3Q120M Magnet Transport Project Proposal

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Abstract:

The tunnel for the Neutrinos at the Main Injector project (NuMI) is currently under construction. A beam of protons will be shot through this tunnel. These protons will strike a fixed target and produce neutrinos. The neutrinos will be detected at two different places and whether or not they have mass will be explored. A near detector at Fermilab and a far detector in Soudan, Minnesota are also being constructed. The magnets that will be used to steer and focus the beam in various ways have to be moved into this tunnel. It is my task to design a device to do just this. The purpose of this paper is to walk you through the design process, and explain why the device I have designed is an adequate design for the task.

Introduction:

BACKGROUND: The construction for the NuMI project began in 1998. The tunnel being excavated will contain the beam line that will shoot neutrinos to an iron mine in Minnesota. Experiments will hopefully begin in 2005. The goal of the project is to further study the elusive neutrino. A neutrino is very small particle that has little or no mass. What makes it so hard to study the neutrino is it rarely interacts with matter. There are three different neutrinos that make up the standard model and fall under the category of leptons. The electron, muon, and tau neutrinos are the three types of neutrinos. Recently, experiments in Japan have been carried out that indicate neutrinos may oscillate between the types of neutrinos. If this is the case, neutrinos do indeed have mass. The NuMI project will try to confirm the results of the Japanese experiments.

To create a muon neutrino beam, a beam of protons, from the main injector, will proceed down the tunnel and collide with a fixed target. The collision will produce a secondary beam of pions and kaons. Some of these pions and kaons will decay into muon neutrinos and muons. The pipe in which this decay takes place is 2215 feet long. Layers of aluminum and steel are also used to absorb various particles that need to be removed. The beam then travels through 750 feet of rock and steel. By doing this, the pions, kaons, and muons will be absorbed leaving a nearly pure beam of muon neutrinos.

This beam of neutrinos will pass through Fermilab's "near" detector. The composition of the beam will be analyzed. The near detector will be used to calibrate the beam and the "far" detector. The "far" detector is in an old iron mine in Soudan, Minnesota. The beam will pass through the ground all the way to Minnesota. This can be done because neutrinos rarely interact. By having two detectors hundreds of miles apart, the neutrinos are given a chance to oscillate, if the neutrinos really do. The "far" detector will look for the two of the three different kinds of neutrinos, muon and electron, and will take into account the other natural sources of neutrinos. If neutrinos are found to oscillate then it means they have mass. This mass, no matter how small, will account for some percentage of the dark matter in the universe and will lead to even more questions about neutrinos.

PROBLEM: The beam of protons, produced from the main injector, needs to travel down this tunnel. The beam line needs to be constructed. The problem that I need to solve is how to get the 3Q120M quadrupole magnets into place in the tunnel. These quadrupole magnets are ones with two sets of north and south poles. Their purpose is to focus or defocus the beam of protons. Since the particles do not occupy the same path, the quadrupole magnets push the particles more toward the center if they are focusing, or move them away from the center if they are defocusing. Each magnet weighs 8,200 pounds. The device to be designed must be able to raise and lower the magnets onto and off its stands. The center of the magnet, also the center of the beam line, needs to be lifted up to a height of 2'7 ½". The tunnel has various dimensions that need to be considered when designing this device. The tunnel's minimum height is six feet tall. The width from the edge of the tunnel to the stands will accommodate a device that is forty inches wide. The alignment pads on the side of the magnet must not be damaged or bumped since they have to be very accurate to properly align the magnet. Finally, the magnets must be pulled up an incline that is about nine degrees steep. This paper describes the design process and gives an explanation of why the device which I have designed is adequate.

Design Process:

The design process began by familiarizing myself with the problem. Schematics of the tunnel and magnet to be transported were reviewed and helped me to visualize a solution to the problem. Attention was also given to existing methods of carrying and moving heavy loads. I enhanced my knowledge of the topic of "statics", and became familiar with the properties of steel, various types of connections, and other accessories available.

Once all the preliminary work was done, I then began to sketch some ideas of how to accomplish the task. I continually revised these ideas and modified them if something did not work out. There were three conceptual designs:

Design #1: Hydraulic push cylinders are used on the bottom to raise and lower the magnet. The problem with this design is the magnet can't rest on the hydraulic cylinders since the path of the magnet is on an incline. The device cannot be connected to the bottom of the magnet because it has to be raised onto the stands and the bottom of the magnet attaches to the stands. The coils at the ends of the magnets prevent any attachment to the end of the magnets. Finding a way to attach the magnet to the device is complicated, if there is any way to do so, so this idea is discarded.

Design #2: Hydraulic pull cylinders are used and attach to the top of the magnet. This will allow the magnet to be easily lowered onto the stands. This design is very simple, has very few moving parts, and will be easy to operate. Upon further analysis into this design, the height constraints play a big role. It turns out this design cannot be used, because the height of the device turns out to be greater than six feet, which is one of the constraints of the tunnel.

Design #3: The idea behind an engine lift is modified to contain two pivoting arms with two hydraulic cylinders. This design allows the largest range of motion for the magnet. This means that it will make moving the magnet into place as easy as possible. Further analysis into this design has been carried out.

The next task was to see what kind of stresses would be applied to this device. A stress analysis was done by hand on the different areas of the design. At the same time, various members were being sized based on the results of the calculations. Beam dimensions, column dimensions, hydraulic cylinder capacity, dimensions of frame, lug, weld, pivot and bearings, and wheel sizes were determined during this process. (See hand calculations) They were determined with adequate safety factors to ensure the device will hold up to the various stresses it will take. While doing the calculations, revisions to the design were continuously made. Once the calculations were complete, the design could be sketched to scale.

Due to the fact most spots on the model have more than one stress acting on it, it is very difficult to hand calculate an accurate stress. To double check the work that was just done, many models were made using the IDEAS CAD software. The model in mind was drawn and various stress and displacement analyses were used. Along the way, various problems with the design were encountered. Sometimes multiple solutions arose to these problems and were further looked into to find the best one. IDEAS uses the finite element method to perform stress and displacement tests. This requires that nodes be placed throughout the model. For a simulation to be as accurate as possible, it is important the nodes are placed strategically on the model. Before moving on, the results should be consistent. Different mesh types and different element lengths have to produce similar results.

The modified engine lift has many advantages. The biggest advantage is this model allows the magnet to have the biggest range of motion. For instance, a cylinder with 7" of movement, moves the magnet a total of 11 ½" vertically. This has a couple of advantages. First off, the magnet can be carried even lower in this design than the other two. This has the effect of lowering the center of gravity, making it more stable and harder to tip the device while transporting. Also, it will be easier to place the magnet where it needs to go. Since the dimensions in the beam tunnel vary, a large range of motion make the device more versatile. Another advantage is the ease at which the magnet attaches to the device. The device will slide around the magnet from the side. A couple of bolts will be used to attach the magnet to the device. The magnet can then be moved into place.

Operation of Proposed Device:

This device is designed to be as easy to operate as possible. First, the magnet is attached to the device. A total of four bolts are screwed into the magnet. Once this is done, the magnet is lowered as far as the device allows. This ensures the device is as stable as possible when it is being transported. A winch will be used to pull the magnet up hill. Once the device is in position to place the magnet, it will have to slide sideways into place. First, the magnet is raised up over the stands. It is then pushed or pulled sideways to place the magnet on its stands. Next, it will be lowered onto its stands. Finally, it is disconnected from the magnet lift and attached to the stands.

Sizings and dimensions:

- 2 - 5-ton capacity Hillman rollers NT Series w/ turntable 5-NT with ¾" thick shim and 4 - 3.75-ton capacity Hilman Rollers FT series swivel padded 3.75-SP
- Beam Dimensions: 6" x 4" x ½" A500 Structural Steel Tube
- Column Dimensions: 4" x 4" x ½" A500 Structural Steel Tube
- 2 - Hydraulic Cylinder: Enerpac General acting single purpose cylinder model #RC-57 (5-ton capacity)
- Length of Frame: 120"
- Height of Frame (pivoting arm parallel to ground): 53"
- Width of Frame: 40"
- Length of Pivoting Arm: 30.5"
- Diameter of Pivoting Pin: ½" (Appendix 7.5 and 7.6)
- Lifting Lug: 4 total 6" long x 3" tall x 1" thick, with 2-1"x4" Lug Stiffeners
- Lifting Bolts: A307 Steel ½" Bolts (Appendix 7.11)
- Weld Sizing: Lifting Lugs – 5/16" fillet weld Other Connections – full penetration welds E70XX electrodes for both types
- Cross Bracing Length: 65.1"
- 4 – 5/8" stock# galvanized steel shackle with work load limit of 10,000 pounds
- Sling sizing: 23,800 pound breaking strength plain steel wire rope

See Appendix 7 for justification of dimensions

Wheel Selection:

Two variations of a modified engine lift were made. One model had 4 Hilman Rollers. (See Appendix 2.1) The other model has six Hilman Rollers. (See Appendix 2.2) Stress analyses were done to further investigate these two variations. (See Table 1) It was found that the maximum stresses in the 6-wheel design were about half as great as 4-wheel design. Also, the deflection in the 6-wheel design was about an order of 10 times smaller than the 4-wheel design. (See Appendix 3) Six wheels were chosen because it will be more durable and adds strength to the frame of the device.

Design	Max Stress	Max Displacement
4-Wheel (flat)	7430 p.s.i.	0.0373"
6-Wheel (flat)	3090 p.s.i.	0.00652"
4-Wheel (inclined)	7430 p.s.i.	0.0359"
6-Wheel (inclined)	3690 p.s.i.	0.006"

Table 1: Shows the maximum stresses and displacements Encountered by the different models. Results were obtained from the Ideas CAD software and are approximates and not exact solutions.

The loads that the wheel would have to support were calculated. The maximum load found for the design would be from the cylinder. It is just over 4 tons, so 2 - 5-ton capacity NT series w/ turntables will be placed under the hydraulic cylinders. 4 - 3.7-ton capacity FT series swivel padded rollers will be used on the other four places for wheels. Since the dimensions of the products are different, a ¾" shim will need to be placed on the wheels under the hydraulic cylinders. The wheels were selected based on these loads, safety factors, and available products. As mentioned, the wheels have to rotate to allow the movement in two degrees of freedom. The magnet has to move laterally as well as the long way of the tunnel.

Beam and Column Selection:

Hand calculations were done to determine the sizes of the beams and columns. Beams were sized based on the highest single stress they would encounter. Bending moments on the pivoting arm were calculated and were sized off that. (See Appendix 7.4) Torsional stresses were calculated on the main beam, and sized off that. (See Appendix 7.9 and 7.10) Tensional stresses in the main column were used when determining the size of the column. An appropriate safety factor was used when determining the sizes of the beams and columns. All the beams will be 6" x 4" x ½" thick A500 structural steel tube. This is the smallest size that will hold up to all the various stresses. Other members that can have small stresses applied to it use this size for uniformity and it is easier to connect them together. The columns will be the size of 6" x 4" x ½" thick A500 structural steel tube. This size will remain undamaged by the loads and will attach nicely to the various beams.

Hydraulic Cylinder Selection:

There are a couple of variables that need to be known when determining the model of the hydraulic cylinders. The load on the cylinder and the stroke of the cylinder need to be determined. The stroke of the cylinder refers to how much the cylinder moves up and down. The highest calculated load on the cylinder was calculated to be 8,435 pounds. (Appendix 7.1-7.3) A 5-ton capacity cylinder will suffice. The magnet needs to be carried fairly low to the ground to avoid tipping. A stroke of 7" will provide a total of over 11" of vertical movement of the magnet. (See Appendix 7.7) This allows the magnet to be low to the ground and also allows a big range of motion. This specific cylinder is the Enerpac RC series, Single-Acting Cylinder model # RC-57. Two of these cylinders will adequately raise and lower the magnet.

Dimensions of Frame:

The length of the frame was chosen to be 120". The reason for this is the length of the magnets to be transported is 120". The height of the frame, including the wheels, is 53". This allows the magnet to fit the height constraint when the magnet is fully raised. (See Appendix 5 for dimensions) The width of the frame is going to be 40". This is the maximum allowable width of the device. It is chosen because this allows the wheels to be out as far as possible. This moves center of gravity of the device safely within the support base of the wheels. (See Appendix 7.8) The pivoting arm dimensions were chosen for a couple of reasons. The distance from the pivot to the center of the hydraulic cylinder should be as large as possible to minimize the load on the cylinder. The distance from the cylinder to the magnet needs to be as small as possible. The proposed distance from the cylinder to the magnet is 12.25". This is the smallest distance that allows the proper clearance of the magnet when it is moved. The distance from the pivot to the cylinder is 15.25". (See Appendix 1) This is the greatest distance that moves the center of gravity to fall safely within the support base.

Lug Sizing:

There will be a total of four lifting lugs on this device. Two of them will attach to the pivoting arm. The other two will be attached to the plate that attaches to the magnets. The maximum load on the lifting lug will be 4,677.64 pounds. The lug will be 1" thick, 6" long, and 3" tall. The diameter of the hole in the lifting lugs will be 1". This allows a shackle with adequate strength to be attached to the lifting lug. (See Appendix 7.12) These dimensions also reduce the stress to an allowable amount. One problem is the pivoting beam cannot support the lug. So, lug stiffeners were added to the pivoting arm - lug connection. They are 1" x 1" x 4" blocks of steel. These added stiffeners reduce the stress on the beam to a safe level. (See Appendix 7.13)

Weld Sizings:

Various spots on this design need to be welded together. Spots on the frame include the main column to the bottom frame, all the pieces the wheels go on, and the main beam to the hydraulic cylinder beam. E70XX electrodes are to be used for all joints. All these welds will be full penetration groove welds. Since these welds are actually stronger than the metal of the frame, no calculations are necessary. The lifting lug needs to be welded to the pivoting beam at the top, and the plate that attaches to the magnet at the bottom. Since the thickness of the thicker piece being joined together is 1", the minimum weld size for this is 5/16". Calculations were done with this weld size, and it was found that 5/16" fillet weld would be fine for the lifting lugs. (See Appendix 7.14) The welds will be 6" long on each side, but a minimum length of 1.25" will suffice. The welds will adequately attach the pieces of the frame together and hold up to any stresses that will be applied to the device.

Shackle and Sling Sizing:

Four shackles are needed for this device, two to attach to the top lifting lug and two to attach to the bottom lifting lug. The maximum load the shackle will have to sustain is 4,677 pounds. A galvanized alloy steel 5/8" diameter shackle will do the job. The steel rope connecting the two shackles have to also support the 4,677 pound load. A safety factor of 5 times is recommended. A steel rope that has a breaking strength of 23,800 pounds will do the job. The particular rope is a plain steel wire rope with a 6 x 19 class compacted IWRC core. The diameter is 7/16".

Problems and Solutions:

There was concern about the stresses on the pivoting arm and the main column. So, whether or not a support was needed was investigated. (See Appendix 4) Both designs were modeled using IDEAS, and a shell mesh was applied to both of them. They both had the same loads on the incline applied to them and the results were analyzed. Deflections varied by only two thousandths of an inch, with the supported design deflecting less. The stresses on the other hand varied by a factor of two. The no-support design more evenly distributed the load. Refer to table 2 for the results. It was decided to recommend the design with no supports because the stresses are much lower and more spread out. Since the deflections were very similar, there is no reason to go with the higher, more concentrated stresses.

Design	Max Stress	Max Displacement
No Support	3620 p.s.i.	0.0497 "
Support	7880 p.s.i.	0.0471 "

Table 2: This table shows the results of modeling a design with and without supports using load conditions representing an inclined state.

The design has a problem. When the lifting device is on an incline, there is an additional shearing stress added to all the members. This is not a problem for the beams and columns, but it is a big problem for the hydraulic cylinders. Hydraulic cylinders are very weak when it comes to shear stresses. This problem has to be addressed and taken care of. One way to do it is to attach a rope that can withstand a load of about 1,000 pounds to the pivoting arm that leads the way up the incline. The other end of the rope is attached to the bottom lifting lug on the low end of the device. (See Appendix 6) The same wire rope will be used for the cross bracing as for the sling. The length needed for the cross bracing has been calculated to be about 65". This allows the shearing load on the hydraulic cylinders to be reduced to nothing. There is an additional torsional stress on the front main column, but the tradeoff is acceptable. (See Appendix 7.15 and 7.16)

Conclusion:

This proposed device has been thoroughly investigated. All the dimensions and specifications have been chosen with care and adequate safety factors. A design that allows easy and reliable operation has been chosen. All the constraints of this design task have been met. If the recommended sizes of various members are used along with the recommended dimensions of the device are used, then there should be no problems moving the 3Q120M magnets into the NuMI tunnel.

References

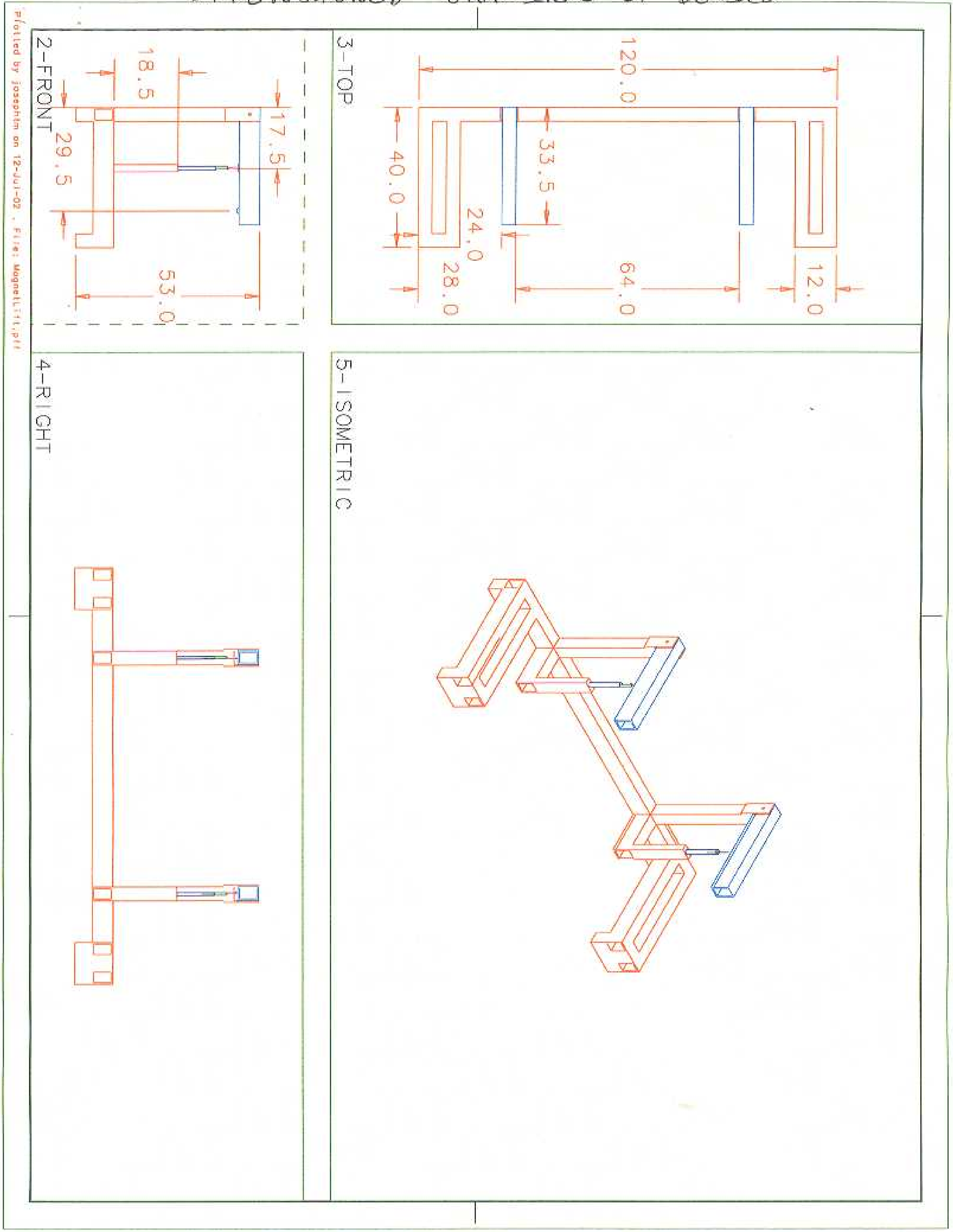
Gordon, J.E. Structures: Or Why Things Don't Fall Down. Penguin Books, 1978

AISC Manual of Steel Construction 9^{th} Edition. A.I.S.C., 1989

Blodgett, Omer W. Design of Welded Structures. The James F. Lincoln Arc Wleding Foundation, 1966

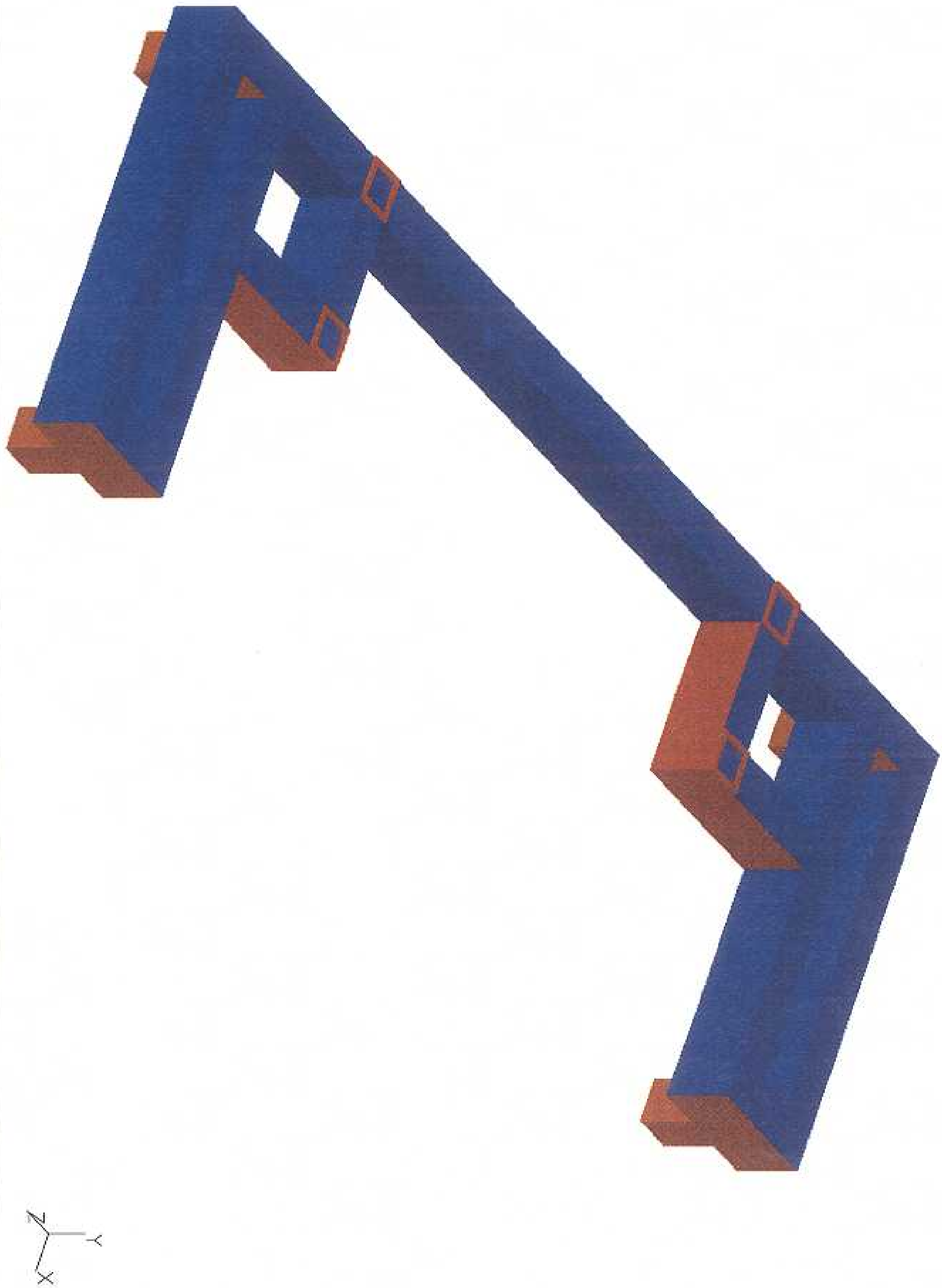
APPENDICES

DIMENSIONED DRAWING OF DEVICE

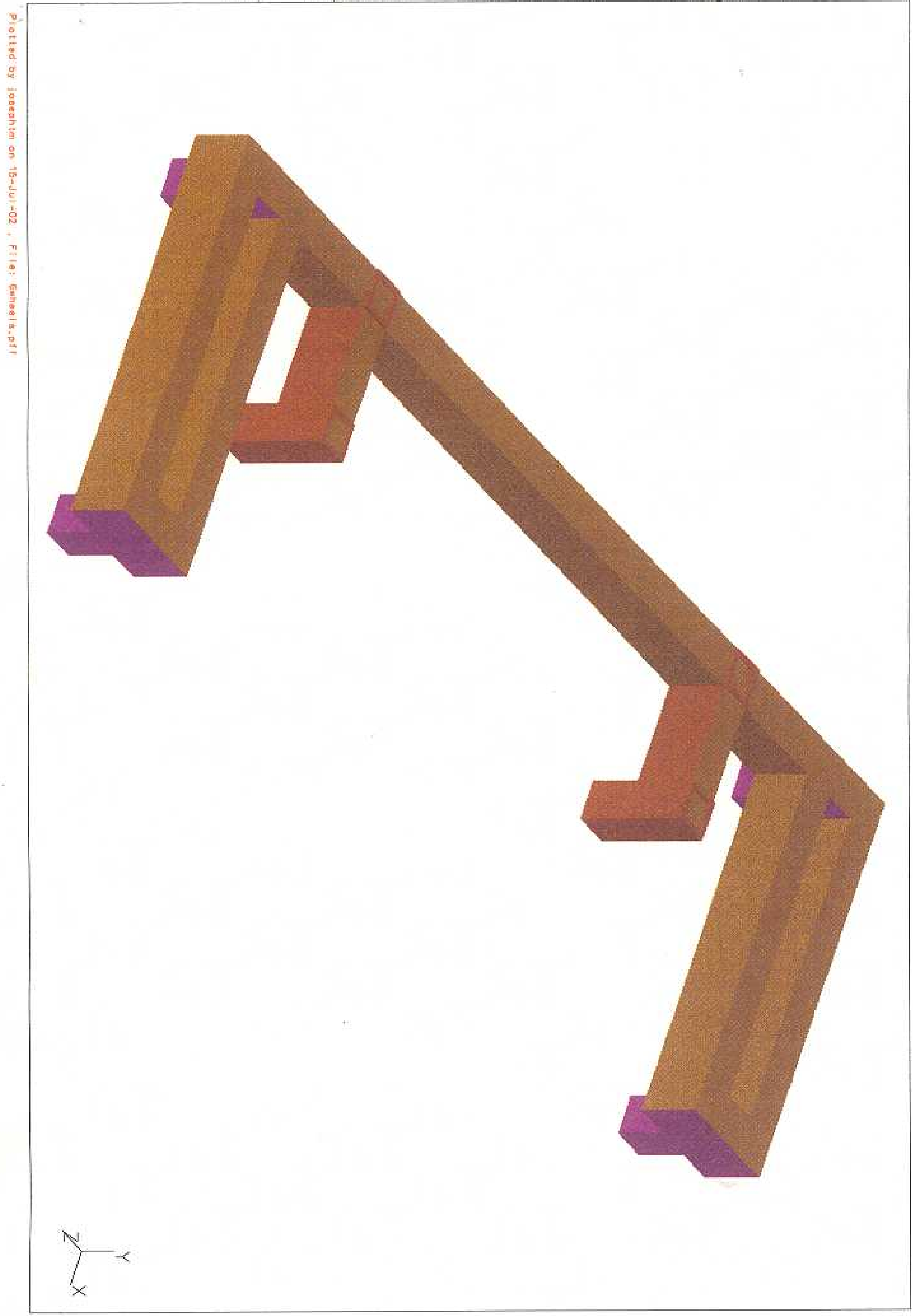


FOUR WHEEL DESIGN SKETCH

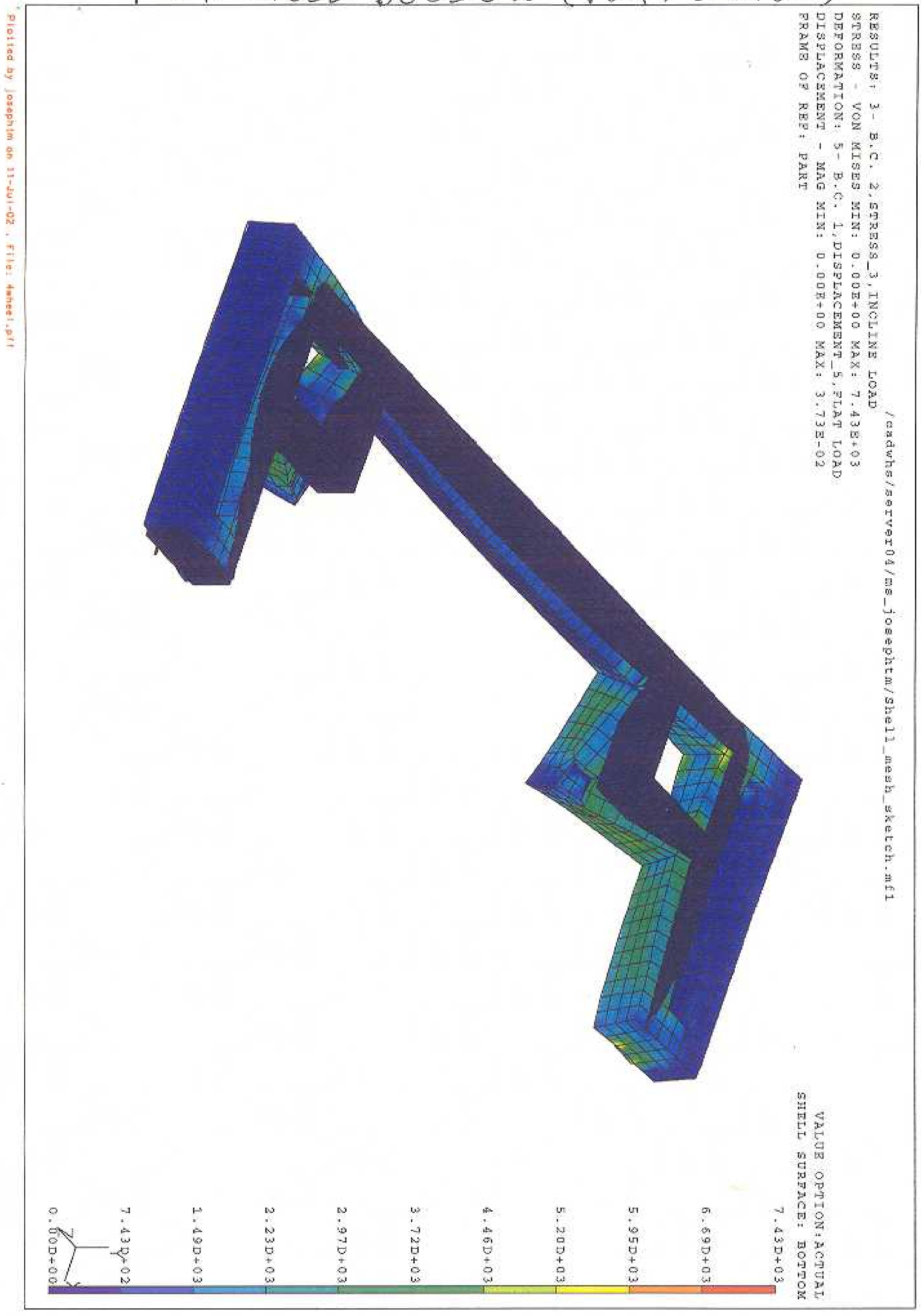
Plotted by josephm on 15-Jul-02 , File: 4wheels.prt



SIX WHEEL DESIGN SKETCH

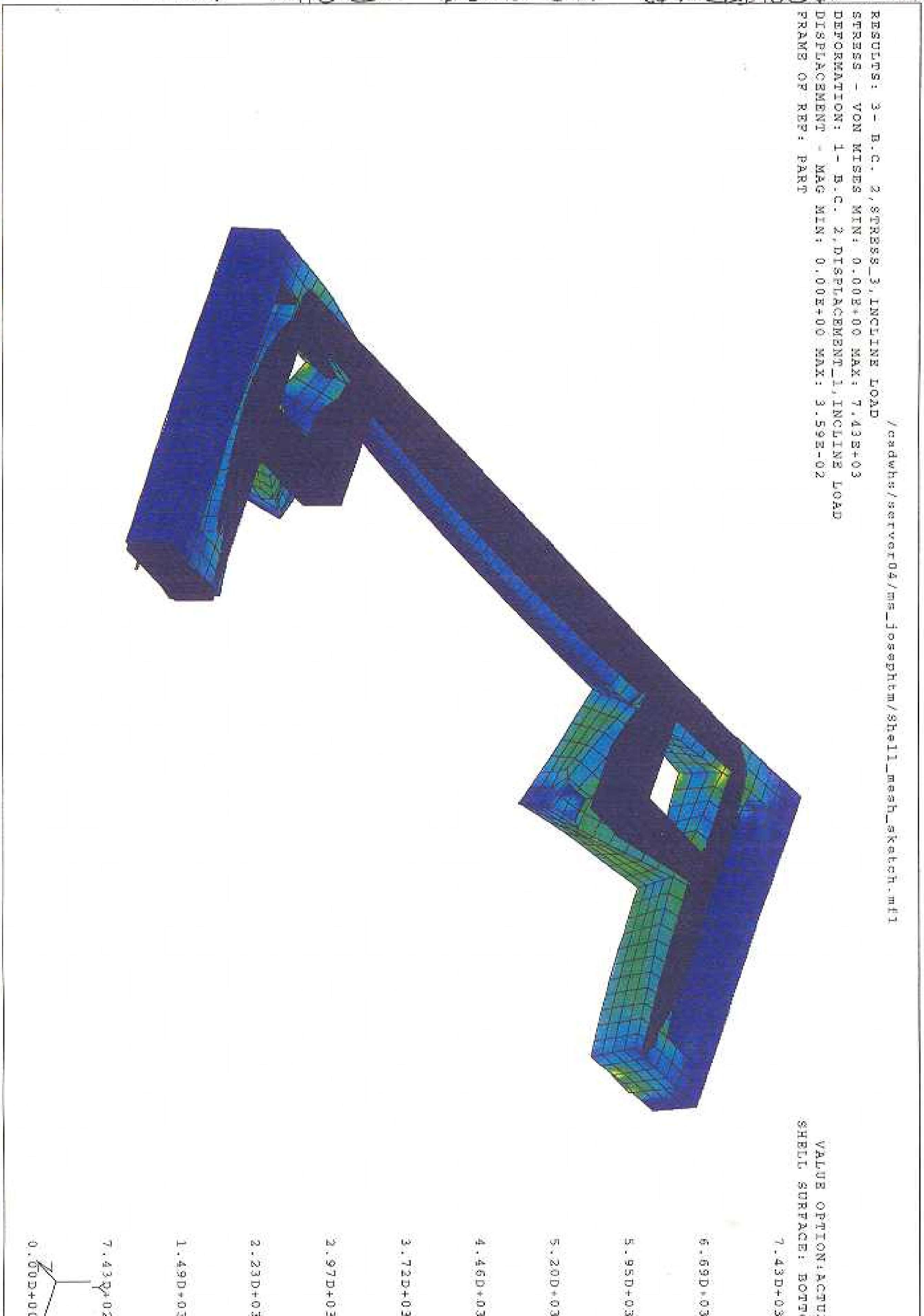


FOUR WHEEL DESIGN (FLAT LOADS)



A-3.1

FOUR WHEEL DESIGN (INCLINED LOADS)



Plotted by josephm on 11-Jul-02 , File: 4wheelinc.plt

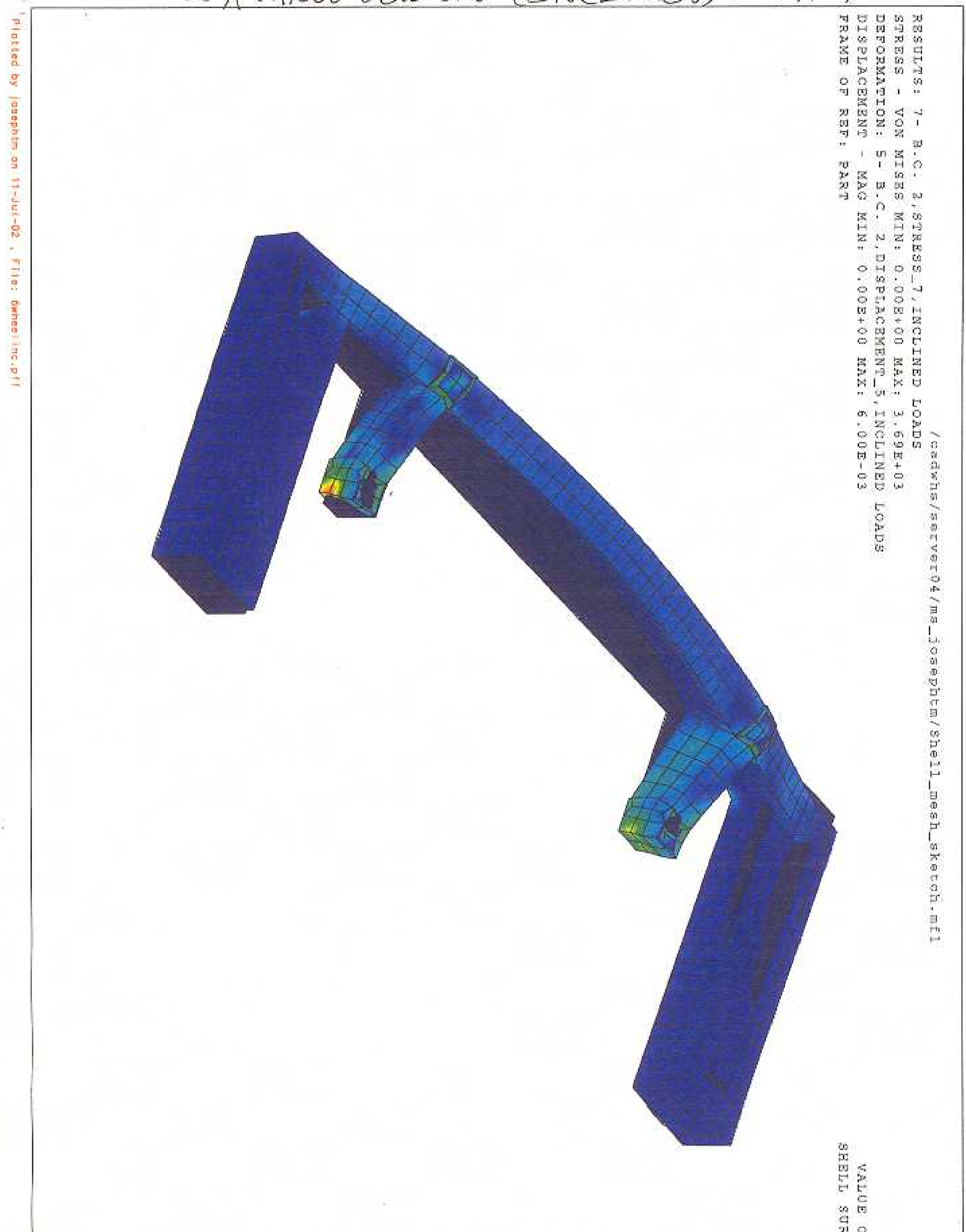
A-3.2

SIX WHEEL DESIGN (FLAT LOADS)

Plotted by josephtm on 11-Jul-02 , File: 6wheel.plt

A-3.3

SIX WHEEL DESIGN (INCLINED) LOADS)

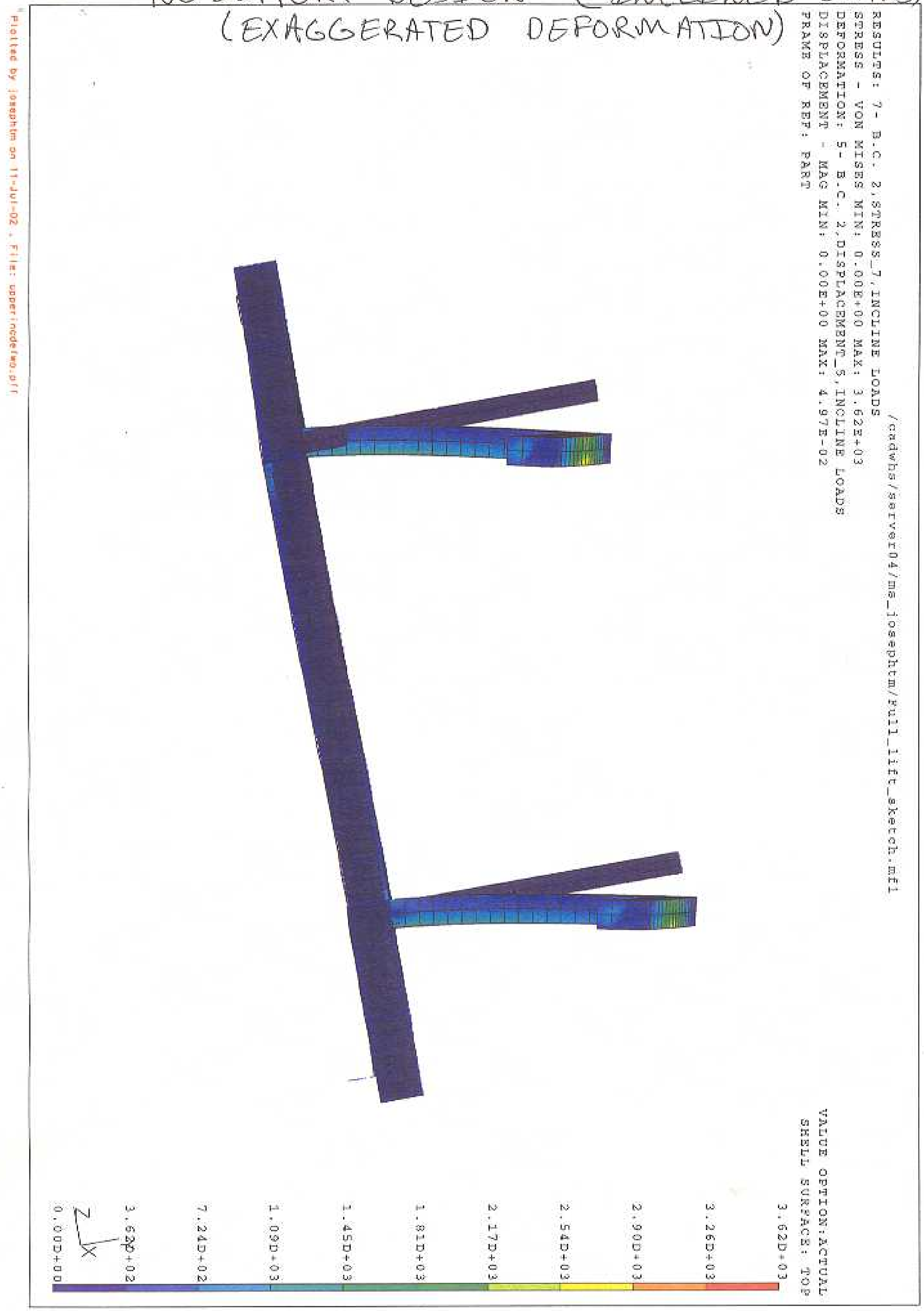


A 3.4

NO SUPPORT DESIGN (INCLINED LOADS)

Plotted by josephtm on 11-Jul-02 , File: upperinc.pf1

NO SUPPORT DESIGN (INCLINED LOADS)
(EXAGGERATED DEFORMATION)



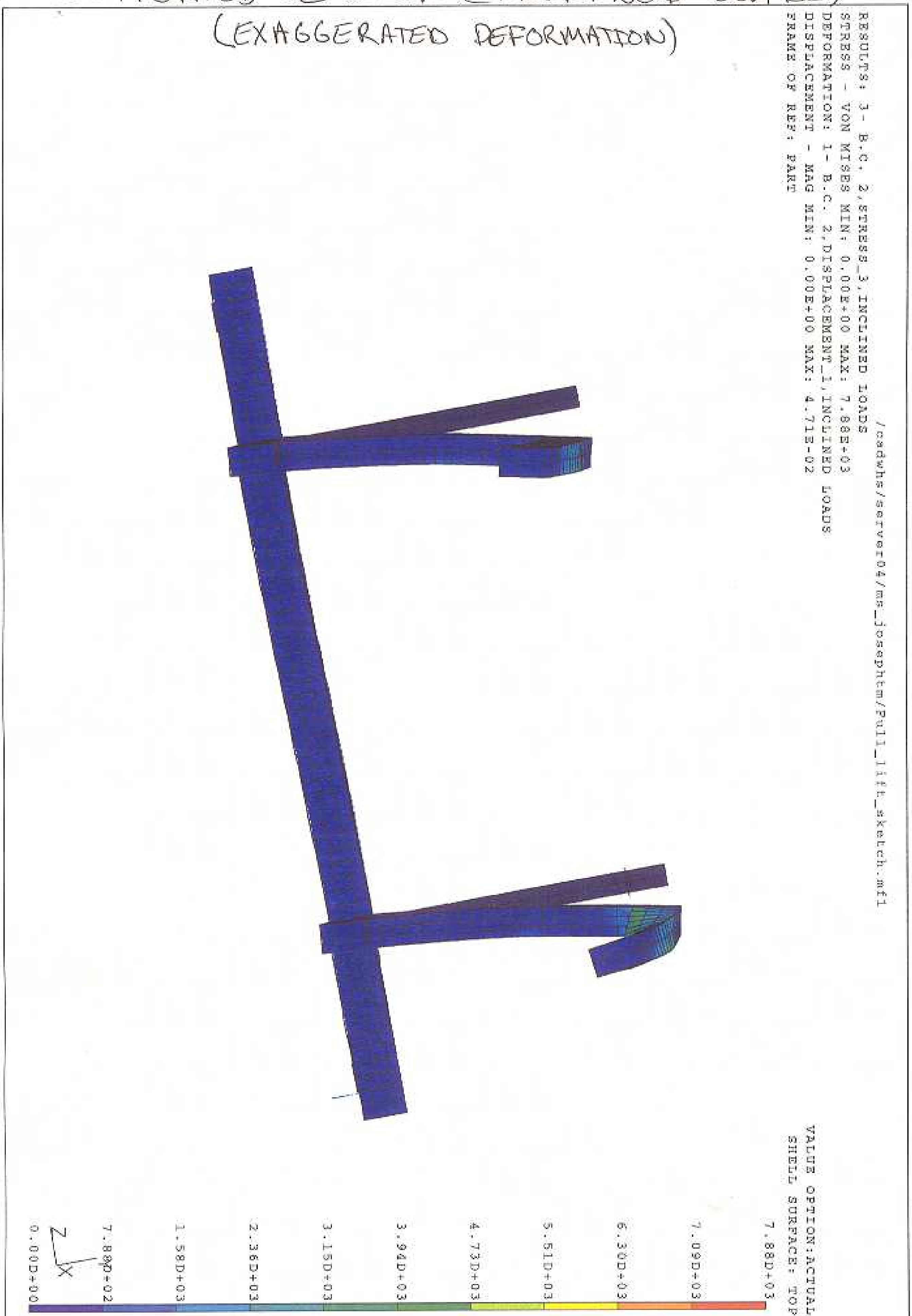
A 4.2

SUPPORTED DESIGN (INCLINED LOADS)

Plotted by josephtm on 11-Jul-02, File: upperincwo.plt

SUPPORTED DESIGN (INCLINED LOADS)

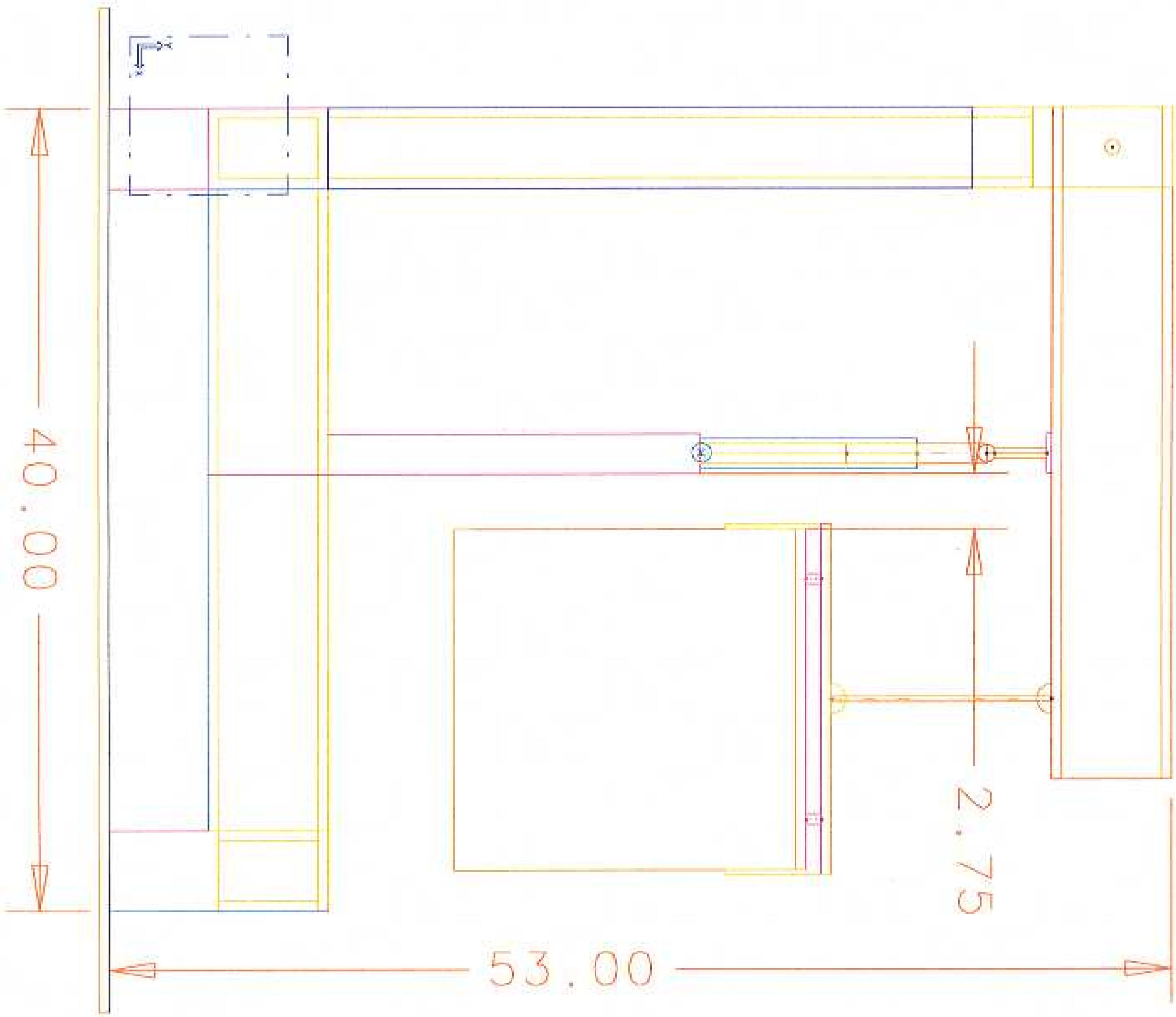
(EXAGGERATED DEFORMATION)



Plotted by josephtm on 11-Jul-02 , File: upperincdefmo.plt

A-4.4

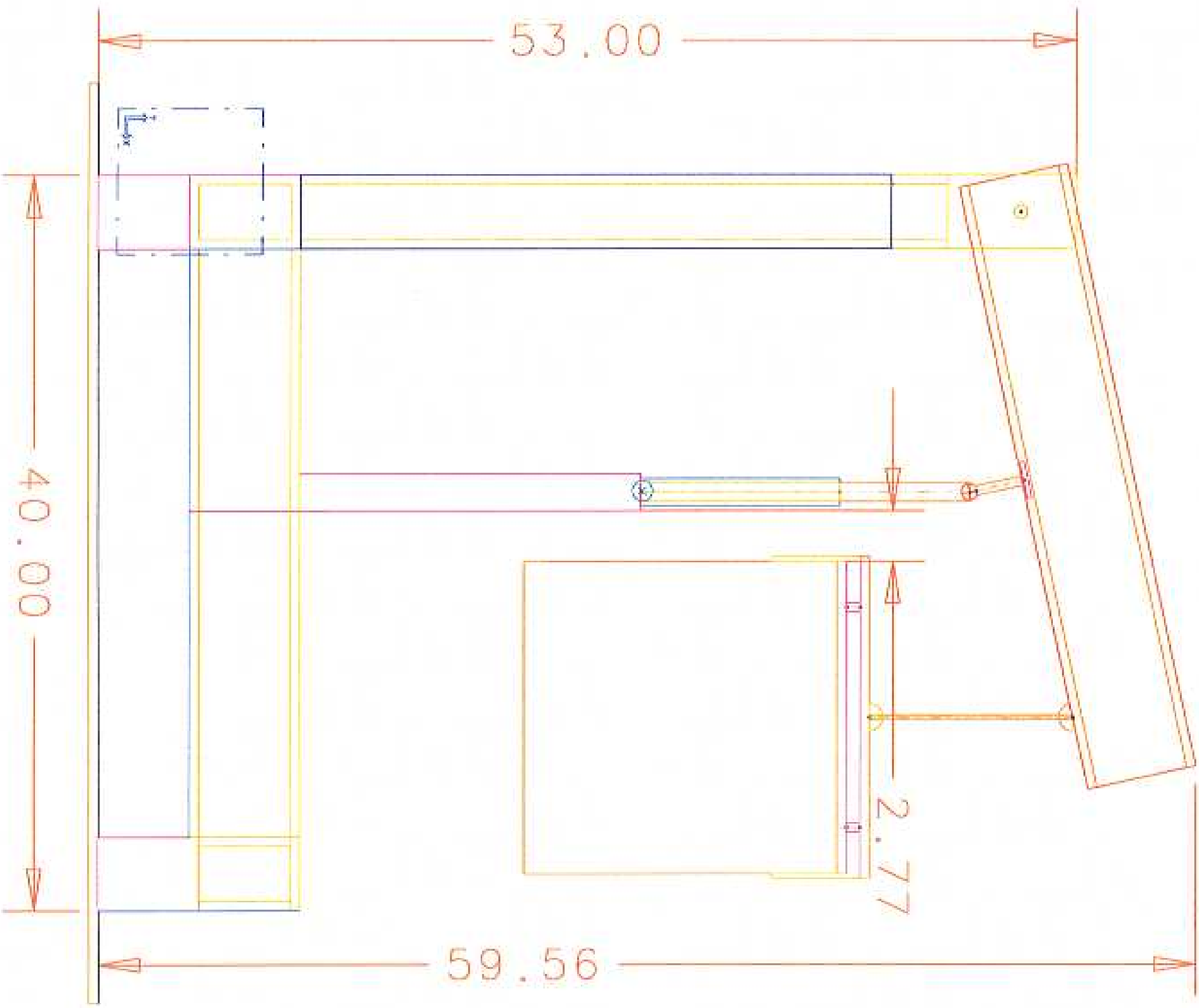
DIMENSIONS OF LIFT (LEVEL)

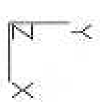


A-5.1

DIMENSIONS OF LIFT (ELEVATED)

Plotted by josephim on 10-Jul-02 , file: LiftElevated.plt

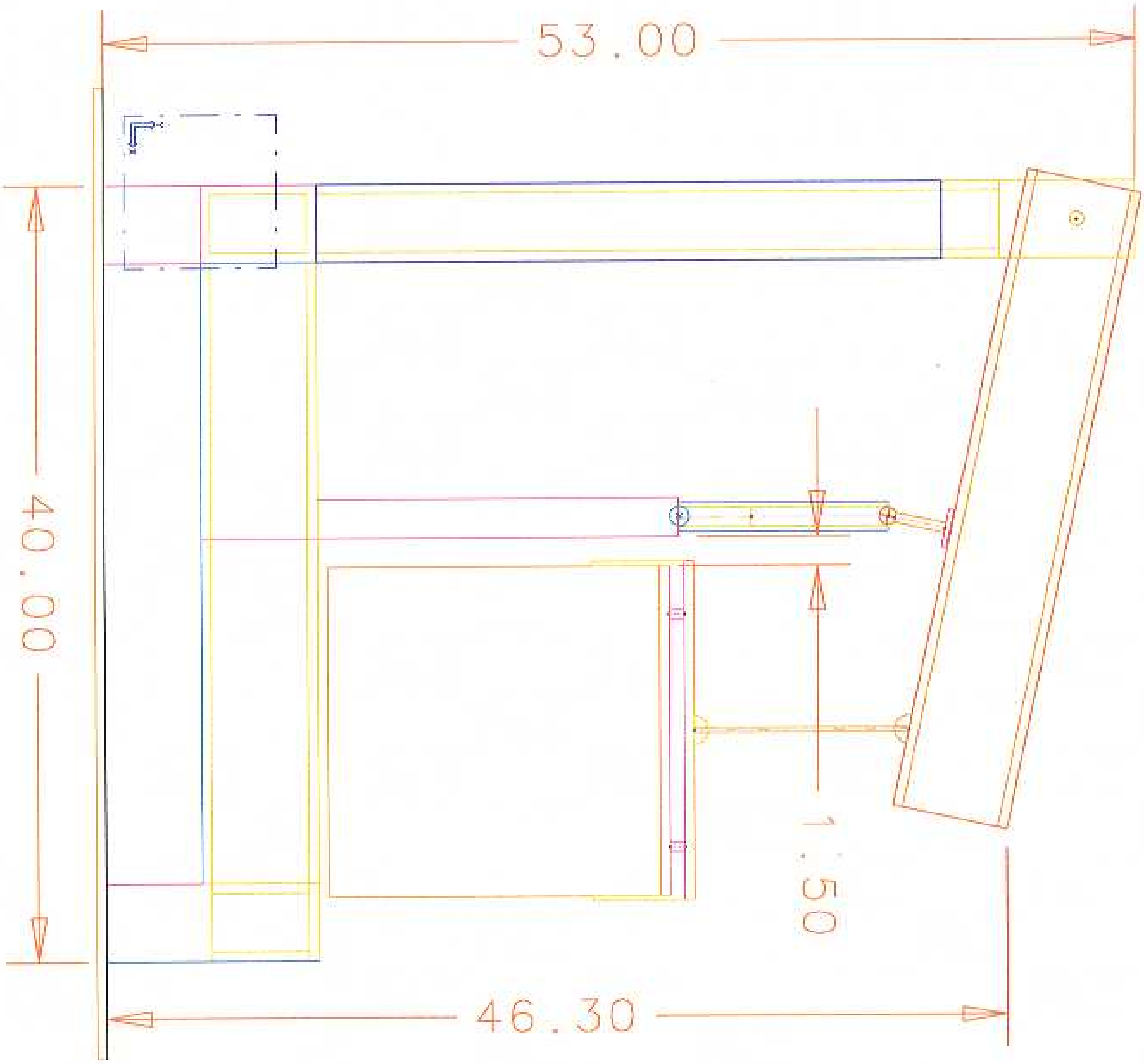




A-5.7

DIMENSIONS OF LIFT (LOWERED)

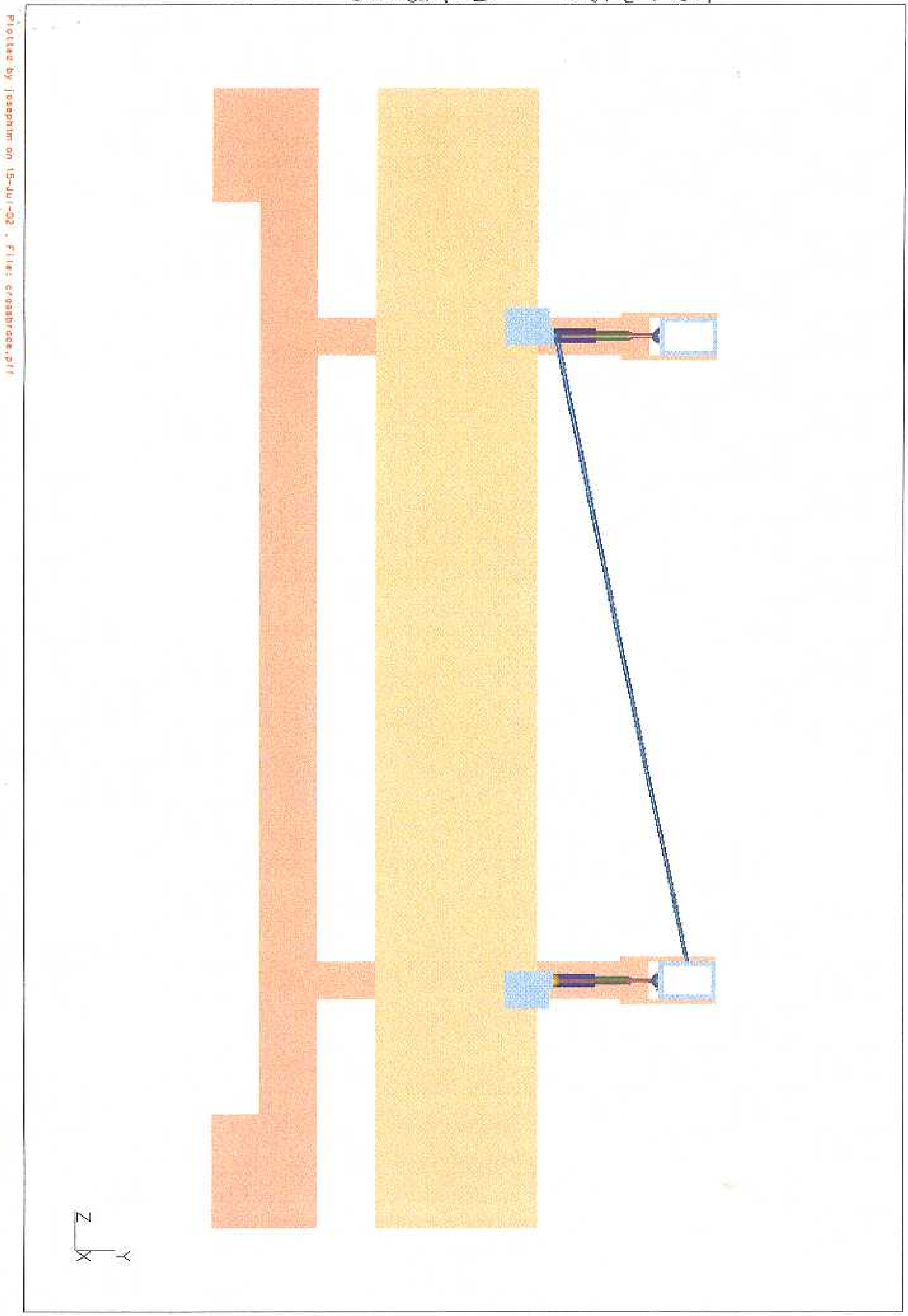
Plotted by josephtm on 10-Jul-02 , File: LiftLowered.prt



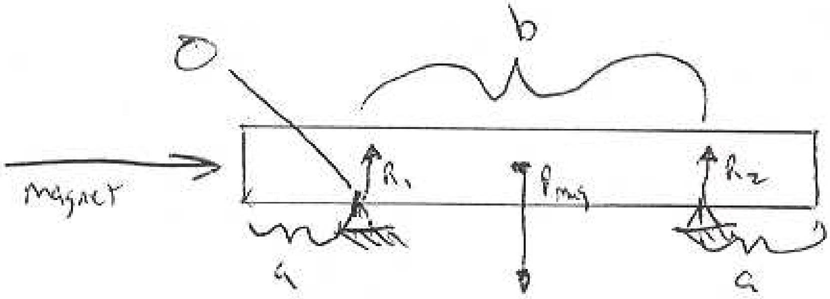
Z Y X

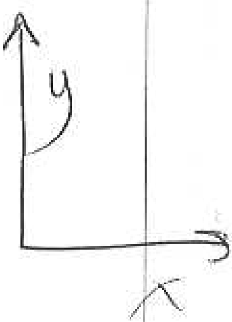
A-5.3

GROSS BRACING SKETCH



Flat Surface





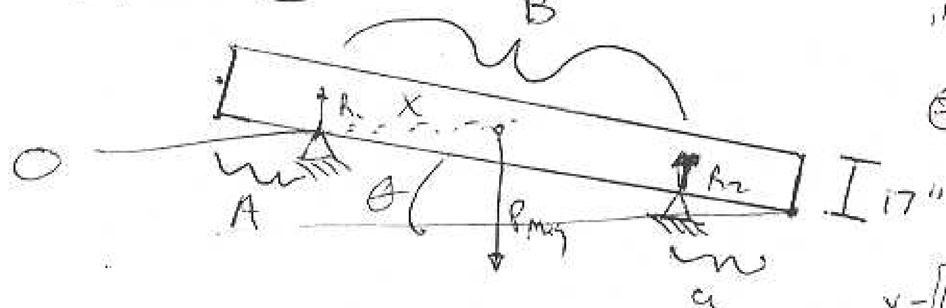
$$\Sigma F_y = 0 = R_1 + R_2 - P_{mag}$$

$$\Sigma M_o = -\frac{B}{2} \cdot P_{mag} + R_2 \cdot b = 0$$

sum of forces and moment equal zero because in equilibrium.

$$R_1 = \frac{P_{mag}}{2} = R_2 \qquad R_1 = R_2 = 4500\#$$

On Incline



$\theta = 8.97292°$

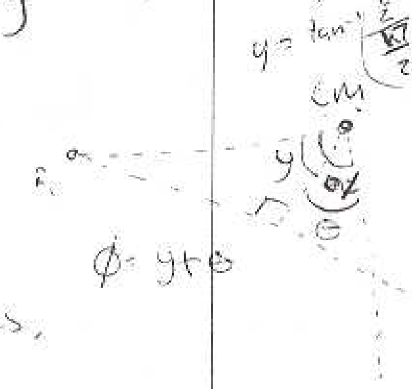
$$x = \sqrt{\left(\frac{B}{2}\right)^2 + \left(\frac{17}{2}\right)^2}$$

$$\Sigma F_y = 0 = R_1 + R_2 - P_{mag}$$

$$\Sigma M_o = 0 = B \cdot R_2 \cdot \sin(90 - \theta) - X \cdot P_{mag} \cdot \sin(\theta + \tan^{-1}\tfrac{b}{17})$$

$$R_2 = \frac{X \cdot mg \cdot \sin(\theta + \tan^{-1}\frac{b}{17})}{b \cos\theta}$$

$$R_1 = mg - \frac{X \cdot mg \cdot \sin(\theta + \tan^{-1}\frac{b}{17})}{b \cos\theta}$$

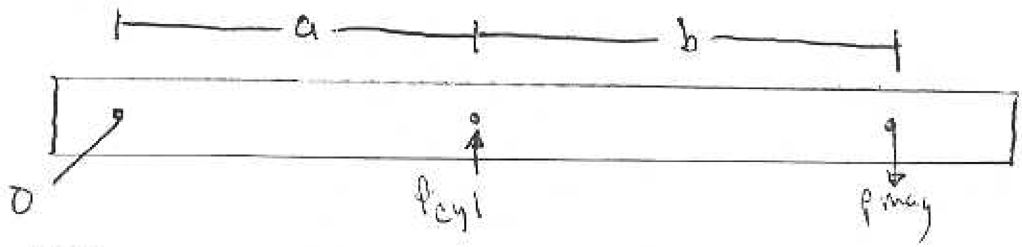


$\phi = y + \theta$

if b = ~~[illegible]~~ 68" because that's where the magnet attaches.

$\boxed{R_2 = 4677.64\#}$

$\boxed{R_1 = 4322.36\#}$



"B" must be at least 8.5" (half magnet width) + .75" (half hydrau width + clearance (2")

$$B \geq 11.25"$$

$$\circlearrowright^{+} \sum M_o = 0 = a \cdot P_{cyl} - (a+b) \cdot P_{mag} \qquad P_{mag} = 4677.64 \#$$

$$P_{cyl} = \frac{(a+b) \cdot 4677.64}{a}$$

As "a" increases, P_{cyl} decreases

As "b" decreases, P_{cyl} decreases

So, we want "b" as small as possible and "a" as large as possible

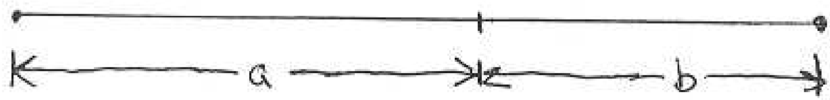
Constraint $a + b +$ clearance $\leq 40"$ (width of passage way

a = 15.25"

b = 12.25"

clearance = 3" (from end to pivot) + 8.5" (width of magnet) + .75" other

FLAT



$a = 15.25$
$b = 12.25$

$$P_{cyl} = \frac{(a+b) \cdot P_{mag}}{a} = \frac{(12.25 + 15.25) \cdot 4500}{15.25} = \boxed{8114.75 \text{ \#}}$$

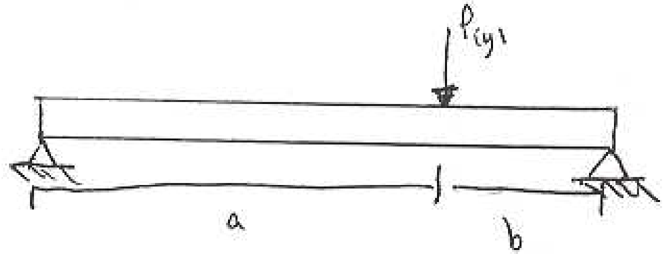
INCLINED

$P_{mag} = 4677.64 \; , \quad 4322.36$

$$P_{cyl} = \frac{27.5}{15.25} \cdot 4677.64 = 8435.09 \text{ \#}$$

$$P_{cyl} = \frac{27.5}{15.25} \cdot 4322.36 = 7794.41 \text{ \#}$$

Flat



$a = 15.25$
$b = 12.25$
$P_{cy1} = 8114.75$

$$\sigma = \frac{M_{max} \cdot h}{I}$$

M: moment on the Beam
h: height of center of gravity
I: moment of inertia

$$M_{max} = \frac{P \cdot a \cdot b}{(a+b)} = \frac{8114.75 \cdot 15.25 \cdot 12.25}{27.5} = 56028.66 \text{ in\#}$$

$I = 35.3 \text{ in}^4$ – $6'' \times 4'' \times \frac{1}{2}''$ steel structural tube
$h = 3''$

$$\sigma = \frac{56028.66 \cdot 3}{35.3} = 4761.64 \text{ psi}$$

Max deflection

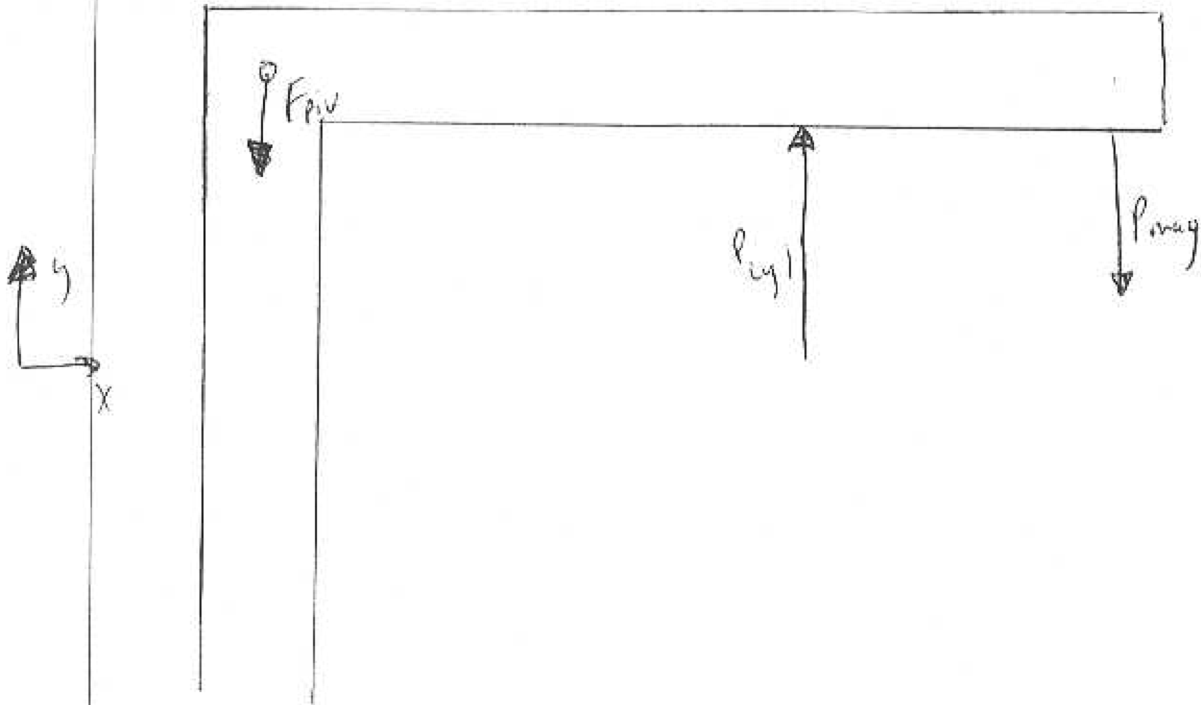
$$\frac{P_{cy1} \cdot a \cdot b \cdot (a+2b) \cdot \sqrt{3 \cdot a \cdot (a+2b)}}{27 \cdot E \cdot I \cdot l}$$

$E = 29{,}000{,}000$ psi. = Yield Strength
l: length of beam

$= .6034''$

These formulas were obtained from the ASD 9th Edition

FLAT



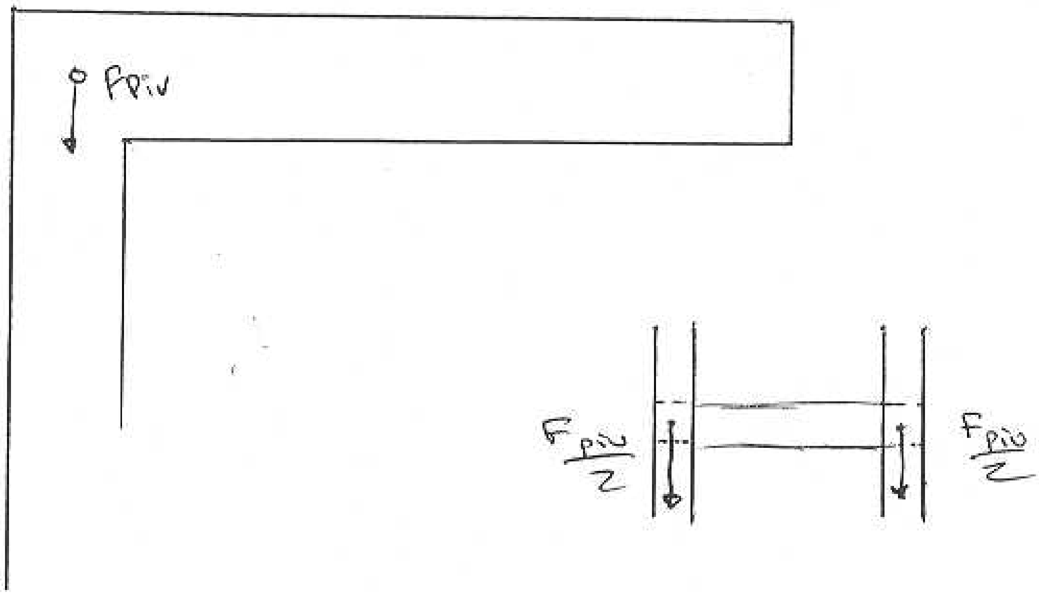
$$\Sigma F_y = 0 = P_{mag} + F_{piv} - P_{cyl}$$

$$F_{piv} = P_{cyl} - P_{mag}$$

$$= 3614.75\ \#$$

Joe Martinez 07/15/02 | Calc. for diameter | of pin size.

FLAT



Shearing Stress on Pin = τ

$\tau = \frac{F}{A}$ $\quad A = \frac{\pi \cdot D^2}{4}$

$\tau \leq 10000$ psi

$10000 \geq \dfrac{3614.75}{\frac{2 \cdot \pi \cdot D^2}{4}}$ $\quad D \geq .4797''$

$D = .5''$

INCLINE

$F_{piv} = 3735.45$

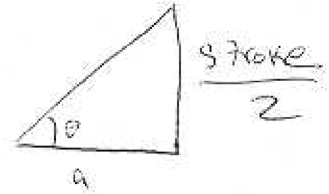
$10000 \geq \dfrac{3735.45}{\frac{2\pi \cdot D^2}{4}}$ $\quad D \geq .4891$

$D = .5''$

FLAT

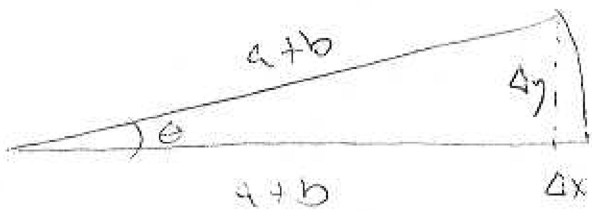
Stroke = 7"

RC-57 model



$$\theta = \tan^{-1}\left(\frac{3.5}{16.25}\right) = 12.1549^\circ$$

$$\theta = \tan^{-1} \frac{\frac{\text{stroke}}{2}}{a}$$



$\Delta y = \sin\theta \cdot (a+b)$

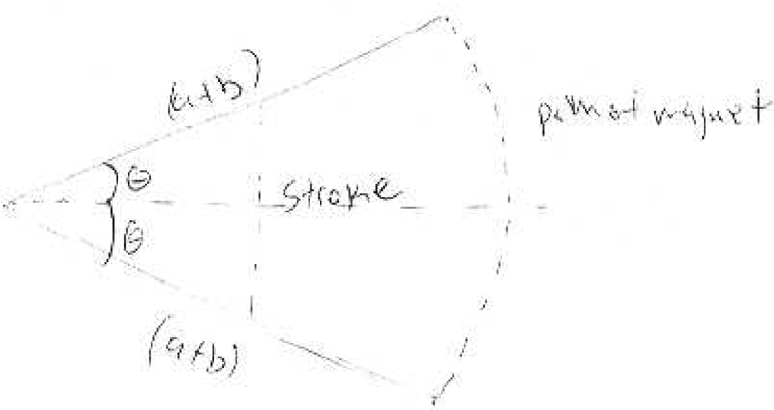
$\Delta x = (a+b) - \cos\theta \cdot (a+b)$

~~$\Delta y = \sin\theta \cdot$~~

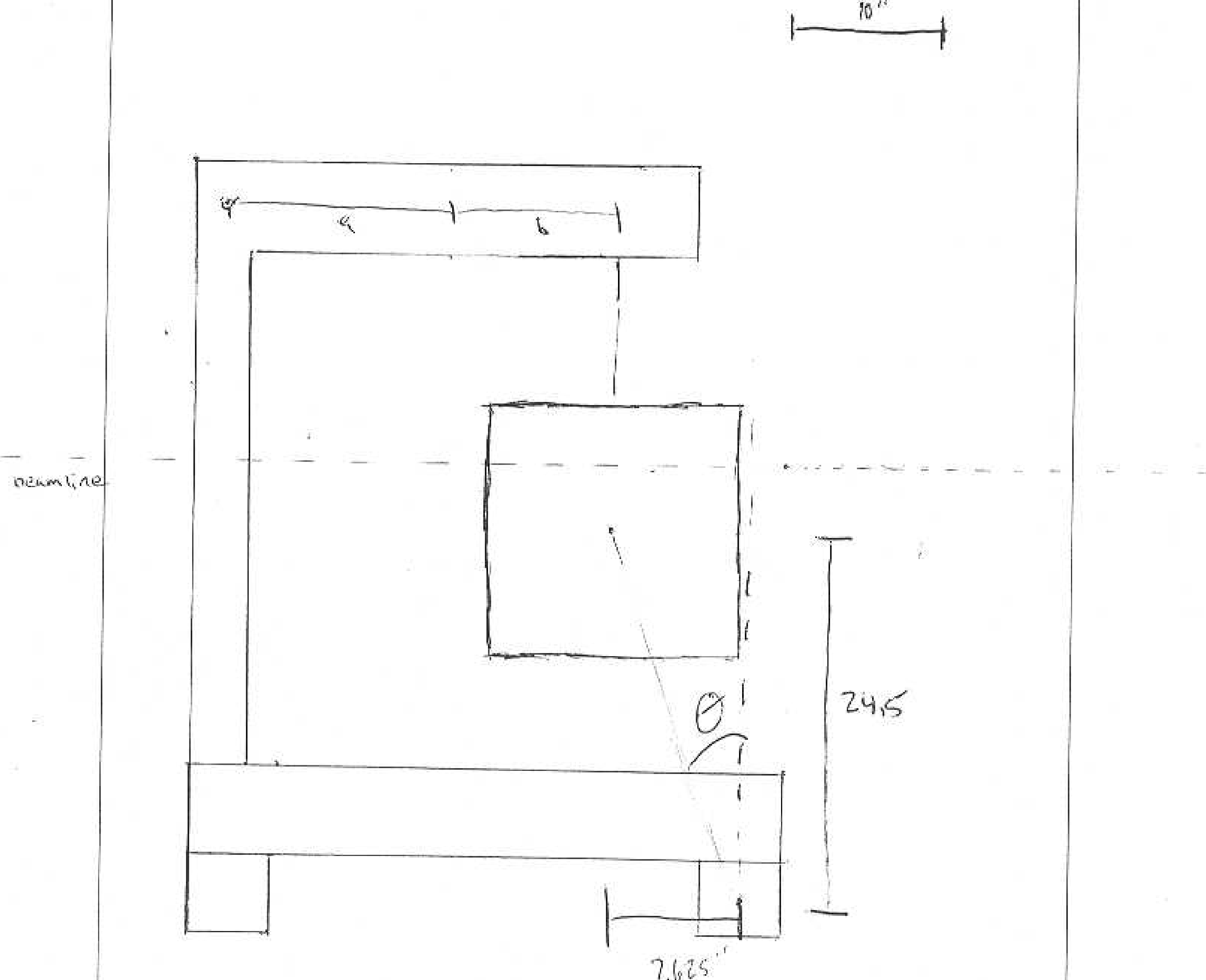
$\Delta y = \sin 12.1549 \cdot (27.5)$

$\Delta x = 27.5 - \cos 12.1549 \cdot 27.5$

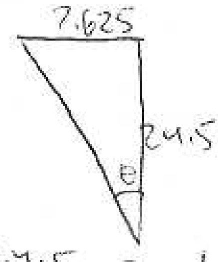
$\boxed{\Delta y = \quad 5.790292" \quad \Delta x = .62"}$



Joe Martinez 7/16/02 Tipping Angle & energy to tip (side view)



Tipping angle



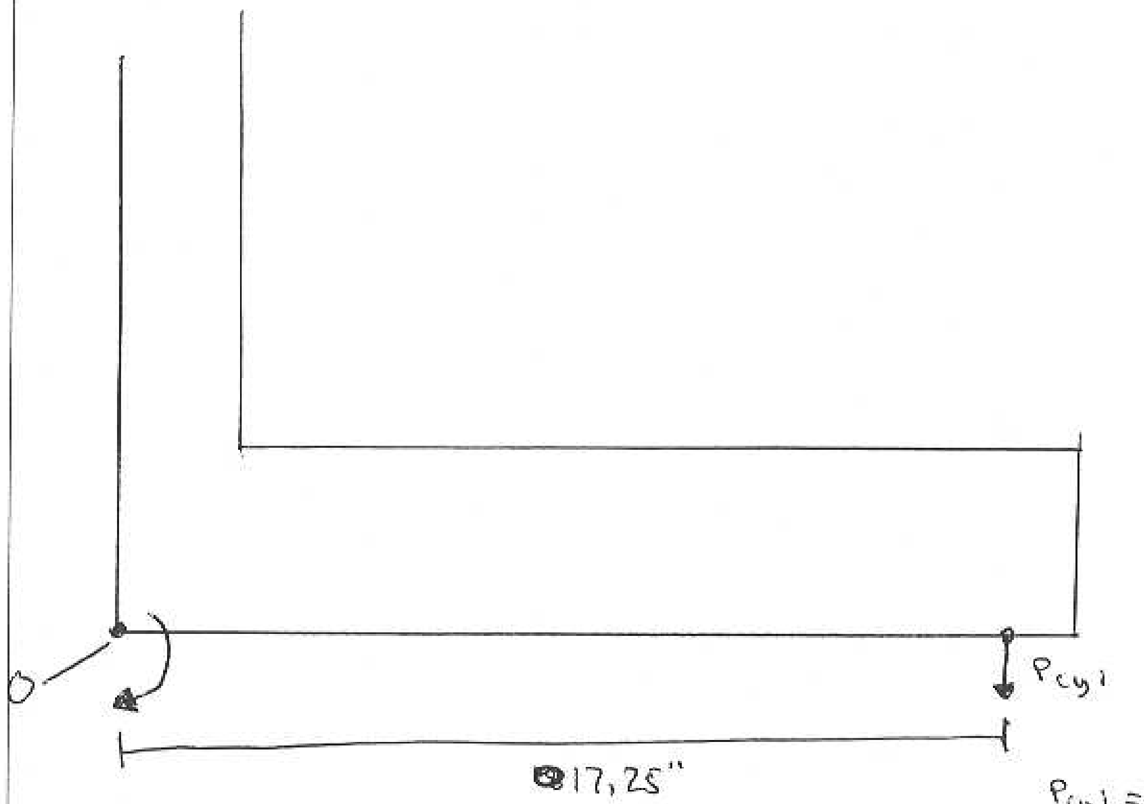
$$\theta = \tan^{-1} \frac{7.625}{24.5} = 17.29°$$

$$\Delta c.g = \sqrt{24.5^2 + 7.625^2} - 24.5 = 1.159"$$

$$\frac{1.159"}{12"} \cdot 9000\# \cdot \frac{32ft}{s^2} = \frac{1}{2} \cdot 9000 \cdot v^2$$

conservation of energy
$m \cdot g \cdot h = \frac{1}{2} m v^2$

$v = 2.486$ ft/s $=$ $\boxed{1.695 \text{ m.p.h to tip device}}$



$P_{cyl} =$

$M_o = 17.25'' \cdot 8435.09\,\# = 145505.3$ in-#

τ = torsional stress $= \dfrac{M_o \cdot c}{J}$

c = Radius of Gyration = 2.06 in
J = Polar moment of inertia = 42.1 in^4

specific to 6"x4"x1/2" steel tubing obtained from A.S.D. 9th Edition

$$\tau = \frac{145505.3 \cdot 2.06}{42.1} = 7119.72 \text{ p.s.i.}$$

angle of Rotation

- Polar moment of inertia method

$J = 42.1\ in^4$ Shear Elasticity $= E_s = 12{,}000{,}000$ l = length = 20"

$$\theta = \frac{M_o \cdot l}{E \cdot J} = \frac{145505.3 \cdot 20}{12000000 \cdot 42.1} = .00576 \text{ rad}$$

$\boxed{= .33^\circ}$

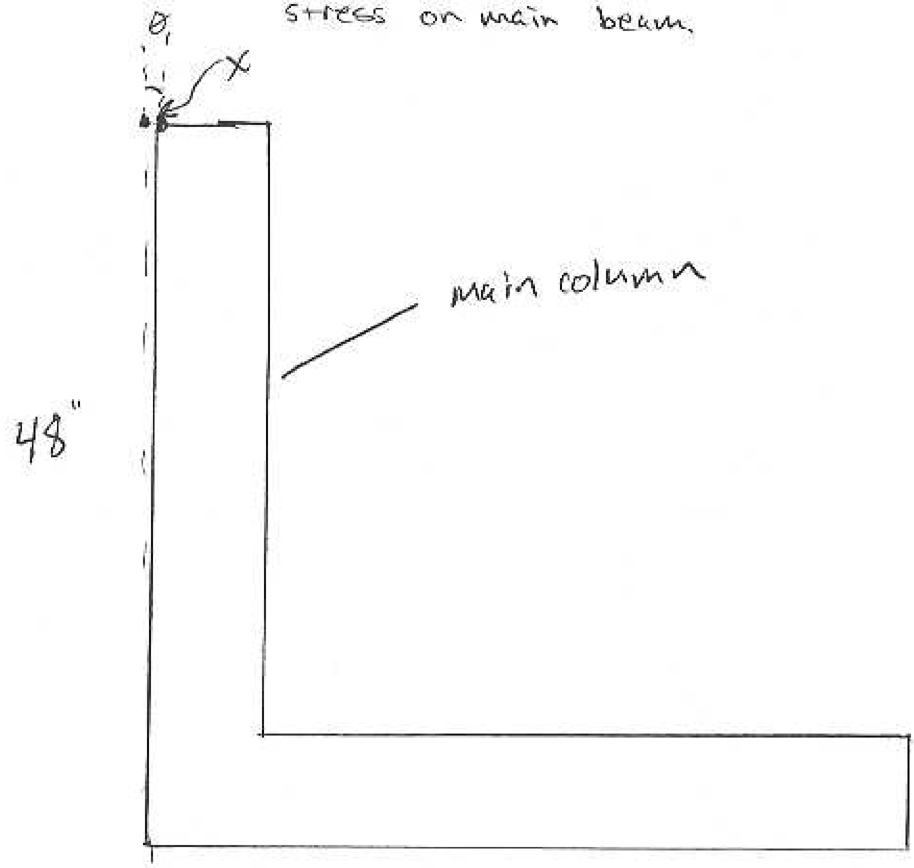
- Resistance method

R = resistance $= \dfrac{2 \cdot t \cdot b^2 \cdot d^2}{b + d} = 57.6$

t = thickness = 1/2
b = length of X-sect = 6
d = width of X-sect = 4

$$\Theta = \frac{M_o \cdot l}{E_s \cdot R} = .0042 \text{ rad} \; \boxed{= .24^\circ}$$

Horizontal movement on main column caused by torsion stress on main beam.



$\tan \theta = \frac{X}{48}$ $\qquad X = \tan\theta \cdot 48$

$.24^\circ \leq \theta \leq .33^\circ$

$\tan .24 \cdot 48 = .201$

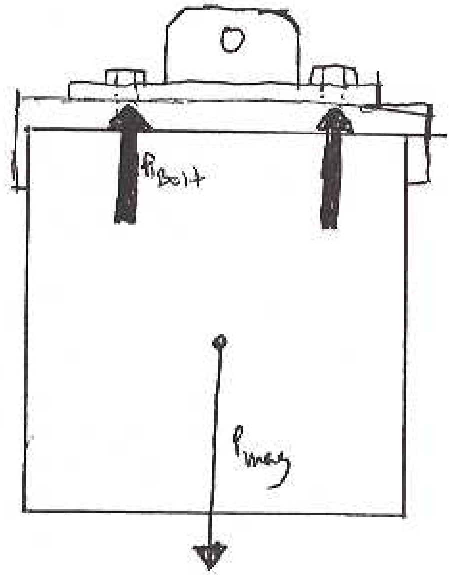
$\tan .33 \cdot 48 \approx .276$

$\boxed{.201" \leq X \leq .276"}$

This is how much the top of the lift will move.

Joe Martinez 6/7/02 | Load on the Bolts attaching to the magnet

Flat Surface



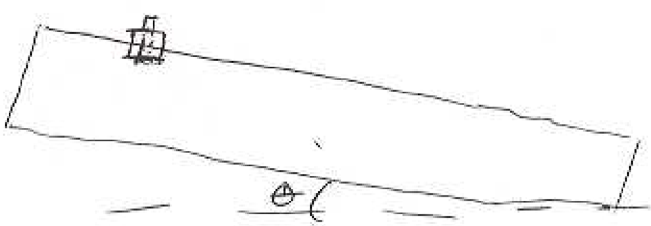
$P_{mag} = 4500$ #

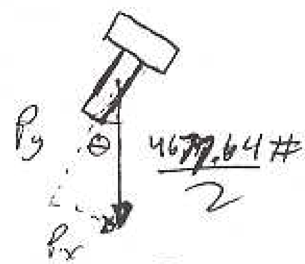
$R_{Bolt} = \frac{P_{mag}}{2} = 2250$ #

$\frac{1}{2}$" diameter bolts

σ_T = Tensile stress $= \frac{2250 \#}{\pi \cdot \frac{1}{4}^2} = \boxed{11459.2 \text{ p.s.i}}$

INCLINE:





P_y = Tensile component

$= \frac{4677.64}{2} \cdot \cos\theta$

$= 2,310.2$ #

shear component $= P_x$

$= \frac{4677.64}{2} \cdot \sin\theta$

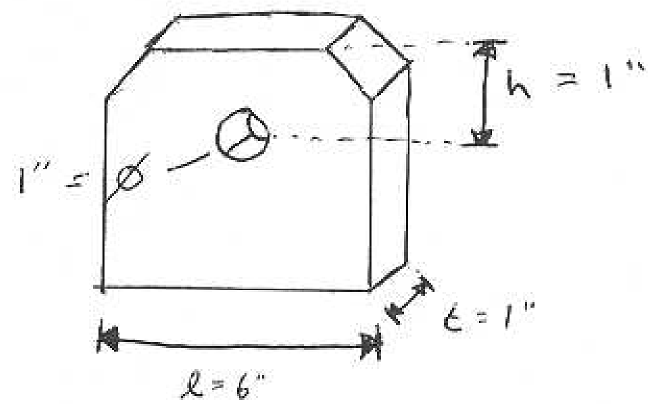
$= 364.78$ #

Tensile Stress $= \sigma_T$

$\frac{2310.2}{\pi \cdot .25^2} = \boxed{11765.7 \text{ p.s.i.}}$

A 307 $\frac{1}{2}$" Bolt should allow a load of $6.1 - 2.2 = 3.9$ Kips $= 3900$ #
and a shear of about 2.1 Kips = 2100 #
so an A307 $\frac{1}{2}$" Bolt is ok. But σ_T might be too great then need A325 stee

A-7.11



$$A_{tensile} \geq \frac{P}{\sigma_{max}} \geq \frac{4677.64\ \#}{10,000\ psi} \geq .467764\ in^2$$

$$A_{shear} \geq \frac{P}{\tau_{max}} \geq \frac{4677.64}{10,000} \geq .467764\ in^2$$

$$A_{\tau} = \frac{1}{2} \cdot (l - \varnothing)$$

$$A_{\tau} = 2\ \text{sides} \cdot 1"\text{thick} \cdot h$$

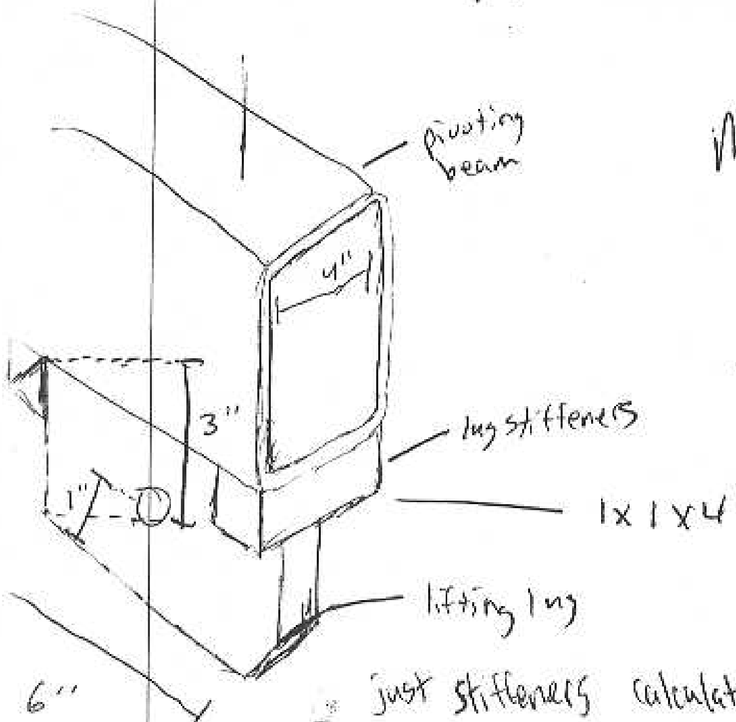
$$A_T = 2.5 in^2$$

$$A_{\tau} = 2 \cdot 1 \cdot 1 = 2\ in^2$$

$$\sigma_T = \frac{4677.64}{2.5} = 1871.1\ psi. \Rightarrow O.K.$$

$$\tau_s = \frac{4677.64}{2} = 2338.82\ psi \Rightarrow O.K.$$

Joe Martinez 6/11/02 ~~Add~~ sizing of lug stiffeners.



$M_{max} = \frac{P \cdot l}{4}$

$M_{max} = 4500 \text{ in\#}$

$\sigma = \frac{M_{max} \cdot y}{I}$

just stiffeners calculated for.

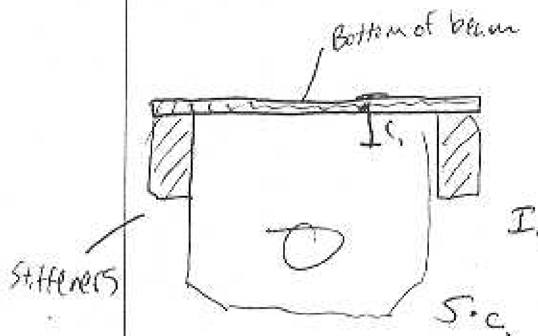
try 1" thick x 1   2 lug stiffeners

$I = \frac{1}{12} \cdot 1 \cdot 1^3 = .0833 \quad \cdot 2 \text{ stiffeners} = .16667 \text{ in}^4$

$\sigma = \frac{My}{I} = \frac{4500 \cdot .5}{.166667} = 13{,}500 \text{ p.s.i.}$

---

check w/ contribution of beam



$A_{beam} = \frac{1}{2} \cdot 6 = 3 \text{ in}^2 \qquad I_{beam} = .0625 \text{ in}^4$

$A_{stiffeners} = 2 \cdot 1 \cdot 1 = 2 \text{ in}^2 \quad I_{stiffeners} = .1666667 \text{ in}^4$

$A_{totals} = 5 \text{ in}^2$

$I_{composite} = I_1 + d_1^2 A_1 + I_2 d_2^2 A_2$

$x_1 = .25 \qquad x_2 = 1.5''$

$5 \cdot c_1 = A_1 \cdot x_1 + A_2 x_2$

$c_1 = .75''$

$d_1 = c_1 - x_1 = .5$

$d_2 = c_1 - x_2 = .25$

$.0625 + .5^2 \cdot 3 + .1666666 \cdot .75^2 \cdot 2 = 2.10417 \text{ in}^4$

$\sigma = \frac{4500 \cdot .75}{2.10417} = 1603.96 \text{ psi.}$

Lug stiffeners are needed because the stress on the pivoting arm are too great

Joe Martinez 6/7/02 | weld sizing of lug

FLAT

Since the thicker of the two parts welded together is over 3/4", a minimum fillet weld of 5/16" is needed

Use E70XX, $\frac{5}{16}$" fillets

$P = A \cdot F_v$

$Area = .707 \cdot \frac{5}{16}$

$F_v = l \cdot .3 \cdot 70000$

$P = load$

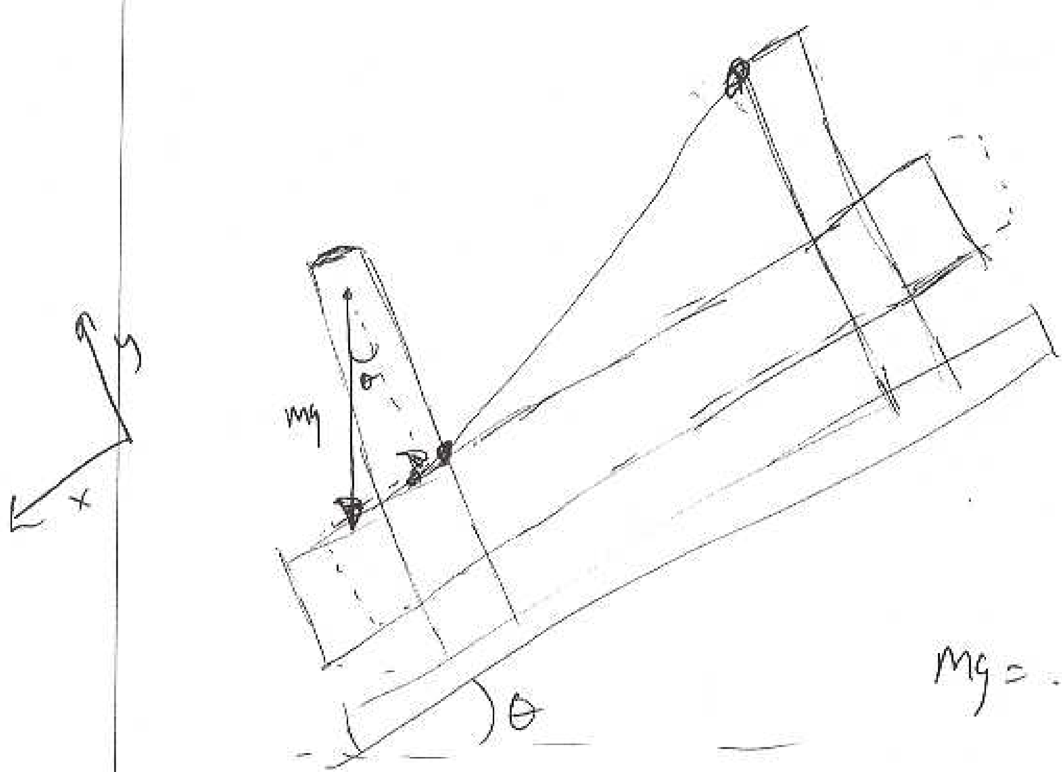
$4500 = .707 \cdot \frac{5}{16} \cdot l \cdot .3 \cdot 70000$

$\frac{4500}{.707 \cdot \frac{5}{16} \cdot .3 \cdot 70000} = l = .969893"$

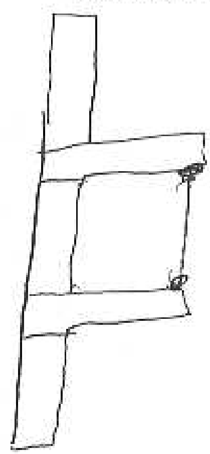
The lug is specified as 6" which is about 6 times what is necessary

Any weld metal in the E60-series can also be used.

Joe Martinez 6/11/02 | Torsion on main column w/ cross support.

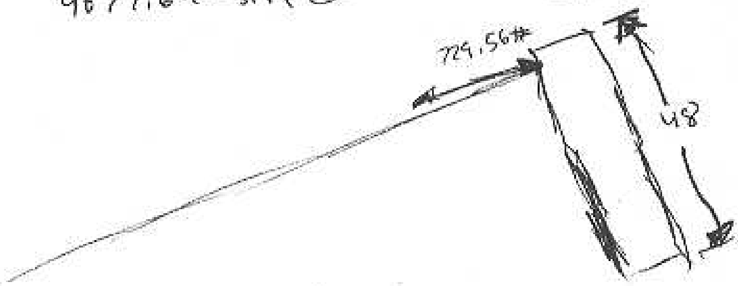


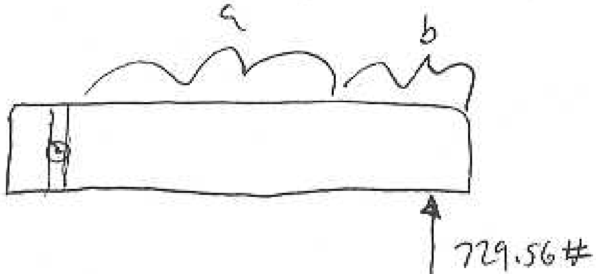
Mg = 4677.64#



$F_y = \cdot 4677.64\# \cdot \cos\theta = 4620.4\#$

$F_x \quad 4677.64 \cdot \sin\theta = 729.56\#$





$M_o = 729.56 \cdot 27.5'' = 20047.5 \text{ in\#}$ $\quad 4''\times 4''\times \tfrac{1}{2}''$

τ = torsional stress = $\dfrac{M_o \cdot c}{J}$ $\quad$ c = radius of gyration = 1.39 in

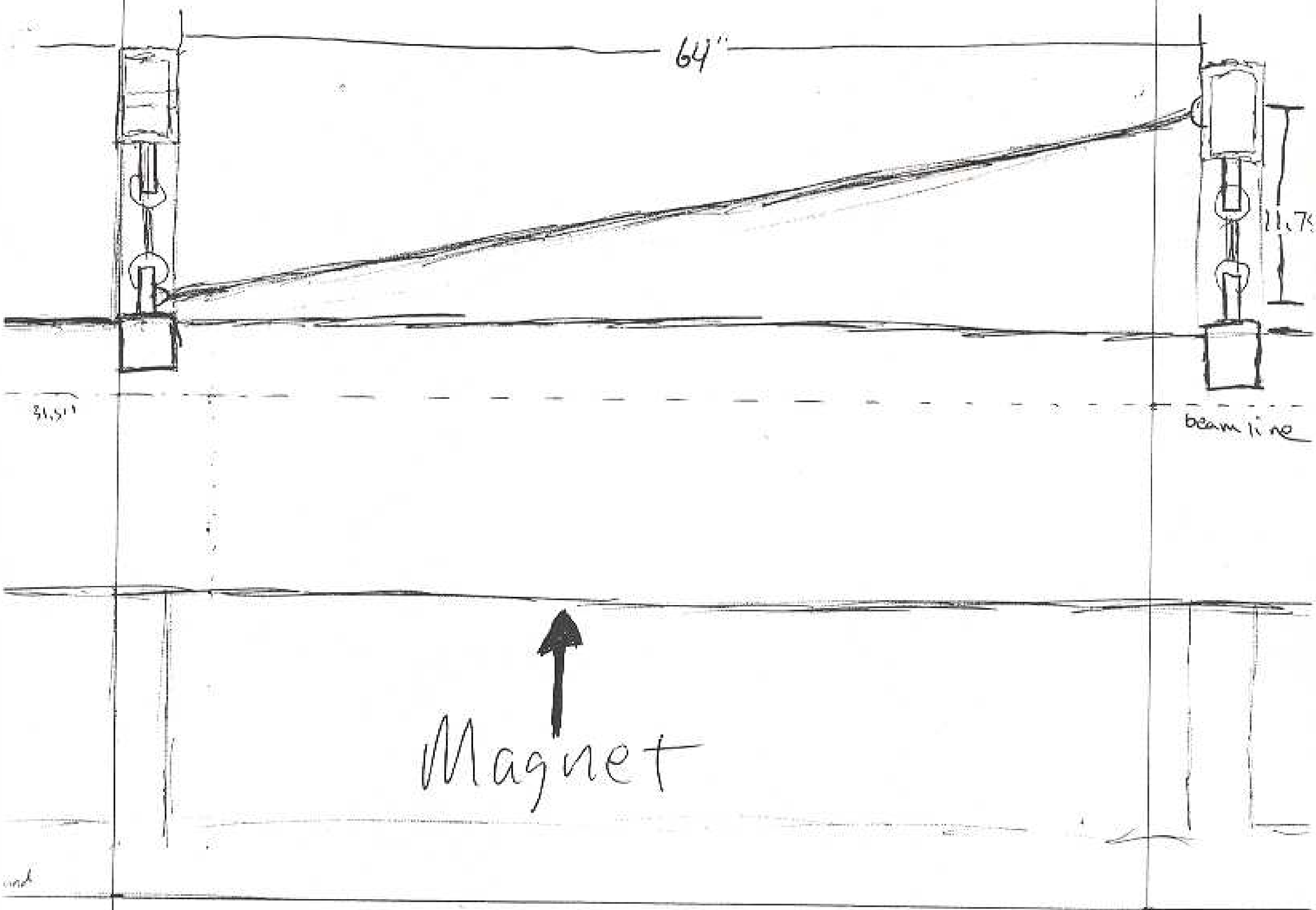
J = 21.8 in^4

E_s = 12,000,000 psi.

= 1278.26 psi

$\theta = \dfrac{M_o \cdot l}{E_s J} = .210759^\circ$

Joe Martinez 7/16/02 Cross bracing length



length of cross support $= \sqrt{64^2 + 11.75^2} = 65.07$"