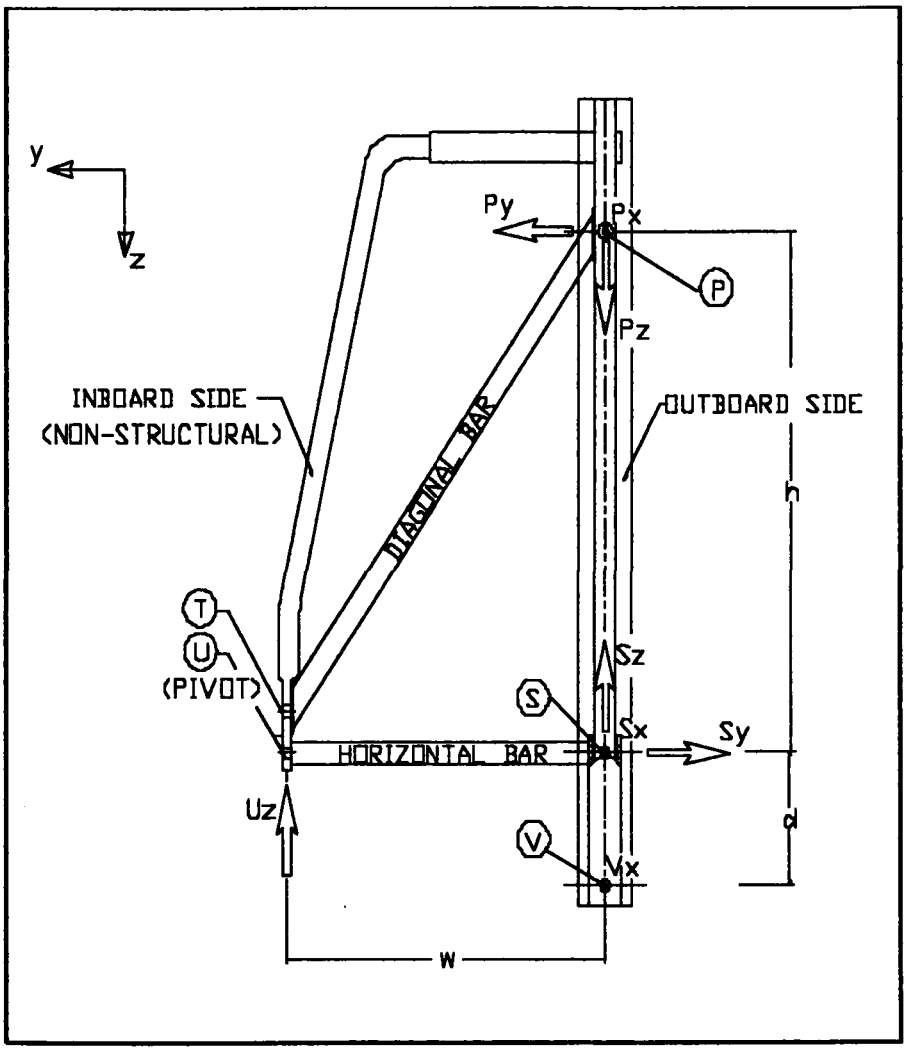


Fig. 2-5-1. Front view of free body diagram of the upper frame.

The magnitude of Px was measured with a dynamometer during static test no.7 (see Test Report No. 7 in appendix A). The magnitude of Px when each of the applied forces, F1 and F2, equalled 3,000 lbs is:

$$P_x = 1460 \cdot lb$$

The dimensions defined in Fig. 2-5-1., are:

$$h := 22 \cdot in$$

$$d := 5.625 \cdot in$$

$$w := 17.375 \cdot in$$

The angles of attack of the upper torso belt measured at point P are:

$$\alpha_p := 52 \cdot deg$$

$$\beta_p := 55 \cdot deg$$

$$\gamma_p := 57 \cdot deg$$

The tension of the upper torso belt, Tu, is given by:

$$T_u := \frac{P_x}{\cos(\alpha_p)}$$

then,

$$T_u = 2.371 \cdot 10^3 \cdot lb$$

The components of force in the y and z directions, respectively, may be evaluated from the following equations:

$$P_y := T_u \cdot \cos(\beta_p)$$

or

$$P_y := \frac{\cos(\beta_p)}{\cos(\alpha_p)} \cdot (P_x)$$

then,

$$P_y = 1.36 \cdot 10^3 \cdot lb$$

Also,

$$P_z := T_u \cdot \cos(\gamma_p)$$

or

$$P_z := \frac{\cos(\gamma_p)}{\cos(\alpha_p)} \cdot P_x$$

then,

$$P_z = 1.292 \cdot 10^3 \cdot lb$$

2.5.1.2. RIGID BODY REACTIONS (FROM FIG. 2-5-1):

From equilibrium of moments about the x-direction at point S, the following is obtained:

$$U_z := P_y \cdot \left(\frac{h}{w}\right)$$

then,

$$U_z = 1.722 \cdot 10^3 \cdot lb$$

From equilibrium of moments about the y-axis at point S, the following is obtained:

$$V_x := P_x \cdot \left(\frac{h}{d}\right)$$

then,

$$V_x = 5.71 \cdot 10^3 \cdot lb$$

By taking equilibrium of forces in the x-axis the following is obtained:

$$S_x := P_x + V_x$$

then,

$$S_x = 7.17 \cdot 10^3 \cdot lb$$

By taking equilibrium of forces in the y-axis the following is obtained:

$$S_y := P_y$$

then,

$$S_y = 1.36 \cdot 10^3 \cdot lb$$

By taking equilibrium of forces in the z-axis the following is obtained:

$$S_z := P_z - U_z$$

then,

$$S_z = -430.69 \cdot lb$$

2.5.1.3. OUTBOARD BEAM (SEGMENT PV IN FIG. 2-5-1)

The inboard side of the upper frame is free to pivot about the y-axis at point U. Therefore, the outboard side alone absorbs all the bending about the y-axis, which is generated by the component of force Px. The outboard side of the upper frame behaves as a cantilivered beam with end load Px and length h. Therefore, the maximum moment in the beam, which occurs at the cross section containing point S, is given by:

$$My_S := -h \cdot (P_x)$$

The maximum elastic compressive stress at S, due to bending alone is:

$$\sigma z_b = \frac{-My_S}{Ze_S}$$

where Zes is the elastic section modulus of the beam at S.

The compressive force between points P and S based on the free body diagram depicted in Fig. 2-5-1 equals Sz. However, because this model does not consider the differences in stiffness among the members in the

upper frame, the results obtained in section 2.5.1.2. must be used with caution. Since the outboard beam has a much higher stiffness than the diagonal bar, it will absorb more of Pz than the rigid-body calculations predicted. For design purposes, it is assumed that the outboard beam absorbs all of Pz. Therefore, the elastic compressive stress due to Pz alone is given by:

$$\sigma z_f = \frac{P_z}{A_S}$$

where As is the cross sectional area of the beam at S.

Therefore, the maximum total elastic compressive stress at S is:

$$\sigma z_S = \frac{-My_S}{Ze_S} + \frac{P_z}{A_S}$$

At collapse, the compressive stress becomes Yps and the elastic section modulus becomes Zpmin in the expression above. After making these substitutions and solving for Zpmin, we obtain the following expression.

$$Zp_{min} = \left(-h \cdot P_x\right) \cdot \frac{A_S}{\left(Yp_S \cdot A_S - \frac{\cos(\gamma_p)}{\cos(\alpha_p)} \cdot P_x\right)}$$

The equation above, which evaluates Zpmin, may be interpreted as follows:Any selected cross section of area As which has a plastic section modulus larger than Zpmin would be adequate to withstand Px. Several cross sections were analyzed and the cross section with highest strength over weight ratio was selected. Fig. 2-5-2. shows the selected cross section.

A computer program was written to evaluate the plastic section modulus, Zp, of the cross section depicted in Fig.2-5-2.

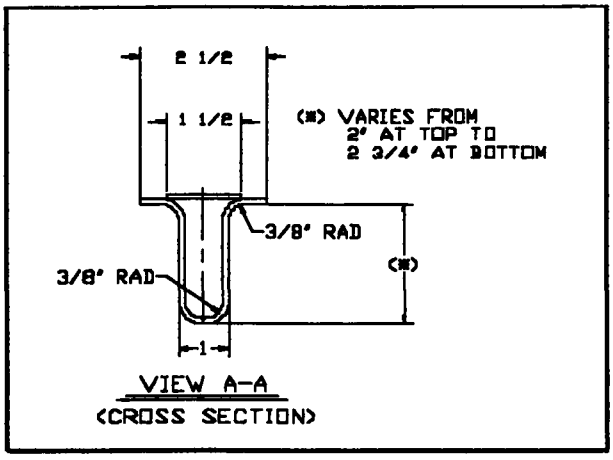


Fig. 2-5-2. Cross section of the outboard beam.

The calculation of Zp, as described in ref. [3] p.448, proceeds as follows:

"In any fully plastic cross section, forces parallel to the length of the beam must sum to zero. Therefore, half of the cross-sectional area carries tensile stress Y and force F, and half carries compressive stress Y and the same force F. The fully plastic moment is Mp = (YA/2)d, where d is the distance between centroids of the two areas A/2. Therefore, the plastic section modulus is Zp = Mp/Y = Ad/2".

Detailed calculations to obtain Zp have been omitted from this section to keep the presentation simple. A copy of the computer program is included in Appendix C for reference. For the selected beam, the cross sectional area and plastic section modulus at S , the cross section at which the bending moment is maximum, are:

$$A_S = 1.05 \cdot in^2$$

and

$$Zp = 1.17 \cdot in^3$$

The minimum yield stress of the selected material, Corten A per ASTM A606, is:

$$Yp_S = 50000 \cdot \frac{lb}{in^2}$$

Rearranging (1) and plugging the values of As and Yps, we find that:

$$Zp_{min} = 0.659 \cdot in^3$$

The selected cross section is adequate since Zp > Zpmin.
It must be noted that the outboard beam also carries the force Px in shear. However, this force has not been taken into account in the calculations since the effect of shear in a beam is not significant.

2.5.1.4. UPPER FRAME TRUSS (SEGMENTS PT, US AND SP IN FIG 2-5-1)

As discussed before, the outboard beam, segment PS, carries Px and Pz as well as all the moment about the y-axis, which is generated by Px.
However,the outboard beam could not carry the moment generated by Py because it is mounted on a gusset, pc # 5, which has little ability to resist a moment about the x-axis. In addition, the lower end of the outboard beam, point V in Fig. 2-5-1, is not restricted in the y-axis.

The two braces, segments PT and US, allow the upper frame to behave as a truss. Thus, the moment about the x-axis is carried by two equal and opposite forces at points U and S, respectively. The two braces are analyzed as follows:

DIAGONAL BRACE (SEGMENT PT):
As stated before, the stiffness of the outboard beam is so much larger than that of the diagonal brace, it may properly be assumed that the diagonal brace is not affected by Pz.
Therefore, the diagonal brace is only designed to resist Py in compression. The diagonal brace is analyzed for compressive stress as well as for buckling, as shown below.
The diagonal bar has the following properties:

Outside Diameter : $OD_{PT} = 1 \cdot in$

Inside Diameter: $ID_{PT} = 0.83 \cdot in$

Material Thickness: $t_{PT} := .083 \cdot in$

Length between Supports: $L_{PT} := 26.9 \cdot in$

Modulus of Elasticity: $E_{PT} := 30 \cdot 10^6 \cdot \frac{lb}{in^2}$

Moment of Inertia: $I_{PT} := 0.026 \cdot in^4$

Cross Sectional Area: $A_{PT} := 0.24 \cdot in^2$

Angle as Defined in Fig. 2-5-1: $\theta_y := 51 \cdot deg$

The compressive force in the diagonal brace which is required to resist Py is given by:

$$P_{PT} := \frac{P_y}{\cos(\theta_y)}$$

then,

$$P_{PT} = 2.161 \cdot 10^3 \cdot lb$$

The minimum yield stress for this member, which is made from ASTM A513 carbon steel tubing, is:

$$Y_{PT} := 30000 \cdot \frac{lb}{in^2}$$

The compressive stress in the diagonal brace is given by:

$$\sigma_{PT} := \frac{P_{PT}}{A_{PT}}$$

then,

$$\sigma_{PT} = 9.006 \cdot 10^3 \cdot \frac{lb}{in^2}$$

The selected diagonal brace is adequate from stress considerations alone, since the compressive stress is well below the yield stress of the material.

The critical buckling load for the diagonal brace, assuming that its ends are pinned, is given by:

$$Pcr_{PT} := \frac{\pi^2 \cdot E_{PT} I_{PT}}{L_{PT}^2}$$

then,

$$Pcr_{PT} = 1.064 \cdot 10^4 \cdot lb$$

The selected diagonal brace is also adequate for buckling, since the critical buckling load is larger than the actual compressive force.

HORIZONTAL BRACE (SEGMENT US IN FIG. 2-5-1):

It may be noticed in Fig.2-5-1 that no reaction in the y-axis has been considered at point U. This is because the inboard gusset, pc # 6, has little ability to resist a force applied in the y-direction. The diagonal brace, which acts in compression, exerts a force at point T. This force exerted at point T has a component in the z-direction and a component in the y-direction. The component of force in the y-direction exerted at point T is absorbed by the horizontal brace and transferred to point S. Based on the above, the design calculations which justify the selected horizontal brace are shown next.

The horizontal brace has the following properties:

Outside Diameter :	$OD_{US} := 1 \cdot in$
Inside Diameter:	$ID_{US} := 0.83 \cdot in$
Material Thickness:	$t_{US} := .0083 \cdot in$
Length between Supports:	$L_{US} := 26.9 \cdot in$
Modulus of Elasticity:	$E_{US} := 30 \cdot 10^6 \cdot \frac{lb}{in^2}$
Moment of Inertia:	$I_{US} := 0.026 \cdot in^4$
Cross Sectional Area:	$A_{US} := 0.24 \cdot in^2$

The tensile force that the horizontal brace must resist in the y-direction is:

$$P_{US} := P_y$$

The minimum yield stress for this member, which is made from ASTM A513 carbon steel tubing, is:

$$Y_{US} := 30000 \cdot \frac{lb}{in^2}$$

The tensile stress in the horizontal brace is given by:

$$\sigma_{US} := \frac{P_{US}}{A_{US}}$$

then,

$$\sigma_{US} = 5.667 \cdot 10^3 \cdot \frac{lb}{in^2}$$

The selected horizontal brace is adequate since the tensile stress is well below the minimum yield stress of the material.

2.5.1.5. BOLTS (POINTS P, T, U, S AND V IN FIG. 2-5-1)

BOLT AT POINT P:

The bolt at point P transfers the force applied by the upper torso belt to the upper frame. This bolt must resist the components of force Px and Pz in shear. The component of force Py does not generate additional shear stress on this bolt. The resultant shear force that this bolt must resist is given by:

$$F_P := \sqrt{P_x^2 + P_z^2}$$

then,

$$F_p = 1.949 \cdot 10^3 \cdot lb$$

The average shear stress in the bolt is given by:

$$\tau_p = \frac{F_p}{A_p}$$ *where Ap is the cross sectional area of the bolt*

The expression above may be rearranged as follows:

$$d_p = \frac{2 \cdot \left(P_x^2 + P_z^2\right)^{\frac{1}{4}}}{\sqrt{\pi \cdot \tau_p}}$$ *where dp is the bolt diameter*

In line with the maximum shear stress failure criteria, the allowable bolt shear stress is taken as 50% of the yield stress. The minimum yield stress for the selected bolt material, SAE J429 Gr.5,is 92000 psi. Therefore:

$$\tau_{Pallow} := 0.5 \cdot (92000) \cdot \frac{lb}{in^2}$$

or,

$$\tau_{Pallow} = 4.6 \cdot 10^4 \cdot \frac{lb}{in^2}$$

The minimum required bolt diameter is:

$$d_{Pmin} := \frac{2 \cdot \left(P_x^2 + P_z^2\right)^{\frac{1}{4}}}{\sqrt{\pi \cdot \tau_{Pallow}}}$$

then,

$$d_{Pmin} = 0.232 \cdot in$$

The value of dpmin obtained confirms that the 5/16"dia. bolt being used at point P is adequate.

BOLT AT POINT T:

The diagonal brace rests on the inboard side of the frame. Because this brace is in compression, it does not exert a significant load on the bolt located at point T. The bolt at point T may be considered a non-structural member for practical purposes. A 5/16" dia. bolt is used at this location.

BOLT AT POINT U:

The bolt at point U resists the component of force Py, which is exerted by the diagonal brace at point T, and transfers it to the horizontal bar in tension.

The diameter of the bolt and its cross sectional area, respectively, are:

$$d_U = 0.5 \cdot in$$

$$A_U = \frac{\pi}{4} \cdot d_U^2$$

then,

$$A_U = 0.196 \cdot in^2$$

The tensile stress in the bolt is given by:

$$\sigma_U = \frac{P_Y}{A_U}$$

then,

$$\sigma_U = 6.927 \cdot 10^3 \cdot \frac{lb}{in^2}$$

The minimum yield stress for the bolt material, SAE J429 Gr.5, is:

$$Y_U = 92000 \cdot \frac{lb}{in^2}$$

The selected bolt is adequate since the tensile stress is well below the minimum yield stress of the material.

BOLT AT POINT S:

The diameter of the bolt and its cross sectional area, respectively, are:

$$d_S = 0.5 \cdot in$$

$$A_S = \frac{\pi}{4} \cdot d_S^2$$

then,

$$A_S = 0.196 \cdot in^2$$

This bolt resists the reaction force Sy in tension. Therefore, the tensile stress in the bolt is given by:

$$\sigma_S := \frac{S_y}{A_S}$$

then,

$$\sigma_S = 6.927 \cdot 10^3 \cdot \frac{lb}{in^2}$$

This bolt also resists the reaction forces Sx and Sz in shear. The resultant shear stress resisted by the bolt is given by:

$$F_S := \sqrt{S_x^2 + S_z^2}$$

then,

$$F_S = 7.183 \cdot 10^3 \cdot lb$$

Because the bolt at point S resists the force Fs in double shear, the average shear stress in the bolt is given by:

$$\tau_S := \frac{1}{2} \cdot \left(\frac{F_S}{A_S} \right)$$

then,

$$\tau_S = 1.829 \cdot 10^4 \cdot \frac{lb}{in^2}$$

The effective stress in the bolt in accordance with the von Mises failure criterion is given by:

$$\sigma e_S := \frac{1}{\sqrt{2}} \cdot \sqrt{2 \cdot \sigma_S^2 + 6 \cdot \tau_S^2}$$

then,

$$\sigma e_S = 3.243 \cdot 10^4 \cdot \frac{lb}{in^2}$$

The minimum yield stress for the bolt material, SAE J429 Gr.5, is:

$$Y_S := 92000 \cdot \frac{lb}{in^2}$$

The selected bolt is adequate since the effective stress is well below the minimum yield stress of the material.

BOLT AT POINT V:

The force Px generates a moment of magnitude (Px)(h) about point S. This moment is resisted by the bolt at point V, which is located at a distance, d, below point S. Therefore, the force which this bolt must resist is given by:

$$F_V := P_x \cdot \left(\frac{h}{d}\right)$$

then,

$$F_V = 5.71 \cdot 10^3 \cdot lb$$

Because the bolt at point V resists the force Fv in double shear, its average shear stress is:

$$\tau_V = \frac{1}{2} \cdot \frac{F_V}{A_V}$$

where Av is the cross sectional area of the bolt

The expression above may be rearranged as follows:

$$d_V = \sqrt{\frac{2 \cdot P_x \cdot h}{\pi \cdot d \cdot \tau_V}}$$

In line with the maximum shear stress failure criterion, the allowable bolt shear stress is taken as 50% of the yield stress.

The minimum yield stress for the selected bolt material, SAE J429 Gr.5, is 92000 psi.

Therefore:

$$\tau_{Vallow} := 0.5 \cdot (92000) \cdot \frac{lb}{in^2}$$

then,

$$\tau_{Vallow} = 4.6 \cdot 10^4 \cdot \frac{lb}{in^2}$$

The minimum required bolt diameter is:

$$d_{Vmin} := \sqrt{\frac{2 \cdot P_x \cdot h}{\pi \cdot d \cdot \tau_{Vallow}}}$$

then,

$$d_{Vmin} = 0.281 \cdot in$$

The value of dvmin obtained confirms that the 1/2"dia bolt being used at points V is adequate.

2.5.2. Recline Mechanism

Althoughthe full implementation of a functional recline mechanism is beyond the scope of this work, a conceptual design has been developed and tested with the prototype. Furthermore, a layout of the final recline mechanism, which is yet to be built, is included in Appendix C.

A free body diagram of the recline mechanism is depicted in Fig 2-5-3.

The force Px generates a moment about the pivot point S. This moment is reacted by a force Fv, which must be ultimately absorbed by the recline mechanism. The recliner bar must be designed to take Fv in tension, whereas the recliner pins must take Fv in double shear. The calculations proceed as follows:

The diameter of the bolt which simulates the recliner set pin is:

$$d_p = 0.5 \cdot in$$

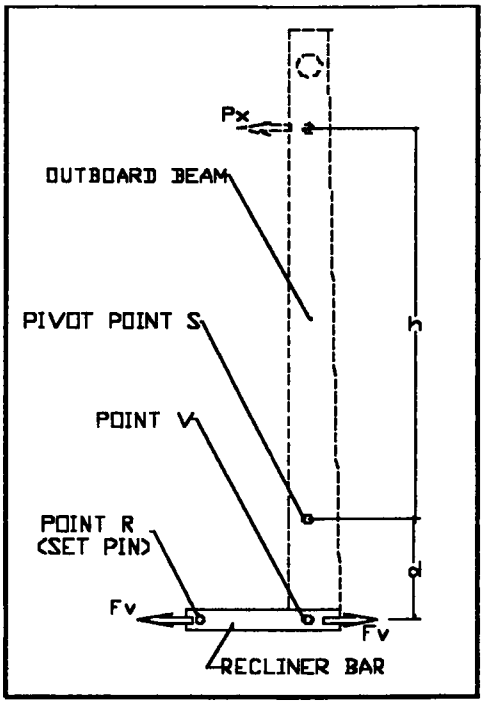


Fig. 2-5-3 Free Body Diagram of the Recline Mechanism.

The selected material for the bolt, SAE J429 Gr.5, has a minimum yield stress:

$$Yp = 92000 \cdot \frac{lb}{in^2}$$

In line with the maximum shear stress failure criteria, the allowable shear stress for the set pin equals 50% of the yield stress. Therefore,

$$\tau_{Rallow} = 0.5 \cdot (92000) \cdot \frac{lb}{in^2}$$

then,

$$\tau_{Rallow} = 4.6 \cdot 10^4 \cdot \frac{lb}{in^2}$$

This recliner pin resists the force Fv in double shear.

Therefore, the minimum required pin diameter is given by:

$$d_{pmin} := \sqrt{\frac{2 \cdot P_x \cdot h}{\pi \cdot d \cdot \tau_{Rallow}}}$$

Plugging values into the expression above, we find that:

$$d_{pmin} = 0.281 \cdot in$$

The selected diameter, dp, is adequate since it exceeds the minimum diameter, dpmin.

The recliner bar depicted in Fig 2-5-3, a 1 1/4" wide by 1/2" thick flat bar, was made from ASTM A36 carbon steel. The minimum yield stress for this material is:

$$Y_R := 36000 \cdot \frac{lb}{in^2}$$

The required cross sectional area of the recliner bar is given by:

$$A_{Rmin} := \frac{F_V}{Y_R}$$

then,

$$A_{Rmin} = 0.159 \cdot in^2$$

The cross section where the tensile area is minimum contains the 1/2" dia. pin hole. The effective area at this location is:

$$A_R := 0.375 \cdot in^2$$

The selected cross section is adequate since it exceeds the minimum required tensile area.

It should be obvious that the selected bar and set pin far exceed the minimum strength requirements. However, the intent of the formulation above is to establish the calculation procedure that would govern the design of the final mechanism which is yet to be manufactured. Upon selecting the construction materials for the final mechanism, the calculations above would be adjusted to reflect the actual minimum yield strengths.

2.5.3. Lower Frame

2.5.3.1. OUTBOARD SIDE PANEL

Fig. 2-5-4 shows the simplified model on which the calculations for the outboard side panel have been based.

The outboard side panel has been designed to withstand the moment generated by Px, which is located at a distance Zg above its neutral axis. The effects of Py and Pz have been omitted from this model. Also, the cross bar reactions have been placed along the neutral axis of the side panel for simplicity.

Dimensions B and E, as defined in Fig. 2-5-4, are:

$$B = 11.5 \cdot in$$

$$E = 3 \cdot in$$

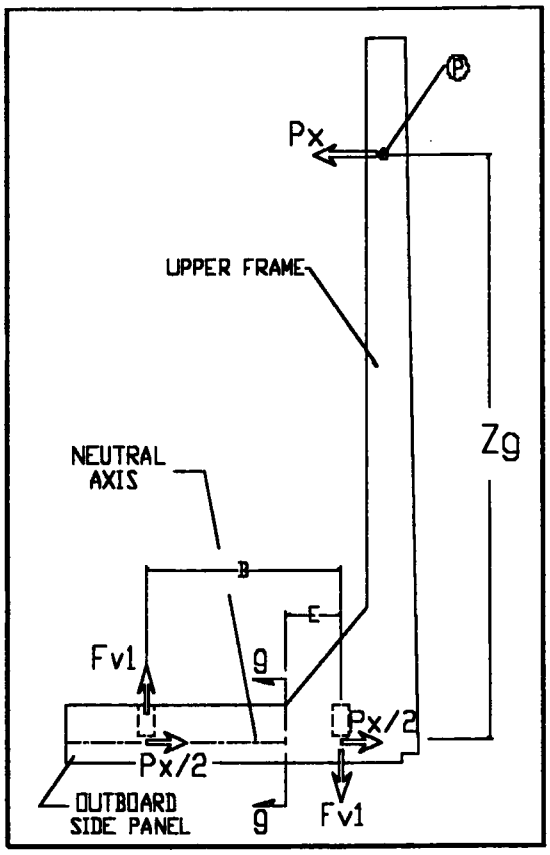


Fig. 2-5-4. Model for the Design of the Outboard Side Panel.

The selected cross section of the outboard side panel is depicted in Fig. 2-5-5.

A computer program was written to evaluate the distance to the neutral axis, Zg, as well as the plastic section modulus of the selected outboard side panel, Zpg. A copy of the computer program is included in Appendix C. The detailed calculations are omitted from this section in order to keep a simple presentation.

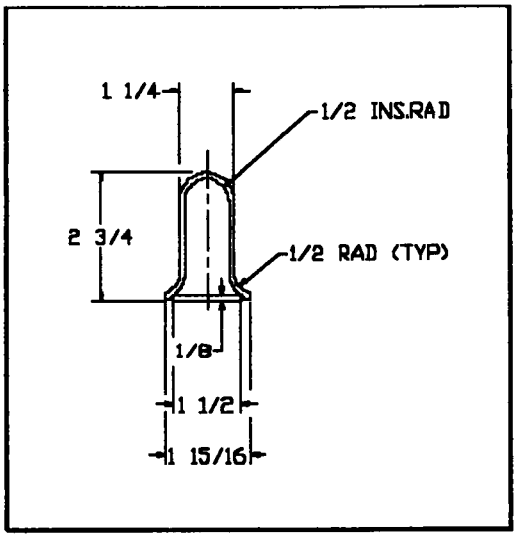


Fig. 2-5-5. Cross Section of the Outboard Side Panel.

The following results were obtained from the computer program:

Moment Arm: $Z_g := 27.62 \cdot in$

Plastic Section Modulus: $Zp_g := 0.712 \cdot in^3$

Cross-Sectional Area: $A_g := 0.877 \cdot in^2$

From static equilibrium, Fv1 is obtained as follows:

$$Fv_1 := P_x \cdot \left(\frac{Z_g}{B}\right)$$

then,

$$Fv_1 = 3.507 \cdot 10^3 \cdot lb$$

The maximum compressive stress at the cross section g-g is given by:

$$\sigma_g := \frac{Z_g \cdot P_x - Fv_1 \cdot E}{Zp_g} + \frac{P_x}{2 \cdot A_g}$$

then,

$$\sigma_g = 4.269 \cdot 10^4 \cdot \frac{lb}{in^2}$$

The outboard side panel was made from ASTM A606, which has a minimum yield stress:

$$Yp_g := 50000 \cdot \frac{lb}{in^2}$$

The selected outboard side panel is adequate since its maximum stres is lower than the minimum yield stress of the selected material.

2.5.3.2. CROSS BEAMS

The effect of Px on the cross bars is depicted in Fig. 2-5-6.

Dimension C, which is taken from point P to the middle of the cross bars, is:

$$C := 26.875 \cdot in$$

The vertical reaction, Fz1, which must be carried by each cross beam, is given by:

$$Fz_1 := P_x \cdot \left(\frac{C}{B}\right)$$

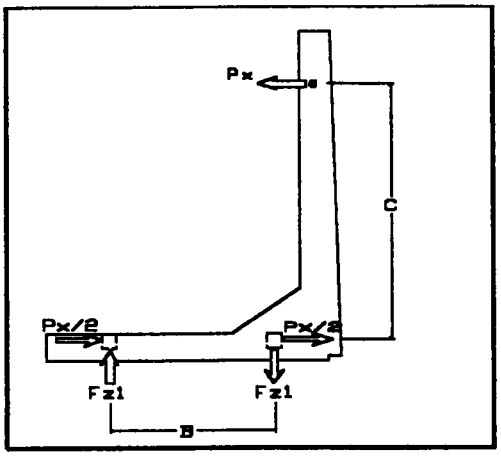


Fig. 2-5-6. Effect of Px on the Lower Frame Cross Beams.

then,

$$Fz_1 = 3.412 \cdot 10^3 \cdot lb$$

The effect of Py on the cross bars is depicted in Fig. 2-5-7.

Dimension D, as defined in Fig 2-5-7, is:

$$D = 16.25 \cdot in$$

From static equilibrium, the vertical force Fz2, which must be carried by each side, is given by:

$$Fz_2 = P_y \cdot \left(\frac{C}{D}\right)$$

then,

$$Fz_2 = 2.249 \cdot 10^3 \cdot lb$$

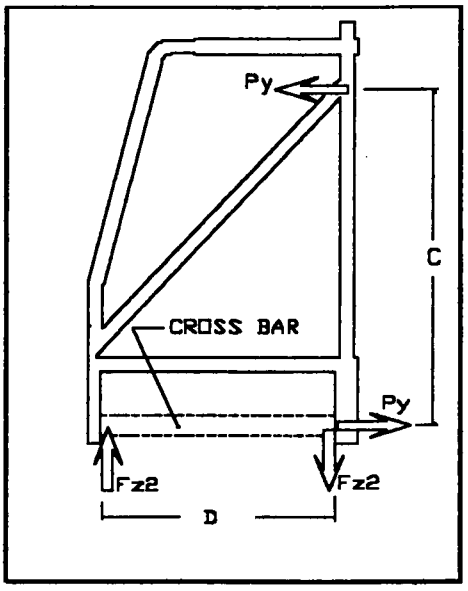


Fig. 2-5-7. Effect of Py on the Lower Frame Cross Beams.

Althoughthe reaction Fz2 on each side would be shared between the front and the rear cross bars, it is obvious that the rear cross bar would take most of Fz2. For practical purposes, it is assumed that the rear cross bar takes all of Fz2.

The effect of Pz on the cross bars is depicted in Fig. 2-5-8.

Dimension G, as defined in Fig. 2-5-8, is:

$$G = 2.5 \cdot in$$

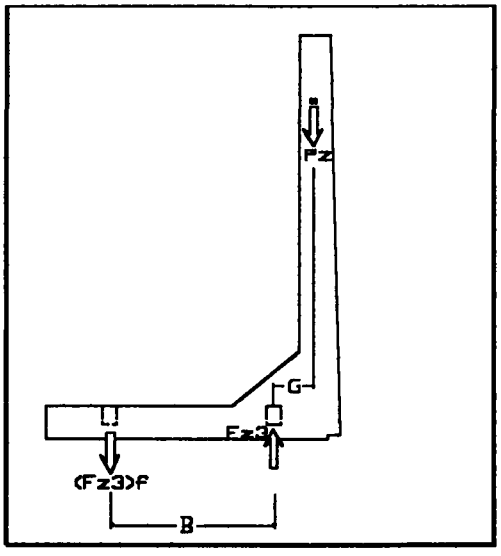


Fig. 2-5-8. Effect of Pz on the Lower Frame Cross Beams.

From static equilibrium, the vertical force Fz3, which the rear cross bar must carry, is given by:

$$Fz_3 = P_z \cdot \frac{B + G}{B}$$

then,

$$Fz_3 = 1.573 \cdot 10^3 \cdot lb$$

$$Fz_{3f} := P_z \cdot \left(\frac{G}{B}\right)$$

then, $Fz_{3f} = 280.87 \cdot lb$

From the above, it is clear that the rear cross bar reactions are larger than those of the front cross bar. The design calculations are made for the rear cross bar only, since the front and rear cross bars are identical.

The resultant vertical reaction at the rear cross bar is given by:

$$Fz := Fz_1 + Fz_2 - Fz_3$$

then, $Fz = 4.088 \cdot 10^3 \cdot lb$

The effect of the horizontal reactions has been omitted since their effect on the cross bars is relatively small.

The rear cross beam is analyzed as a cantiliver beam of length F and vertical end force Fz, as it is depicted in Fig. 2-5-9.

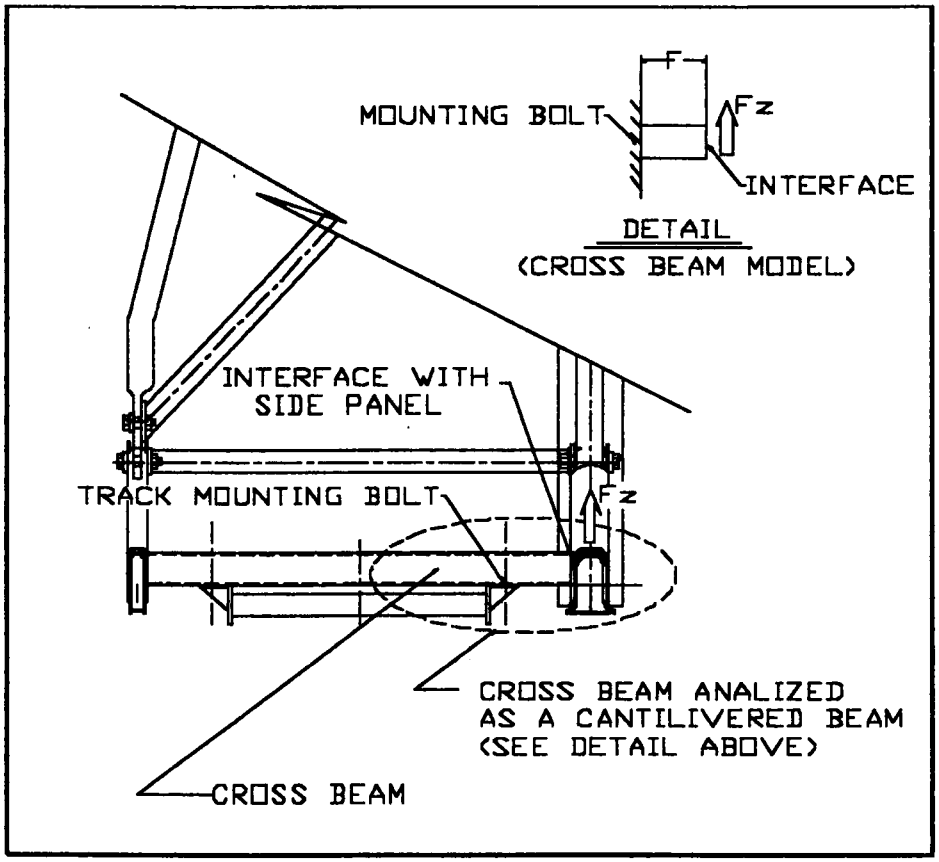


Fig. 2-5-9. Beam Model of the Rear Cross Bar.

The length of the beam, F, which is taken from the track bolt to the inboard face of the side panel, is:

$$F := 2.5 \cdot in$$

The maximum bending moment in the rear cross bar is given by:

$$M_b := F \cdot Fz$$

then,

$$M_b = 1.022 \cdot 10^4 \cdot in \cdot lb$$

The material specified for the cross bars is ASTM A513 with 40,000 psi minimum yield. Thus,

$$Yp_b := 40000 \cdot \frac{lb}{in^2}$$

The minimum plastic section modulus required to carry the moment Mb is given by:

$$Zp_{bmin} := \frac{M_b}{Yp_b}$$

then,

$$Zp_{bmin} = 0.256 \cdot in^3$$

The cross sectional dimensions of the cross bar are:

Long Side (Outside): $d := 1.5 \cdot in$

Short Side (Outside) $b := 1 \cdot in$

Long Side (Inside): $d_i := 1.26 \cdot in$

Short Side (Inside): $b_i := 0.76 \cdot in$

From Ref. [1], Table 1, Case 3, p.62, the plastic section modulus for a rectangular hollow cross section is given by:

$$Zp_b := \frac{b \cdot d^2 - b_i \cdot d_i^2}{4}$$

then,

$$Zp_b = 0.261 \cdot in^3$$

The selected cross bar is adequate since its plastic section modulus exceeds the required section modulus.

2.5.4. FLOOR REACTIONS

Fig. 2-5-10 shows all of the external forces applied to the seat and the corresponding floor reactions.

The upper torso force, F1, and the lap force, F2, as defined previously equal:

$$F_1 := 3000 \cdot lb$$

$$F_2 := 3000 \cdot lb$$

The force F3 must equal 20 times the total weight of the seat including the base.

Therefore, F3 depends on the type of base on which the seat is mounted.

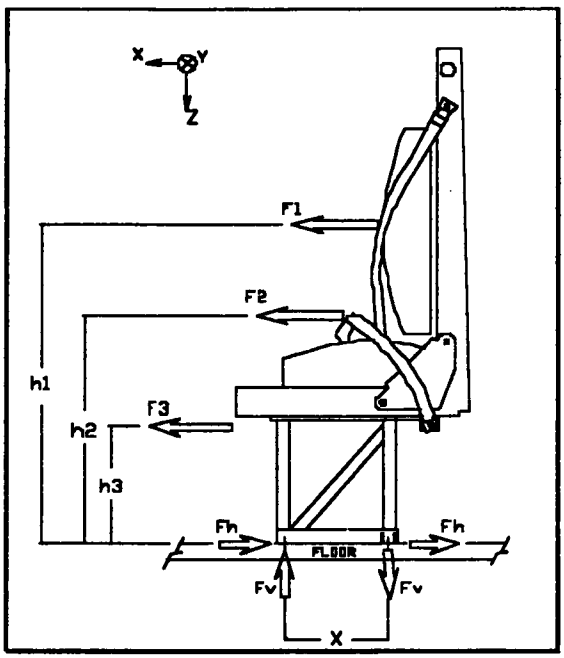


Fig. 2-5-10. Forces affecting the vehicle's floor.

Conservatively assuming that the total weight of the seat including the base equals 100 lb, the force F3 equals:

$$F_3 := 2000 \cdot lb$$

Dimensions h1, h2, and h3 can only be approximated. This is because the base on which the seat is mounted will vary. The following values are assumed:

$$h_1 := 34 \cdot in \quad h_2 := 22 \cdot in \quad h_3 := 15 \cdot in$$

The vertical reaction at the vehicle's floor, Fv, is a function of X. The value of X that applies to most standard applications is used for calculation purposes.

However, since the all-belts-to-seat concept is far from being a standard application, it will be the vehicle manufacturer's option to select a dimension X in order that the vertical force, Fv, may be supported by the vehicle's floor.

The vehicle's manufacturer must also specify the width of the base (dimension Y, not shown) and the number of floor-mounting bolts to be used. Since Fv is the total vertical force on the floor, the magnitude of Y does not affect the following calculations.

The following value of X applies to most current standard applications:

$$X := 11.25 \cdot in$$

The horizontal reaction, Fh, is obtained by taking equilibrium of forces along the x-axis. Thus,

$$F_h := \frac{F_1 + F_2 + F_3}{2} \cdot lb$$

then,

$$F_h = 1.814 \cdot 10^3 \cdot mass \cdot lb$$

The vertical reaction, Fv, is obtained by taking moment equilibrium about the y-axis. Thus,

$$F_v := \frac{F_1 \cdot h_1 + F_2 \cdot h_2 + F_3 \cdot h_3}{X}$$

then,

$$F_v = 1.76 \cdot 10^4 \cdot lb$$

CHAPTER 3

DISCUSSION AND CONCLUSIONS

A prototype which meets the strength requirements has been developed and tested. The prototype was validated during test No.9. The applied external forces, F_1 and F_2, were increased in 500 lbs. increments up to 3,000 lbs. The required forces were sustained for 30 seconds. No permanent deformation of the seat structure was noticed after the applied forces were released.

The final prototype weighs 38 lbs., approximately twice the weight of a conventional seat. Even though the weight has doubled, the increased weight might be considered relatively small in the case of a truck.

The weight of the prototype may be reduced by making some design refinements and/or by upgrading the material strength. However, because the current weight might not be excessive, it seems more important to present the as-is prototype to potential customers than it is to devote additional effort to reduce weight. Furthermore, because the all-belts-to-seat concept imposes additional strength requirements on the vehicle's floor, it is most important to determine if any additional modifications are required before the final product is implemented.

If weight became a consideration, the following

modifications could be implemented:

a) *The number of brackets (item #18) to which the inboard tracks are mounted could be reduced from 4 to 2.*

b) *The 1 1/4" OD x 11 GA adapter tube in the upper frame (item #16) could be eliminated by extending the length of the 14 GA upper frame (item # 1) to attach directly to the outboard beam (item #2).*

c) *The margin of safety on the outboard side panel (item #8) could be reduced upon refining the analysis.*

d) *The weight of the inboard gusset (item #6) and the inboard side panel might be reduced and yet satisfy the minimum strength requirements.*

e) *The cross section of the outboard side panel (pc #8), currently being a closed section, might be replaced by a U-shaped cross section while yet satisfying the minimum structural requirements.*

f) *Since the calculations indicate that the current outboard beam has a substantial margin of safety, the weight of this component could be reduced.*

g) *The calculations for the outboard beam have only been made for the cross section where the moment is maximum. Similar analysis could be performed for several cross sections along the beam in order to minimize its weight.*

h) *The material of the outboard beam, ASTM A606, has a minimum yield of 50,000 psi. This material could be upgraded to reduce the size of the beam. The same applies to the outboard gusset (item #5) and the outboard side panel (item #8).*

A finite element model of the prototype could be developed and adjusted to match the test data, although this falls beyond the scope of this thesis. A numerical model that fits the test data could be used as a starting point to convert the prototype to a final design.

If matching the test data with a finite element model were found not to be practical, the finite element model of the prototype could still be used as a starting point and then be modified to obtain qualitative information rather than quantitative data.

LIST OF REFERENCES

LIST OF REFERENCES

[1] Young, Warren C.: "Roark's Formulas for Stress & Strain," 6th ed., McGraw-Hill Book Company, New York, 1989.

[2] Baumeister, T., Avallone, E., and Baumeister II,T.: "Marks' Standard Handbook for Mechanical Engineers," 8th ed., McGraw-Hill Book Company, New York, 1978.

[3] Cook, Robert D., and Young, Waren C.: "Advanced Mechanics of Materials," Mac Millan Publishing Company, New York, 1985.

[4] "Code of Federal Regulations, Part 49: Transportation," Revision October 1, Washington, 1987.

[5] The American Institute of Steel Construction: "Steel Construction Manual," 5th ed., A.I.S.C., New york, 1949.

APPENDICES

APPENDIX A

TEST REPORTS

T E S T R E P O R T N o . 1

A.1.1. OBJECTIVE

a) To observe the overall behavior of a conventional seat during a static test when the upper torso belt is mounted on the seat's upper frame rather than on the vehicle's body.

b) To identify the location of the weakest member or weakest point (the first point of collapse during a static test) of a conventional seat with integrated seat belt.

c) To identify the mode of failure at the weakest member and to strengthen it to withstand the required loads ($F_1 = F_2 = 3{,}000$ lbs).

A.1.2. TEST DATA

TABLE A.1.

- TABULATED DATA - STATIC TEST NO. 1		
F_1, F_2 (Lbs.)*	d (In.)	Notes
100	2.00	(a)
150	2.50	-
200	3.00	-
250	3.50	-
300	4.00	-
350	4.50	(b)
400	5.25	-
450	6.00	-
500	6.50	-
550	7.50	(a)
600	8.50	(c)
650	n/a	(c), (d)

* Pre tension: F_1 = 116 Lbs.
F_2 = 58 Lbs.

Notes:

(a) When viewing the seat from the outboard side, the upper frame rotated counterclockwise as the applied forces increased (see side view of the seat - Fig. A.1.1.). This tendency to rotate was first noticed when F_1 and F_2 equalled 100 lbs. The measured rotation at $F_1 = F_2 =$ 550 lbs. was 5 degrees.



Fig. A.1.1. Side view of the seat. The seat rotated forward as the applied forces increased.

(b) When viewing the seat from the back (see Fig. A.1.2.), the structure rotated clockwise as forces F_1 and F_2 increased. The interconnecting bar to which the end points of the lap

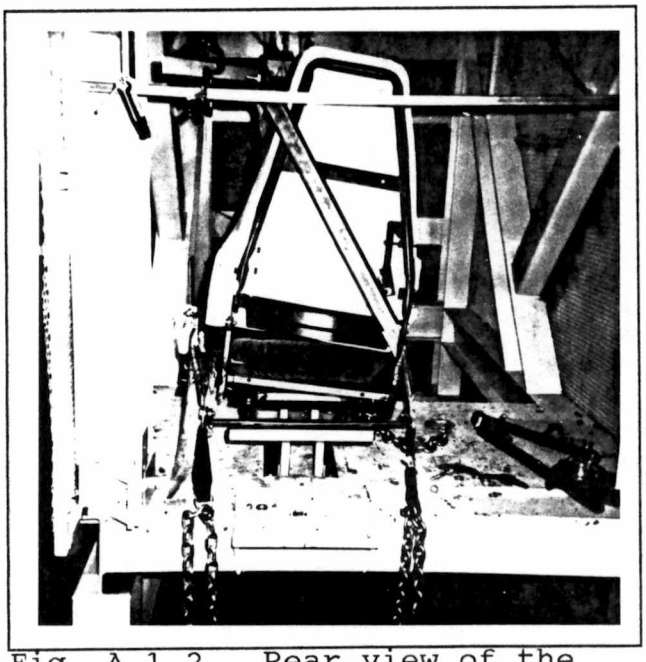


Fig. A.1.2. Rear view of the seat. The seat rotated sideways as the applied forces increased.

belt were mounted (see Fig. A.1.3.) began to bow upward when the applied forces equalled 350 lbs.

(c) The outboard side panel became visibly deformed when the upper torso and lap forces, F_1 and F_2, respectively, equalled 600 lbs. The outboard side panel collapsed when the applied forces equalled 650 lbs.

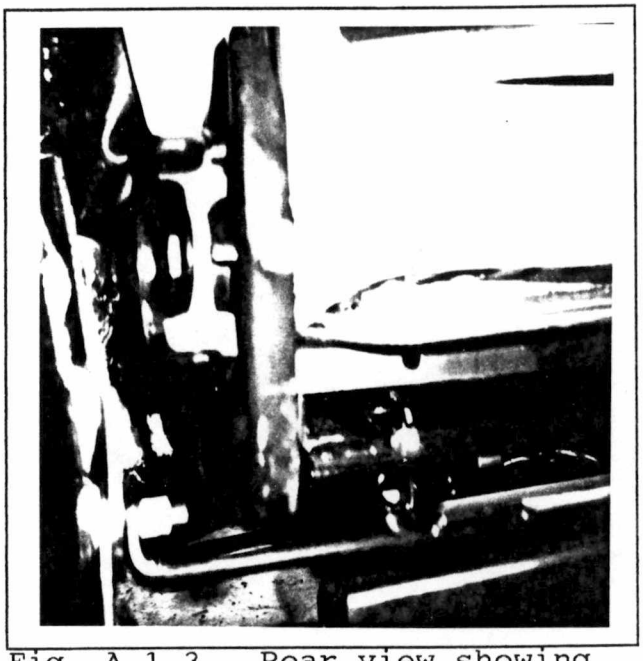


Fig. A.1.3. Rear view showing one end of the interconnecting bar attached to the lap belt.

(d) The rear bar of the bearing support frame appeared to be permanently deformed when the outboard side panel collapsed. However, because the deformation was so minor, it was difficult to determine whether or not it had been caused by the applied external forces.

A.1.3. TECHNICAL DISCUSSION

The counterclockwise rotation of the structure when observed from the outboard side of the seat was caused solely by the component of force P_x. No other components of applied force relate to this mode of deformation. From this reasoning, the magnitude of P_x could be obtained by allowing

the upper frame to collapse, i.e., knowing the plastic moment of the upper frame and the distance from the point of application of the force to the failure point makes it possible to identify the applied force, P_x. However, before F_1 and F_2 may be increased in order to collapse the upper frame, all the weaker members of the structure must be strengthened.

Using the collapse mode of the outboard side panel to identify P_x would be far more complicated. The mode of failure of the outboard side panel was a combination of bending and torsion. The bending and torsion of the outboard side panel are produced by P_x and P_y, respectively. Because the side panels have a complex geometry, their structural properties are not easily obtained without numerical analysis (finite elements or other similar method). Also, future modifications to the seat structure will affect the loading pattern on the side panels.

Rather than attempting to finalize the design of the side panels at this early



Fig. A.1.4. Original outboard side panel, which had a C-shape cross section.

stage, the outboard side panel was reinforced temporarily. In order to increase the torsional capacity of the outboard side panel, its cross section, originally being a C-shape (see Fig. A.1.4.), was replaced by a rectangular cross section (see Fig. A.1.5.); in general, a closed section takes far more torsion than an open section of comparable dimensions. It must be kept in mind that the main objective at this point is to identify P_x by allowing the upper frame to collapse before the side panel does. Once P_x is known, all the other components of force at points P and Q may be obtained from static equilibrium.

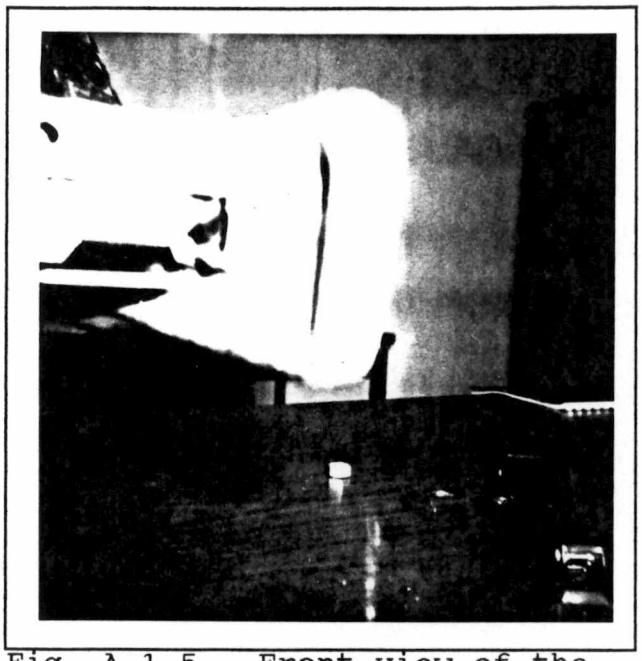


Fig. A.1.5. Front view of the modified outboard panel, which had a rectangular cross section.

The deflection of the interconnecting bar may be disregarded since the ends of the lap belt may be mounted to a different point in the structure. Furthermore, the interconnecting bar will be eliminated because no tethers are required in an integrated belt-in-seat design.

A.1.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 2

The outboard panel was reinforced as explained above.

T E S T R E P O R T N o . 2

A.2.1. OBJECTIVE

a) To increase the applied external forces, F_1 and F_2, until the upper frame collapses. If collapse does not occur at the upper frame, the failed member will be strengthened to outlast the current upper frame in future static tests.

b) To evaluate P_x, the component of shoulder force in the x-direction.

If the seat collapses at the upper frame, the plastic moment of the upper frame will be used to evaluate P_x.

If collapse occurs at any member other than the upper frame, the mode of failure and structural properties of the failed member will be identified.

c) To inspect the collapsed seat in order to identify any other members that might have permanently deformed.

A.2.2. TEST DATA

TABLE A.2.

- TABULATED DATA - STATIC TEST NO. 2		
F_1, F_2 (Lbs.)*	d (In.)	Notes
200	2.00	-
250	2.25	-
300	2.88	-
350	3.38	-
400	3.75	-
450	4.38	-
500	5.00	-
550	6.00	-
600	8.25	(a)
650	8.75	(b)
700	n/a	(c), (d)

* Pre tension: F_1 = 90 Lbs.
F_2 = 90 Lbs.

NOTES:

(a) Significant bending of the rear cross bar of the bearing support frame occurred at $F_1 = F_2 = 600$ lbs. (see Fig. A.2.1.).

(b) First indication of yield in the upper frame occurred at $F_1 = F_2 = 650$ lbs.

(c) Collapse of the rear cross bar occurred at $F_1 = F_2 = 700$ lbs. Excessive deformation of the rear cross bar forced the rear outboard bearing out of the bearing cup (see Fig. A.2.1.). The applied forces, F_1 and F_2, were released at this point due to the structure's inability to sustain the load because of the loose bearing.

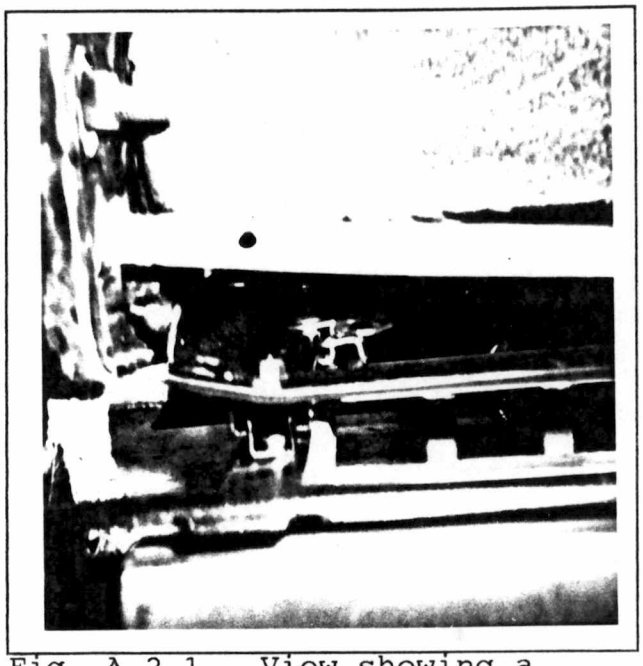


Fig. A.2.1. View showing a loose bearing resulting from excessive deformation of the rear cross bar of the bearing frame.

(d) The chugger track appeared to be permanently deformed upon collapse. Also, the upper frame remained plastically deformed after the applied forces were released (see Fig. A.2.2.).

A.2.3. <u>TECHNICAL DISCUSSION</u>

The rear cross bar, which collapsed at $F_1 = F_2 = 700$ lbs. (see Fig. A.2.1.), was made from flat bar 1 1/4" wide by 1/4" thick. The section modulus for this cross section is

0.0156 in^3. A representative average yield stress for this material, 0.09% maximum carbon hot rolled steel, is 40,000 psi. The shape factor, which is the ratio of plastic section modulus to elastic section modulus, equals 1.5 for a rectangular cross section [Ref. 1, Table 1, case 2, p. 62]. Therefore, the plastic section modulus for this member is 0.023 in^3.

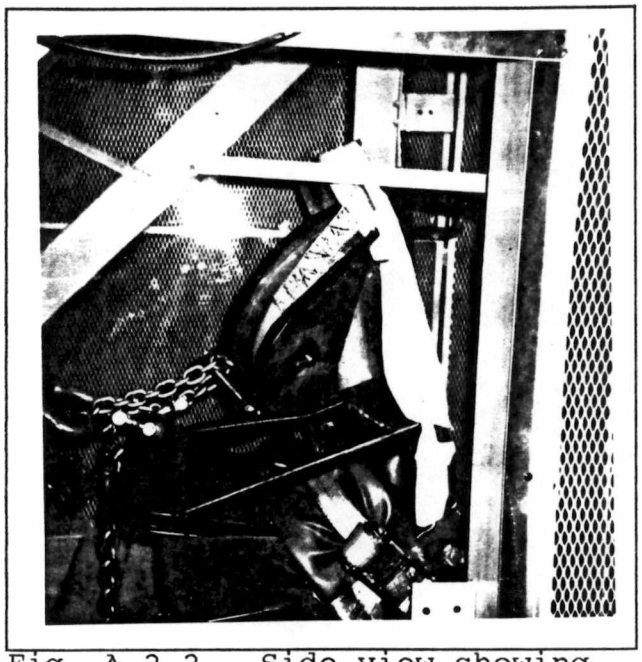


Fig. A.2.2. Side view showing the upper frame permanently deformed after the applied loads were released.

The plastic moment of a particular structural member, which is the bending moment required to produce plastic collapse, equals the product of its yield stress times its plastic section modulus. In the case of the rear cross bar, it may be concluded that the bending moment at its failure point equalled 920 in-lb. (40,000 psi x 0.023 in^3.) when both of the applied forces, F_1 and F_2, equalled 700 lbs.

The tension in the upper torso belt, T_u, may be evaluated from static equilibrium considerations and the conclusions obtained above. The angle between the upper torso belt and each of the reference axes may be measured at

any point. Therefore, the components of force at point P may be evaluated.

An additional conclusion was obtained by analyzing the collapse mode of the rear cross bar. The tension in the 5/16" diameter chugger track bolts equalled 375 lbs. when the rear cross bar collapsed ($F_1 = F_2 = 700$ lbs. at collapse), i.e., the tensile stress in the bolt when F_1 and F_2 reach 700 lbs. equals 4,889 lbs. The existing bolts, which are made from SAE J429 material with minimum yield stress of 81,000 psi, will be adequate for the required external forces, $F_1 = F_2 = 3{,}000$ lbs.

Collapse did not occur at the upper frame. However, the conclusions obtained from the mode of failure are of significant value.

A.2.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 3

a) The existing rear cross bar, a flat bar of dimensions 1 1/4" x 1/4" and elastic section modulus of 0.0156 $in.^3$, was replaced by an L-shaped beam of dimensions 1 3/8" x 1 3/8" x 10 GA (0.1345") with an elastic section modulus of 0.1 in^3.

b) The existing chugger track was replaced by a heavier track.

T E S T R E P O R T N o . 3

A.3.1. OBJECTIVE

The test objective is the same as that of Static Test No. 2 (see Test Report No. 2).

A.3.2. TEST DATA

TABLE A.3.

- TABULATED DATA - STATIC TEST NO. 3		
F_1, F_2 (Lbs.)*	d (In.)	Notes
400	4.00	-
500	7.00	-
600	9.00	-
650	10.00	(a)

* Pre tension: F_1 = 100 Lbs.
F_2 = 100 Lbs.

NOTES:

(a) The seat collapsed when the applied forces, F_1 and F_2, equalled 650 lbs. The upper frame bent forward at a point located 18 inches below point P (see Fig. A.3.1.).

A.3.3. *TECHNICAL DISCUSSION*



Fig. A.3.1. Side view showing the upper frame plastically deformed.

The magnitude of P_x may be evaluated from the collapse mode. The collapsed upper frame was made from 1" OD x 14 GA (0.083") tubing which has an elastic modulus equal to 0.051 in^3. Because this material is very thin, its plastic section modulus is approximately equal to its elastic modulus. A representative average yield stress for this material, hot rolled low carbon steel, is 59,539 psi.

Based on the above, the upper frame's plastic moment, which equals the product of the section modulus times the yield stress, is 3,036 in-lb. The force required to produce this amount of moment about a point at a distance of 18 inches is 169 lbs. Since P_x is the only component of force which contributed to collapse, it may be concluded that P_x equalled 169 lbs. when collapse occurred. This value of P_x is far below the value expected from static equilibrium considerations. If it were assumed that the upper torso force, F_1, was equally resisted at points P and Q, the value of P_x due to F_1 alone, $F_1/2$, would be 325 lbs. (650/2 = 325)

when the seat collapses. Since the contribution of F_2 to P_x has not been taken into account, the actual magnitude of P_x would be more than 325 lbs., which is far greater than 169 lbs. The discrepancy between the calculated P_x from collapse conditions (169 lbs.) and P_x from static equilibrium (325 lbs. minimum) is attributed to the fact that the force gauges used were intended for use at higher forces (force gauges are not necessarily accurate below their rated range). The next logical step was to increase the section modulus of the upper frame and to repeat the test at higher loads. By doing so, a more reliable value of P_x would be obtained upon collapse of the upper frame.

A.3.4. <u>DESIGN MODIFICATIONS FOR STATIC TEST NO. 4</u>

a) *The existing upper frame, which is made from 1" OD x 14 GA (0.083") tubing with section modulus equal to 0.051 in^3., was replaced by rectangular tubing of dimensions 2" x 1" x 0.070" with section modulus equal to 0.2*

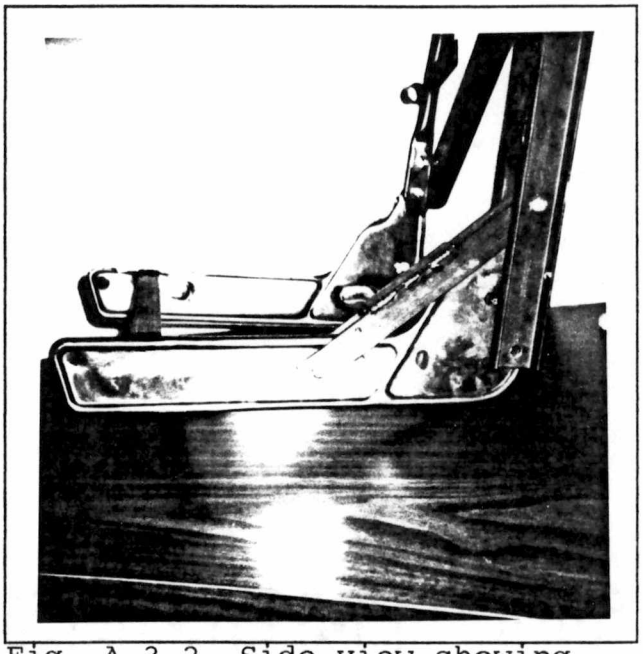


Fig. A.3.2. Side view showing the diagonal brace that was added to re-direct the load path.

in^3.,i.e., the plastic moment was increased by a factor of 10.

b) *The rectangular tubing was reinforced with two channels at the bottom. Braces were also added to prevent collapse of the outboard side panel.*

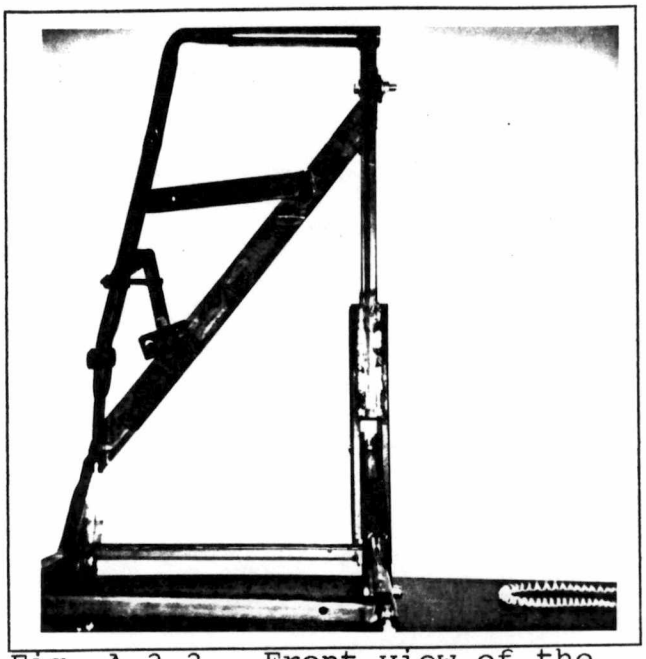


Fig. A.3.3. Front view of the seat. The upper frame was reinforced with channel sections.

T E S T R E P O R T N o . 4

A.4.1. OBJECTIVE

The test objective is the same as that of Static Test No. 2 (see test report No. 2).

A.4.2. TEST DATA

TABLE A.4.

- TABULATED DATA - STATIC TEST NO. 4		
F_1, F_2 *(Lbs.)**	*d (In.)*	*Notes*
650	*3.00*	*-*
750	*3.50*	*-*
900	*8.50*	*(a), (b)*

** Pre tension:* $F_1 = 100$ *Lbs.*
$F_2 = 100$ *Lbs.*

NOTES:

(a) The front bar of the bearing support frame collapsed when the applied forces equalled 900 lbs. The mode of collapse was bending (see Fig. A.4.1.).

(b) Upon collapse, the inboard side panel was slightly bent sideways (see Fig. A.4.2.).

A.4.3. TECHNICAL DISCUSSION

Collapse did not occur at the upper frame. However, some conclusions may be drawn from the mode of failure.

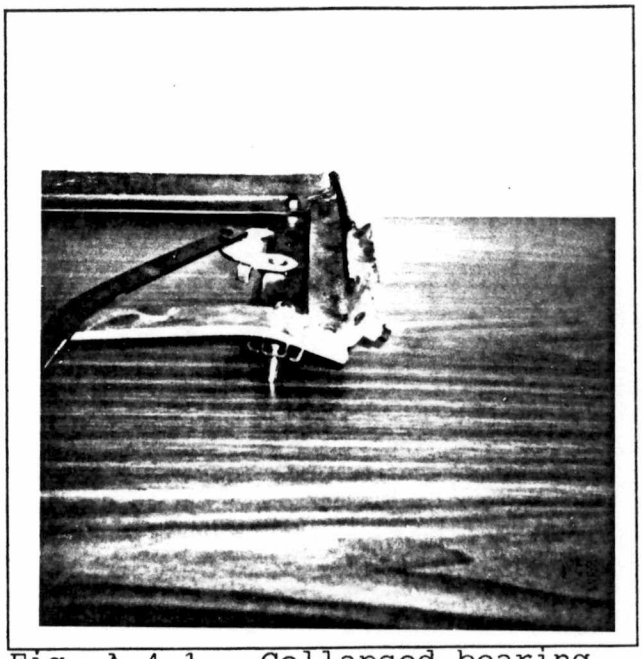


Fig. A.4.1. Collapsed bearing support frame.

The plastic section modulus of the collapsed member, a flat bar of dimensions 1 1/4" x 1/4", equals 0.023 in^3. As a starting point, it is assumed that the stress level on the seat structure increases linearly with F_1 and F_2 until the seat collapses. Since the front bar collapsed when F_1 and F_2 equalled 900 lbs., its plastic section modulus must be increased by a factor larger than 3.3 in order to withstand the required forces, $F_1 = F_2 = 3{,}000$ lbs. Therefore, the plastic section modulus of the front cross bar must exceed 0.077 in^3.

The sideways deflection of the inboard side panel is caused solely by Py, which is transferred by the upper frame's cross bar. To prevent the inboard side panel from collapsing at higher loads, two approaches may be considered. The first approach consists of increasing the section modulus of the inboard side panel. The alternate approach is to re-direct the load to the outboard side panel.

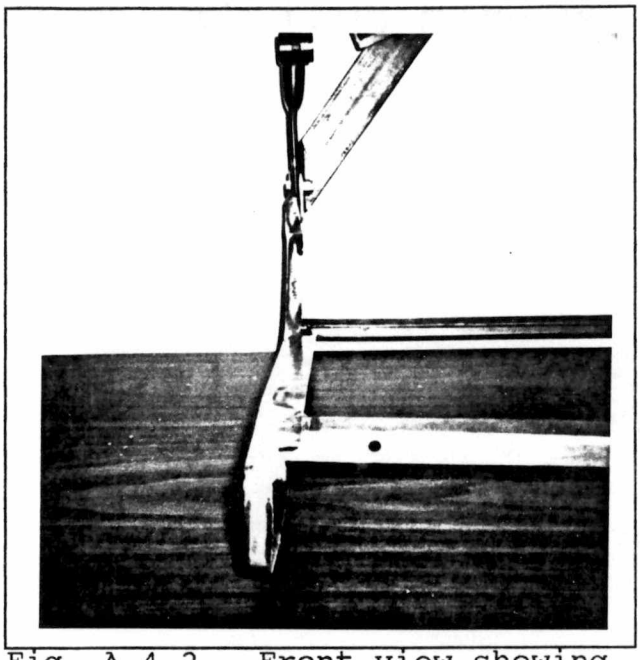


Fig. A.4.2. Front view showing the outboard side panel slightly bent sideways.

The latter of the two approaches is selected. A bar is added to connect the inboard side panel to the outboard side panel (see Fig. A.4.3.). Since this bar will be in tension, buckling is not a concern. Therefore, its weight may be minimized by designing its cross section for tensile stress only.

A.4.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 5

a) A piece made from 1" OD tubing was added to connect the inboard and outboard side panels (see. Fig. A.4.3.).

b) *An oversized base stand on which to mount the upper frame was made. This base stand will replace the bearing frame in future static tests until the upper frame meets the required loading. Once the upper frame design is final, efforts will be made to re-design the bearing frame.*

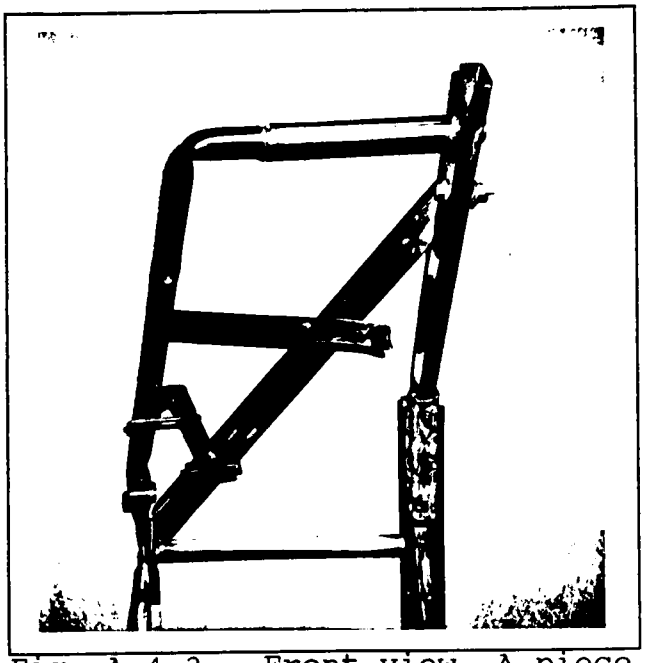


Fig. A.4.3. Front view. A piece of round tubing was added to connect the inboard and outboard side panels.

T E S T R E P O R T N o . 5

A.5.1. *OBJECTIVE*

The test objective is the same as that of Static Test No. 2 (see Test Report No. 2).

A.5.2. *TEST DATA*

TABLE A.5.

- TABULATED DATA - STATIC TEST NO. 5		
F_1, F_2 (Lbs.)*	d (In.)	Notes
900	2.00	-
1000	2.25	-
1100	2.50	-
1200	3.00	-
1300	3.25	-
1500	4.00	(a)
1700	n/a	(b)

* Pre tension: F_1 = 100 Lbs.
F_2 = 100 Lbs.

NOTES:

(a) The upper frame began to bend forward.

(b) The upper frame collapsed at a cross section located 13" below point P (see Fig. A.5.1.). The mode of failure was forward bending of the upper frame's left vertical post, which was made from 2" x 1" x 0.070" thick rectangular tubing.

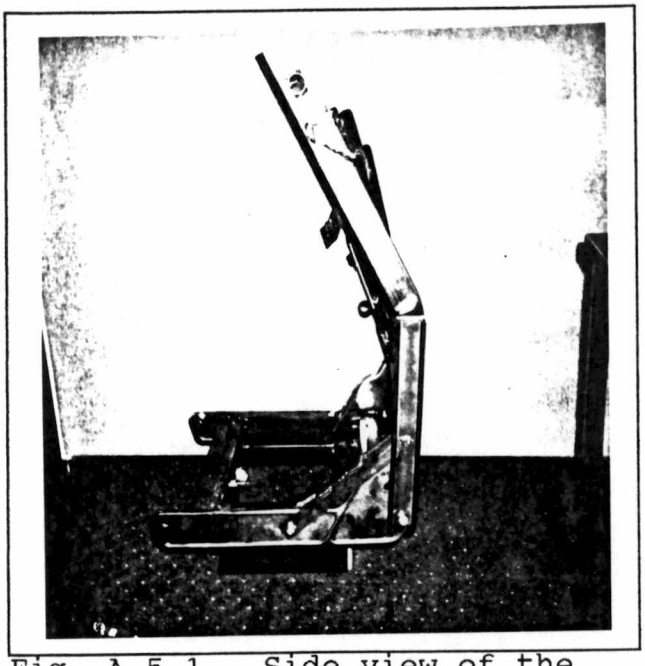


Fig. A.5.1. Side view of the seat showing the collapsed upper frame.

A.5.3. TECHNICAL DISCUSSION

The magnitude of P_x when the upper frame collapsed may be evaluated from the mode of failure.

The plastic section modulus of the collapsed member, which is made from 2" x 1" x 0.070" thick rectangular tubing, equals 0.256 in^3. [Ref. 1, p. 62, Table 1, case 3]. A representative yield stress for this material, low carbon steel, is 40,000 psi. Based on the above, the bending moment at the point of failure equalled 10,240 in-lb. when collapse occurred. The force required to produce a bending moment of 10,240 in-lb. about a point at a distance of 13 inches is 788 lbs. It is concluded that the magnitude of P_x was 788

lbs. when the applied forces, F_1 and F_2, equalled 1700 lbs. If the stress level on the upper frame increased linearly with the applied forces, the magnitude of P_x corresponding to $F_1 = F_2 = 3{,}000$ lbs. would be 1,391 lbs. (788 x 3,000/1700 = 1,391).

The magnitude of P_y and P_z may be defined in terms of P_x, and the angle formed by the belt and the y and z axes, respectively.

A.5.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 6

The upper frame structural post was modified to withstand the required loads. A variable cross section was selected in order to minimize weight. The plastic section modulus at the point of collapse, which was located 13" below P, equalled 0.24 in^3. The plastic section modulus at the point where the bending moment is maximum, which is located 22" below P, now equals 0.95 in^3. Also, the material of the upper frame post was upgraded from ASTM A513 (30,000 psi minimum yield), to ASTM A606 Corten A (50,000 psi minimum yield).

T E S T R E P O R T N o . 6

A.6.1. OBJECTIVE

To test the upper frame at the required loads ($F_1 = F_2 =$ 3,000 lbs).

A.6. TEST DATA

TABLE A.6.

- TABULATED DATA - STATIC TEST NO. 6		
F_1, F_2 (Lbs.)*	d (In.)	Notes
1000	0.500	-
1500	0.675	-
2000	0.750	-
2500	0.875	-
2750	1.000	-
3000	1.125	(a)

* Pre tension: $F_1 = 112$ Lbs.
$F_2 = 115$ Lbs

Angles:
$\alpha_p = 52$ deg.
$\beta_p = 55$ deg.
$\tau_p = 57$ deg.

NOTES:

(a) The applied forces were sustained for 30 seconds. No yield occurred when the applied external forces, F_1 and F_2, equalled 3,000 lbs.

A.6.3. TECHNICAL DISCUSSION

The test confirmed that the upper frame was adequate for the required loads. Figures A.6.1. and A.6.2. show that no plastic deformation of the seat occurred when the applied loads reached their maximum values (F_1 = F_2 = 3,000 lbs). The upper frame will be re-used in future tests, while the effort will be placed on developing an adequate lower frame.

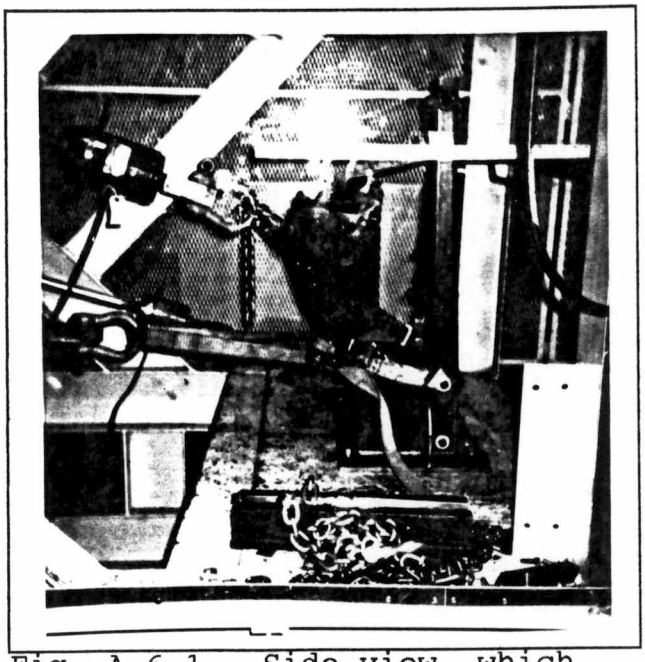


Fig. A.6.1. Side view, which shows the upper frame sustaining the required loads.

α_p, β_p, and τ_p ,the angles formed by the seat belt and the x, y, and z axes, respectively, at point P, were measured with a mechanical protractor. Their values are shown in section A.6.2. The magnitudes of T_u, P_y, and P_z are calculated from P_x, α_p, β_p,

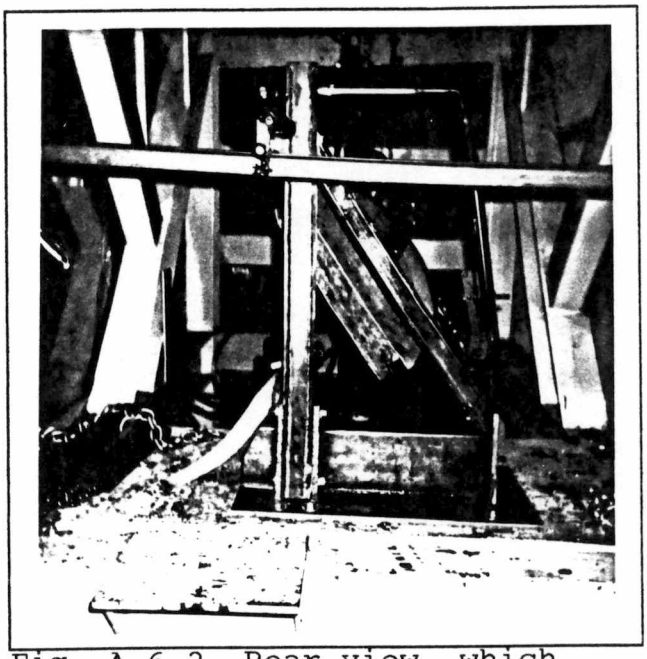


Fig. A.6.2. Rear view, which shows the seat sustaining the required loads.

and τ_p in section 2.2.1.

A.6.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 7

The outboard side of the upper frame is supported by two bolts. The lower bolt was removed to allow the upper frame to pivot about the middle bolt. A dynamometer was attached to point P and affixed to the wall. By doing so, the dynamometer will sustain record the component of force P_x.

T E S T R E P O R T N o . 7

A.7.1. OBJECTIVE

To measure the magnitude of P_x as a function of F_1 and/or F_2.

A.7.2. TEST DATA

TABLE A.7.

F_1 and/or F_2 (Lbs.)	Measured Magnitude of P_x as function of F_1 and/or F_2 (Lbs.)		
	F_1 ONLY	F_2 ONLY	F_1 and F_2
500	180	0	220
1000	400	20	480
1500	640	40	680
2000	880	80	940
2500	1140	100	1160
3000	1400	120	1460

A.7.3. TECHNICAL DISCUSSION



Fig. A.7.1. Test no. 7 setup.

The test results indicate that the effect of F_z on P_x is small. Because of the very high friction produced at the belt loop (point Q in Fig. 2.1.1.), a very small portion of the tension in the lap belt is transferred to the upper torso belt. The seat belt assembly used during this test, which was supplied by Indiana Mills Co., was the most likely one to be used in an actual case scenario.

A.7.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 8

The outboard side of the lower frame was made to fit the upper frame. Detailed sketches of the added items are included in Test Report No. 8.

T E S T R E P O R T N o . 8

A.8.1. OBJECTIVE

a) To validate the design of the outboard gusset (item #5).

b) To measure the magnitude of P_y as a function of F_1.

A.8.2. TEST DATA

TABLE A.8.

- TABULATED DATA - STATIC TEST NO. 8	
F_1 *(Lbs.)*	F_y *(Lbs.)*
520	*120*
1030	*220*
1530	*340*
2040	*400*
2500	*540*
3000	*640*

NOTES:

(a) The forces recorded with the dynamometer were taken without the upper frame's diagonal brace in place.

(b) The test was repeated with the diagonal brace in place but without the dynamometer.

A.8.3. *TECHNICAL DISCUSSION*

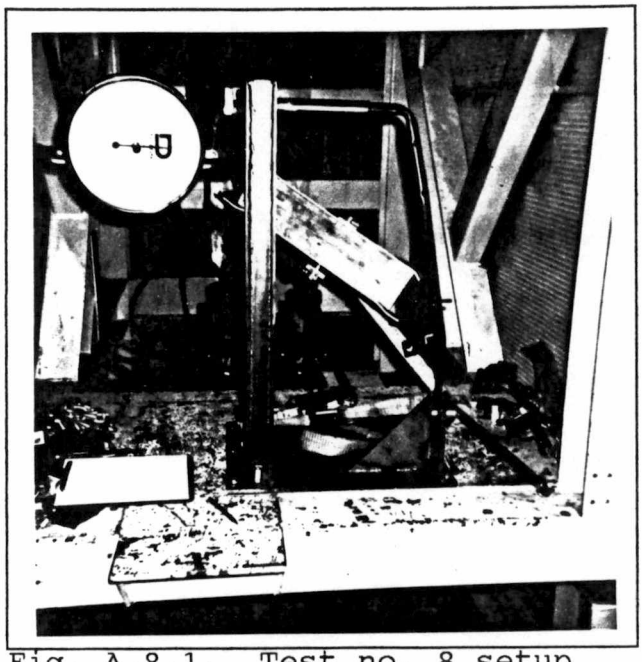


Fig. A.8.1. Test no. 8 setup (rear view).

The outboard gusset (item #5) must be capable of sustaining the moment about the y-axis, which is generated by P_x. The gusset must also resist the upward force required to resist the moment generated by P_y. The component P_z imposes a downward force on the gusset. The loads absorbed by the outboard gussets are then transferred to the outboard side panel. The outboard panel was not validated during test no.8, since the panel was bolted directly to the floor.

Although the main intent was to qualify the outboard gusset, an additional task was accomplished during this test: a dynamometer was affixed to the outboard side of the seat to measure the component of force P_y. As expected, the values recorded for P_y were lower than those predicted analytically. This was because the force recorded in the dynamometer was only a portion of P_y. The remainder of P_y was shared between the outboard gusset (item #5) and the non-structural portion of the upper frame. However, the recorded

values suggested that the predicted magnitude of P_y was not unreasonable.

A.8.4. DESIGN MODIFICATIONS FOR STATIC TEST NO. 9

The bottom flanges of the outboard side panel were trimmed off and the cross section was closed by welding a plate strip to the bottom of the side panel. Two cross bars were welded to tie the left and right side panels. The cross bars were made from 1 1/2" x 1" x 11 ga. rectangular tubing. Two brackets were welded to each cross bar before mounting the complete seat to a fixed heavy base.

T E S T R E P O R T N o . 9

A.9.1. OBJECTIVE

a) To validate the final prototype.

A.9.2. TEST DATA

TABLE A.9.

- TABULATED DATA - STATIC TEST NO. 9	
F_1, F_2 (Lbs.)	Notes (Lbs.)
500	-
1000	-
1500	-
2000	-
2500	-
3000	(a)

NOTES:

(a) The required forces were held for 30 seconds. No permanent deformations were noticed after the loads were released.

A.9.3. *TECHNICAL DISCUSSION*

Fig. A.9.1. is a side view of the final prototype sustaining the required loads. A slight forward rotation of the seat was noticed in this view. However, no plastic deformations seemed to occur since the seat sprang back to its original position when the loads were released.

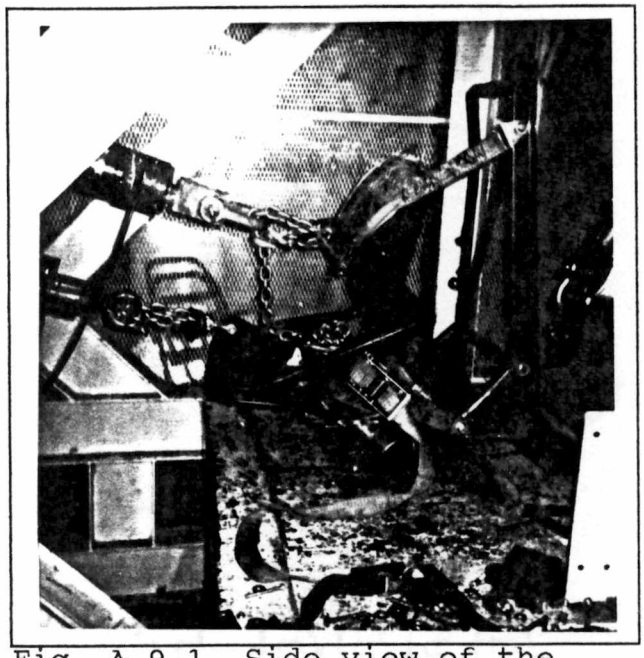


Fig. A.9.1. Side view of the final prototype. The required loads were held for 30 seconds.

The rear view depicted in Fig. A.9.2. was also taken while the required loads were being applied. No sideways deformation of the seat was visible at any time. An all-belts-to-seat prototype which meets the strength requirements has been developed and tested.



Fig. A.9.2. Rear view of the final prototype. No deformations could be noticed from this view while the required loads were being held.

APPENDIX B

MATERIAL PROPERTIES

TABLE B.1.

- MINIMUM YIELD STRENGTH OF SELECTED MATERIALS -	
Material Type (Lbs.)	*Min. Yield (Psi)*
Low Carbon Steel	*30,000*
ASTM A513	*30,000*
ASTM A606	*50,000*
SAE J429 Gr.5	*92,000*

APPENDIX C

MISCELLANEOUS NOTES

VARIABLE SHEET

St	Input	Name	Output	Unit	Comment
					Subject: Plastic Section Modulus (Zp) of Upper Frame Left Post
					By: AEU
					Date of Last Rev.: Oct/25/93
					*** INPUT DATA ***
	2.5	a		in	Overall Width
	1.5	b		in	Width of Reinforcing Bar
	2.875	c		in	Depth (Do Not Include Reincorcing Bar)
	.11	t1		in	Thickness
	.11	t2		in	Thickness
	.4375	R1		in	Radius
	.4375	R2		in	Radius
	50000	Y		psi	Yield Stress
					*** RESULTS ***
		Zp	1.1700744	in^3	Plastic Section Modulus
		A	1.049829	in^2	Area of Cross Section
		Mp	58503.718	in-lb	Plastic Moment

RULE SHEET

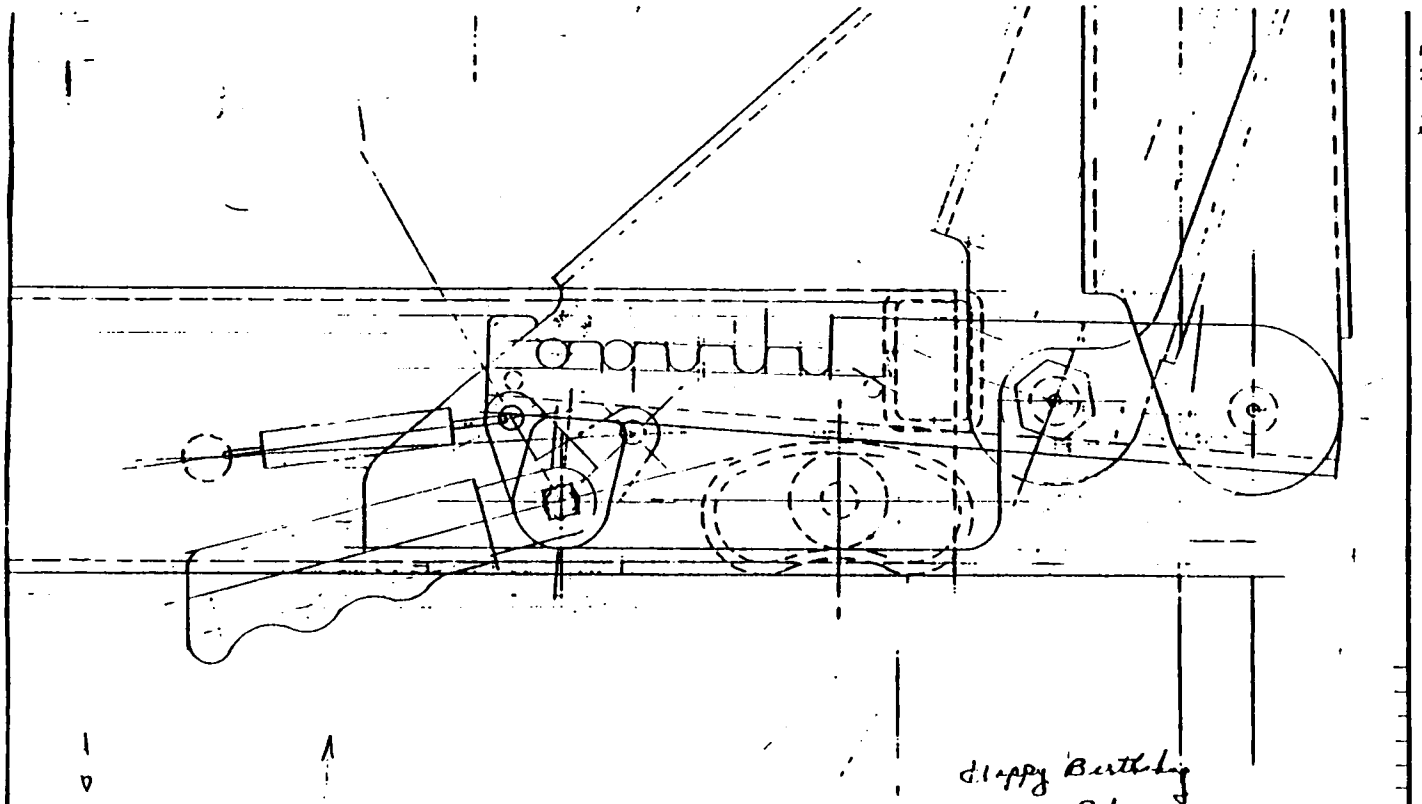
S	Rule
*	R1i=R1-t1/2
*	R1o=R1+t1/2
*	R2i=R2-t1/2
*	R2o=R2+t1/2
*	f=(a-2*(R1+R2))/2
*	pi=3.1416
*	e=c-t1-R2o
*	A1=(pi/2)*(R2o^2-R2i^2)
*	A2=e*t1
*	A3=(pi/4)*(R1o^2-R1i^2)
*	A4=f*t1
*	A5=b*t2
*	A=A1+2*(A2+A3+A4)+A5
*	y3=(1/(2*t1))*(A/2-A1)
*	y1=y3+R2+t1/2
*	y4=(4/(3*pi))*(R2o^3-R2i^3)/(R2o^2-R2i^2)
*	y5=e-y3
*	y6=(4/(3*pi))*(3*R1o^2-3*t1*R1o+t1^2)/(2*R1o-t1)
*	y7=c-y1-t1/2
*	y8=c-y1+t2/2
*	d1=(2/A)*(y3*A2+(y3+y4)*A1)
*	d2=(2/A)*(A2*y5+2*A4*y7+2*A3*(y5+y6)+A5*y8)
*	d=d1+d2
*	Mp=Y*(A/2)*d
*	Zp=Mp/Y

VARIABLE SHEET

St	Input	Name	Output	Unit	Comment
					Subject: Plastic Section Modulus (Zp) of Outboard Side Panel
					file name: panelzp
					By: AEU
					Date of Last Rev.: Feb/24/94
					*** INPUT DATA ***
	.5	ri		in	Inside Radius
	2.75	L		in	Beam Height
	.125	t1		in	Thickness (Bent Plate)
	.125	t2		in	Thickness (Bottom Bar)
	50000	Ypg		psi	Yield Stress
					*** RESULTS ***
		Ag	.87714375	in^2	Cross Sectional Area
		Zg	27.620713	in	Dist. to Neutral Axis from Point P
		Zpg	.71215764		Plastic Section Modulus
		Mpg	35607.882		Plastic moment

RULE SHEET

S	Rule
	pi=3.1416
	ro=ri+t1
	L1=L-ro-t2
	A1=(pi/2)*(ro^2-ri^2)
	A2=L1*t1
	A3=2*ro*t2
	Ag=A1+2*A2+A3
	L3=1/(2*t1)*(Ag/2-A3)
	L2=L1-L3
	bi=4/(3*pi)*ri
	bo=4/(3*pi)*ro
	U1=(L2+bo)*(pi/2*ro^2)
	U2=(L2+bi)*(pi/2*ri^2)
	U3=2*L2*t1*(L2/2)
	d1=(2/Ag)*(U1-U2+U3)
	D1=L3^2*t1
	D2=A3*(L3+t2/2)
	d2=(2/Ag)*(D1+D2)
	d=d1+d2
	Zpg=Ag*d/2
	Mpg=(Ypg*Ag/2)*d
	Zg=ro+L2+26.125



PART NO.	DESCRIPTION	ITEM
	PARTS LIST	

TOLERANCE	DO NOT SCALE	REQUIRED	TITLE	PART NO	REV
DECIMALS FRACTIONS ANGULAR	DRAWN R. Smith	SCALE FULL			
	CHECK	DATE 12-19-93			

NATIONAL SEATING CO. VONORE, TENNESSEE

VITA

Alvaro Eduardo Urrutia was born in Lima, Peru, on July 29, 1959. He attended elementary schools in a San Isidro (Lima) private school and graduated from the Maristas San Isidro high school in December, 1976.

The following March, he entered the National School of Engineering in Peru and completed three years in the program of Mechanical Engineering. In January of 1982, he entered the University of New Orleans and in December, 1983, received the degree of Bachelor of Science in Mechanical Engineering.

While working full time as a mechanical engineer, he completed engineering graduate level courses at the University of Houston in Texas, the Embry Riddle Aeronautical University in Florida, the San Diego State University in California, and the University of Tennessee in Knoxville. In May, 1994, he received a Master of Science degree in Engineering Mechanics from the University of Tennessee, Knoxville.

He is presently employed as a staff engineer at National Seating in Vonore, Tennessee.

To the Graduate Council:

I am submitting herewith a thesis written by Kent Gallaher entitled " Comparative Residual Concentrations and Activity of Selected Corn and Soybean Herbicides ". I have examined the final 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 Plant and Soil Science.

Thomas C. Mueller, Major Professor

We have read this thesis
and recommend its acceptance

Accepted for the Council:

Vice Provost
and Dean of the Graduate School

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William A. Poppen

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and recommend its acceptance:

Marla P. Peterson

Charles F. Thompson

Acceptance for the Council:

The Graduate School

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Acceptance for the Council:

The Graduate School

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Virginia L. Godfrey, Major Professor

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and recommend its acceptance:

John Ely Wilkinson

Liane B. Russell

E M Rinchik

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