LAWRENCE BERKELEY NATIONAL LABORATORY - UNIVERSITY OF CALIFORNIA

ENGINEERING NOTE

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AUTHOR	DEPARTMENT	DATE	checked by
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PROGRAM - PROJECT - JOB

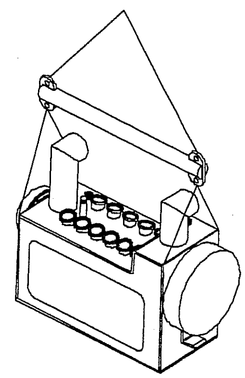
LARGE HADRON COLLIDER (LHC)
IR, FEEDBOX

TITLE

STRESS ANALYSIS OF LHC DFBX LIFTING FIXTURE

SUMMARY/ABSTRACT

This engineering note was written to summarize the required parameters and document the stress analysis for the lifting fixture for the DFBX (distribution feedbox). It will be used during assembly, acceptance test, and shipping.



INDEX:

DESIGN REQUIREMENT AND ASSUMPTIONS

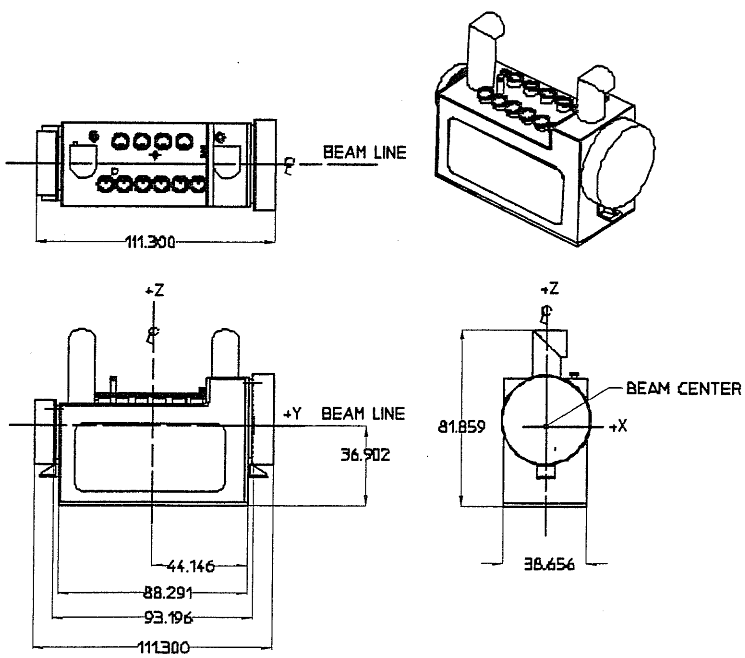
Requirements:

a) The lifting fixture must be able to lift 6.5 ton DFBX.
b) The lifting fixture shall be rated for 6.6 ton lifting and must be tested to twice the design load.
c) When not in use lifting fixture should occupy minimum footprint at assembly floor.
d) Lift fixture must be able to operate with minimum personnel.
e) The lifting fixture will be able to lift the DFBX at least 1 foot off the floor. (This height value will set the limit angle to the allowable swing of the DFBX).
f) Lifting fixture must be able to be used in a 154.5inches height crane at LBNL BLDG. 77A assembly building (Bldg 77A is a future DFBX assembly area).

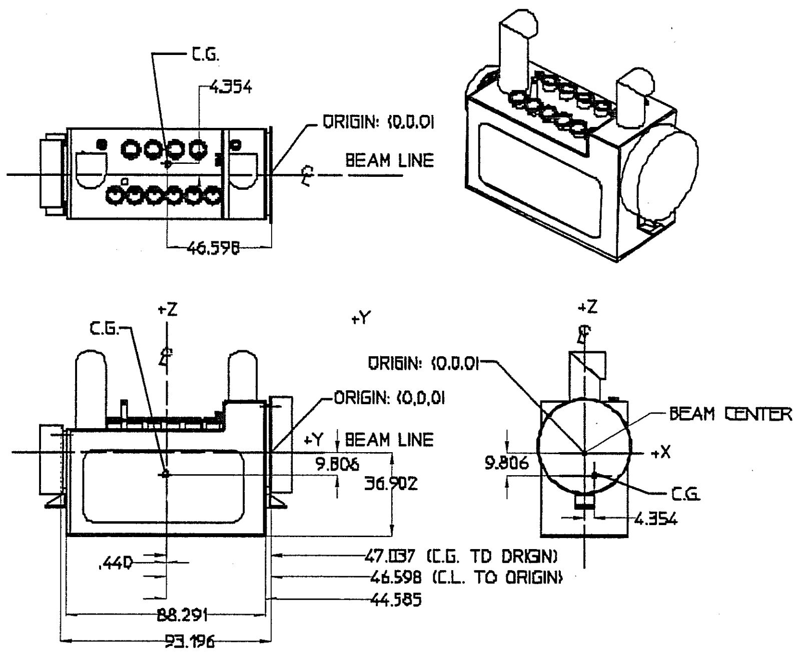
Assumptions:

a) No impulsive loading (No sudden vertical drop nor sudden horizontal deceleration) on the lifting fixture components.
b) The DFBX will not swing more than 5 degrees in any direction.
c) The DFBX lifting will be done at ambient temperature of 68 ~100 deg. F.

1. DFBX LAYOUT (Overall Dimensions)

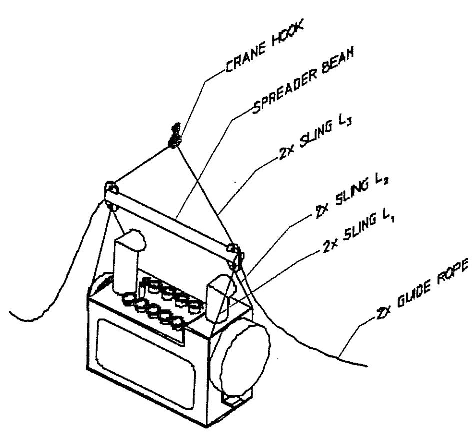


2. THE CENTER OF GRAVITY OF DFBX (X,Y,Z: 4.354 in, -47.037 in, -9.806 in)



THE CENTER OF GRAVITY OF DFBXC OR DFBXD

3. DFBX SUSPENSION METHOD (SPREADER BEAM & SLINGS)



SPREADER BEAM & SLINGS

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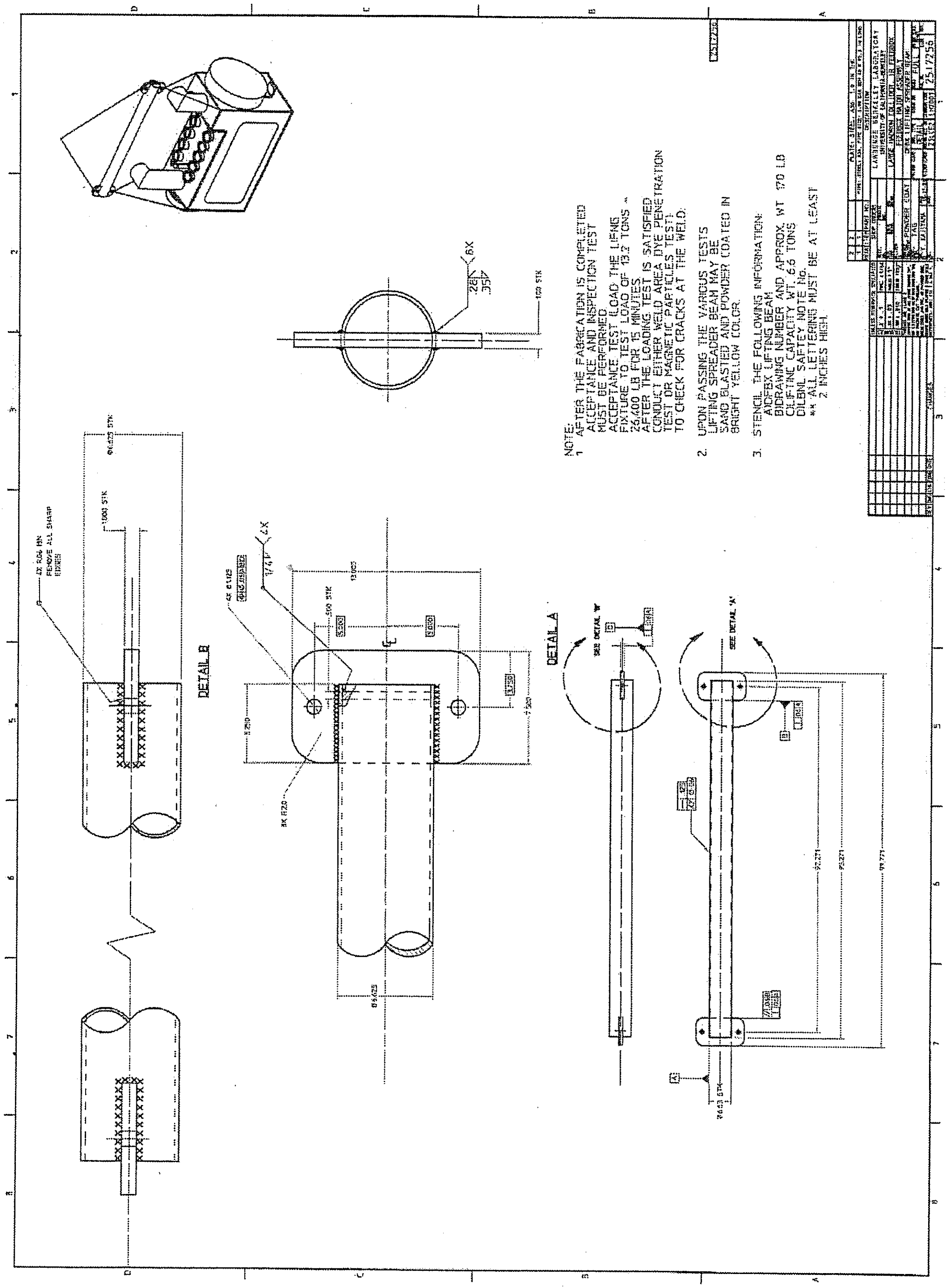
4. DETAIL DRAWING OF DFBX LIFTING FIXTURE SPREADER BEAM



5. LHC DFBX LIFTING FIXTURE PARAMETER AND STRESS ANALYSIS SUMMARY

Since DFBXC is the heaviest distribution box , as result, DFBXC as the lifting fixture design criteria.

a) CAD measured weight of DFBXC W = 5.5 tons = 11,000 lb

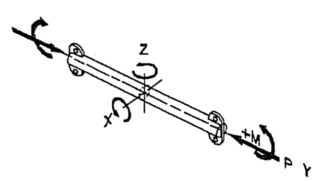
b) Design weight of DFBXC W_{design}= 6.6 tons = 13,200 lb

c) The Center of Gravity of DFBXC:
(X, Y, Z) = (4.354 in, -47.037 in, -9.806 in)

d) DFBX Lifting fixture spreader beam:
   (1) Total length: 99.77 inch
   (2) Hole center to hole center = 92.271 in.
   (3) Total weight = 190 lb
   (4) Material: 6.0 in dia X Sch 40 A36 steel tubing.

Physical characteristics of the spreader beam.

Material: A36 Construction Steel
F_y = Minimum yield strength = 36 x 10^3 psi
F_u = minimum ultimate strength = 58 x 10^3 psi
E = Modulus of Elasticity = 29 x 10^6 psi
ν = Poisson's ratio = 0.30



(5) Total Stress of the lifting fixture spreader beam at the center.

$$\sigma_{total} = \sigma_{axial} + \sigma_{bending} = \frac{F}{A} + \sum \frac{Mc}{I}$$

$$= \sigma_{axial} + \sigma_{\text{bending stress from own weight}} + \sigma_{\text{bending stress from eccentric loading due to deflection and iinherent sagitta from manufacturing tolerance}} + \sigma_{\text{bending stress from eccentric support}}$$

$$= 1{,}409 \text{ psi} + 237.5 \text{ psi} + 80.5 \text{ psi} + 4{,}637.5 \text{ psi}$$

$$= 6{,}365 \text{ psi}$$

Since σ_{total} is less than 1/5 of $S_{ultimate}$ of 58,000 psi therefore the design criteria of Pub 3000 for the design fixture is satisfied.

(6) Stress of the lifting fixture spreader beam flange:

$$\text{Shear Stresss} = \tau_{\substack{\text{tear}\\\text{out}}} = 5{,}134 \text{ psi}$$

$$\text{Bearing Stress} = \sigma_{\text{bearing}} = 11{,}734 \text{ psi}$$

This also satisfied Pub 3000 requirement.

For the location of the each slings see the diagram at page 3 of this eng. note. (Slings, turnbuckles, shackles, and swivel hoist rings are all purchase from Carpenter Rigging Inc. and custom made items will be tested to twice the specified load and certified by them)

e) Primary Lifting Slings L_3:

(1) Primary lifting slings L_3 are equal in length. One has integral turn buckle to adjust the length for other DFBX model.

(2) Total length (include sling, shackle, and hoist ring at DFBX)
Sling L_3 = 60.21 in

(3) Included angle:
(a) Sling L_3: 40 degree from horizontal.
for both sides.

(4) Tension Load = 10,268 lb

(5) Size of Slings: cross section = 6X25 steel wire rope with 0.88 in dia.
6.3 ton capacity (standard size).
(ref. Carpenter Rigging Inc)

f) Secondary Lifting Slings: L_1 and L_2

(1) Secondary lifting slings are unequal in length. The sling L_2 have has built in turn buckle to adjust the length for other DFBX.
(a) Sling L_1: 2 units
(b) Sling L_2: 2 units

(2) Total length (include sling, shackle, and hoist ring at DFBX)
(a) Sling L_1 = 46.74 in.
(b) Sling L_2 = 43.52 in.

(3) Included angle:
(a) For Sling L_1: 63.4 degree. (from horizontal)
(b) For Sling L_2: 73.7 degree. (from horizontal)

(4) Tension Load on the slings.
(a) Sling L_1: T_1 = 2,815 lb
(c) Sling L_2: T_2 = 4,490 lb

(5) Size of Slings:

(a) Sling L_1: cross section = 6X 19 steel wire rope with 0.56 in dia. With 2.8 ton capacity (standard size). (ref. Carpenter Rigging)

(b) Sling L_2: same cross sectional size of wire rope as Sling L_1

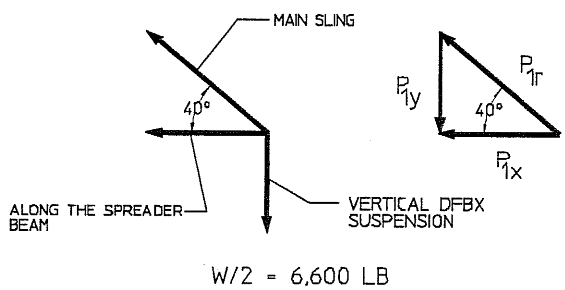
6. STRESS ANALYSIS OF DFBX LIFTING FIXTURE (SPREADER BEAM AND SLINGS)

Although the total actual weight we need to lift is 5.5 tons (11,000 lb) at this time but we will be adding many features before the final assembly is completed. As a result, I added 20% to present estimated weight. Therefore

Design weight W = 5.5 ton + 20% = 13,200 lb.

Tension Load on Primary Sling L_3

Since the center line of the DFBX and the center of gravity of the DFBX along the longitudinal is only offset by 0.440 inches so we will design the Primary Sling L_3 to be equal in distance.



$$\sin\theta = \frac{P_{1y}}{P_{1r}}$$

$$P_{1r} = \frac{P_{1y}}{\sin\theta}$$

$$= \frac{6{,}600\ \text{lb}}{\sin 40^\circ} = \frac{6{,}600\ \text{lb}}{.6428} = 10{,}268\ \text{lb}$$

Therefore the tension T_3 on the sling L_3 is

$$T_3 = P_{1r} = 10{,}268 \text{ lb}$$

$$\tan\theta = \frac{P_{1y}}{P_{1x}}$$

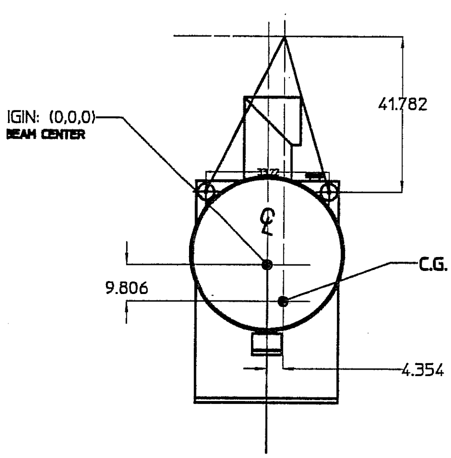
$$P_{1x} = \frac{P_{1y}}{\tan\theta} = \frac{6{,}600 \text{ lb}}{\tan 40^\circ} = \frac{6{,}600 \text{ lb}}{.839} = 7{,}867 \text{ lb}$$

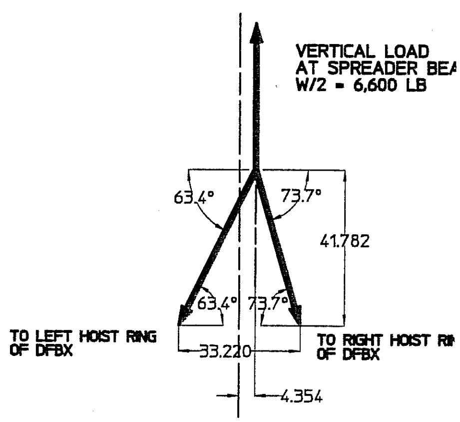
Sling length: L_3

h_1 = Vertical height from Spreader beam eye to Crane hook = 38.712 in

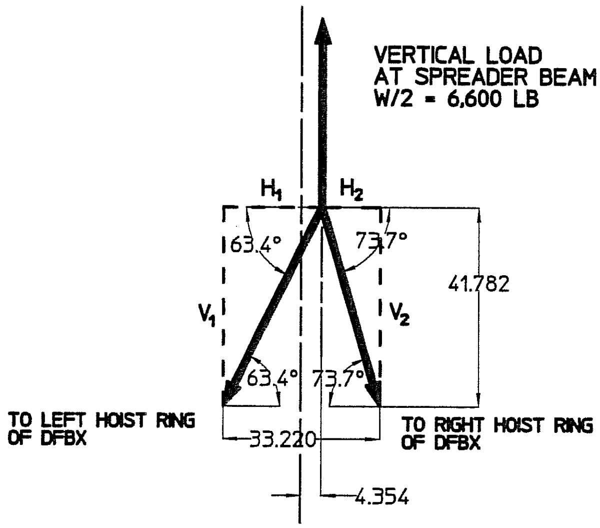
$$\text{Sling } L_3 = \frac{h_1}{\sin 40^\circ} = \frac{38.712 \text{ in}}{0.643} = 60.21 \text{ in}$$

Tension Load on Secondary Slings: L_1 and L_2





Due to offset of C.G. from the center line axis the unequal angle is generated



$$\sum F_x = 0$$

$$= H_2 - H_1$$

$$= T_2 \cos 73.7^\circ - T_1 \cos 63.4^\circ = 0$$

$$= 0.281\,T_2 - 0.448 T_1 = 0$$

$$T_1 = 0.627 T_2$$

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$$\sum F_y = 0$$

$$= -V_1 - V_2 + 6{,}600\,\text{lb} = 0$$

$$V_1 = T_1 \sin 63.4^\circ = 0.894T_1$$

$$V_2 = T_2 \sin 73.7^\circ = 0.960T_2$$

$$-T_1 \sin 63.4^\circ - T_2 \sin 73.7^\circ + 6{,}600\,\text{lb} = 0$$

$$-0.894T_1 - 0.960T_2 + 6{,}600\,\text{lb} = 0$$

$$0.894T_1 + 0.960T_2 = 6{,}600\,\text{lb}$$

$$0.894(0.627T_2) + 0.960T_2 = 6{,}600\,\text{lb}$$

$$1.521T_2 = 6{,}600\,\text{lb}$$

$$T_2 = 4{,}341\,\text{lb}$$

$$T_1 = 0.627T_2$$

$$= 0.627(4{,}341\,\text{lb}) = 2{,}722\,\text{lb}$$

Therefore the tension on sling L_1 and L_2 are T_1=2,722 lb and T_2 = 4,341 lb respectively.

Sling length: L_1 and L_2

h_2 =Vertical height from Spreader beam eye to DFBX Hoist ring = 41.782 in

$$\text{Sling } L_1 = \frac{h_2}{\sin 63.4^\circ} = \frac{41.782\,\text{in.}}{0.894} = 46.74\,\text{in}$$

$$\text{Sling } L_2 = \frac{h_2}{\sin 73.7^\circ} = 0.\frac{41.782\,\text{in.}}{0.960} = 43.52\,\text{in}$$

Spreader Beam Stress Analysis

Physical characteristics of the spreader beam.
Material: A36 Construction Steel

F_y = Minimum yield strength = 36 x 10^3 psi
F_u = minimum ultimate strength = 58 x 10^3 psi
E = Modulus of Elasticity = 29 x 10^6 psi
ν = Poisson's ratio = 0.30

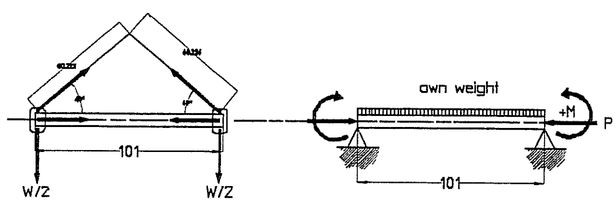
Size: ∅ 6.00 in Sch 40 pipe

O.D. = Outside diameter = 6.625 in.
I.D. = Inside diameter = 6.065 in.
t = Tube wall thickness = 0.28 in.
w = Weight per length = 18.97 lb/ft = 1.58 lb/in
W = Total weight of the pipe = 160 lb
A = Cross sectional Area = 5.58 in^2
I = Moment of inertia = 28.1 in^4
L = Length = 101 in

The stress and the deflection of the beam may be combined together by means of superposition.

Total Stress = Axial Stress
+ Bending Stress from own beam weight
+ Bending Stress from eccentric loading due to beam deflection and inherent sagitta during pipe manufacturing
+ Bending Stress from eccentric suspension

Lifting Beam sketch



A. Stress from Axial Load

$$\sigma_{axial} = \frac{P}{A} = \frac{(-7{,}867\,lb)}{(5.58\,in^2)} = -1{,}409\ psi$$

B. Bending Stress and The Deflection from Own Weight

Simply supported

Ref. Standard Handbook for
Mechanical Engineers
Baumeister & Marks
7th. Edition.
Pg 5~32
Case: Uniformly loaded

Nomenclature:

M_{max} = Maximum bending moment at center (in-lb)

For 6.0 in dia. Sch. 40 steel pipe

w = Weight per unit length. 1.58 lb/in

W = Total weight of the beam = (1.58 lb/in)(101 in) = 160 lb

r = radius of gyration 2.25 in

L = the length of the beam = 101 in

C = neutral axis to outer surface of the beam = 6.625/2 = 3.3125 in

I = moment of inertia of the beam = 28.1in^4

Δ_{ymax} = maximum deflection at the center of the beam (in)

1) At the center of the beam L/2

Max. Stress:

$$M_{max} = \frac{wL^2}{8} = \frac{(1.58)(101)^2}{8} = 2{,}015 \text{ in - lb}$$

$$\sigma_{max} = \frac{Mc}{I} = \frac{(2{,}015 \text{ inlb})(3.3125 \text{ in})}{28.1 \text{ in}^4} = 237.5 \text{ psi}$$

Deflection:

$$\Delta y_{max} = \frac{5wl^3}{384EI} = \frac{(5)(160 \text{ lb})(101)^3}{(384)(29 \times 10^6 \text{ psi})(28.1 \text{ in}^4)} = 0.0026 \text{ in}$$

C. Bending Stress from Eccentric loading caused by the Spreader Bar Deflection and The Curvature Built in During the Pipe Manufacturing

AUTHOR Yoichi Kajiyama | DEPARTMENT Mechanical Engineering | DATE 10/10/2001 | checked by Carol Corradi

Ref. Hand Book of Formula
For Stress & Strain
W. Griffel
Ungar Publishing 1976
Pg. 210~212

Nomenclature:

P = Axial load = 7867 lb
c = neutral axis to outer most surface = 3.3125 in.
$I = 28.1\ in^4$
L = 101 in
$\in_1$ = Eccentricity due to own weight deflection
= 0.0026 in (from above)
$\in_2$ = Eccentricity due to manufacturing tolerance
= 0.08417 in
From Kilsby-Roberts Tube & Bar Co.
Catlog 1992 pg 98
6.0 dia pipe has straightness tolerance of .03 in/3 ft
Therefore sagitta = (0.03 in/3ft)(101in)(1 ft/12 in)
=.0842 in

$\in_{total} = \in_1 + \in_2 = 0.0026 + 0.0842 = 0.0868$ in

λL = Load parameter (from graph below)

$$\lambda L = L\sqrt{\frac{P}{EI}} = (101\,\text{in})\sqrt{\frac{(7868\,\text{lb})}{(29 \times 10^6\,\text{psi})(28.1\,\text{in}^4)}} = 0.314$$

From eccentrically loaded column graph below: at λL = 0.314

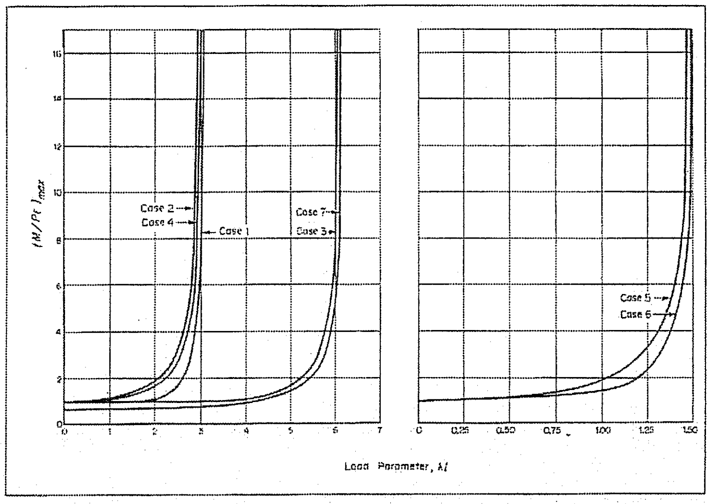
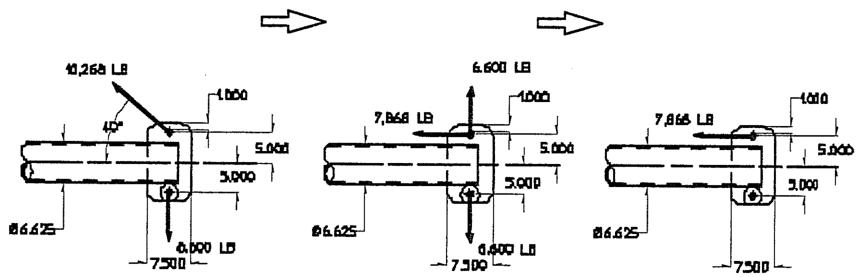


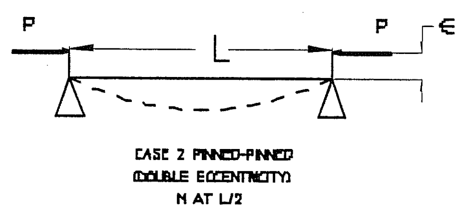
Fig. 2. Maximum moment of eccentrically loaded columns. As the axial load approaches the Euler, or critical load, the moment approaches infinity, indicating instability.

M/P∈ = 1 therefore M = P∈ = (7868 lb)(0.0868 in) = 683 in lb

$$\sigma_{eccl} = \frac{Mc}{I} = \frac{(683\,\text{inlb})(3.3125\,\text{in})}{28.1\,\text{in}^4} = 80.5\,\text{psi}$$

D. Bending Stress from Eccentricity of Suspension Location





$\in$ = Eccentricity of loading = 5.00 in.
Since from previous page of the engineering note,
$\lambda L = .314$

$M/P\in = 1$ therefore $M = P\in$ = (7868 lb)(5.00 in) = 39,340 in lb

$$\sigma_{ecc2} = \frac{Mc}{I} = \frac{(39340\,\text{inlb})(3.3125\,\text{in})}{28.1\,\text{in}^4} = 4637.5\ \text{psi}$$

Since various horizontal component stresses and deflections can be combine algebraically by means of superposition, therefore total stress at the center of the lifting spreader beam
will be:

$$\sigma_{\text{horizontal total}} = \sigma_{\text{axial}} + \sigma_{\substack{\text{bending from}\\ \text{own weight}}} + \sigma_{\substack{\text{bending from}\\ \text{eccl}}} + \sigma_{\substack{\text{bending from}\\ \text{ecc2l}}}$$

$$= 1{,}409\ \text{psi} + 237.5\,\text{psi} + 80.5\text{psi} + 4{,}637.5\ \text{psi}$$

$$= 6{,}365\ \text{psi}$$

The Euler's critical buckling column load and stress for a centrally loaded pinned-pinned column, with K=1 is:

$$P_{cr} = \frac{K\pi^2 EI}{L^2} = \frac{(1)(\pi^2)(29\text{x}10^6)(28.1)}{(101)^2} = 788.4\ \text{x}\ 10^3\ \text{lb}$$

Therefore at Euler's critical buckling column load the maximum axial stress will be

$$\sigma_{\text{axial}} = \frac{P_{cr}}{A} = \frac{788.4\ \text{x}\ 10^3\ \text{lb}}{5.58\,\text{in}^2} = 141.3\ \text{x}\ 10^3\ \text{psi}$$

Therefore our spreader beam see P = 7,868 axial load and total stress generated from eccentric loading is 6,407.3 psi , as result, it will not buckle.

SATISFYING THE ALLOWABLE STRESS CODE OF AISC CODE OF STEEL CONSTRUCTION.

Ref. Manual For Steel Construction
Allowable Stress Design
American Institute of Steel Construction, 9th edition
Pg5-42~43 Chapter E

A. Allowable Axial Compressive Stress

Since Lifting Spreader Beam behave as pin ends supported condition, therefore,

$$\sigma_{axial} = \frac{P}{A} = \frac{(-7{,}867\,\text{lb})}{(5.58\,\text{in}^2)} = -1{,}409\ \text{psi}$$

a) Effective length factor shall be K = 1.0 (pin ends support)
b) Largest effective slenderness ratio of unbraced beam is

$$\frac{KL}{r_g} = \frac{(1)(101)}{2.25} = 44.89$$

c) "C_C" the column slenderness ratio separating elastic and inelastic buckling is

$$C_C = \sqrt{\frac{2\pi^2 E}{F_y}} = \sqrt{\frac{2\pi^2 (29\ \text{x}10^6\ \text{psi})}{36\ \text{x}\ 10^3\ \text{psi}}} = 126.10$$

Since KL/r< C_C

Allowable axial compressive stress

$$F_a = \frac{[1 - \frac{\frac{KL}{r}}{2C_c^2}]F_y}{\frac{5}{3} + \frac{3(\frac{KL}{r})}{8(C_c)} - \frac{3(\frac{KL}{r})^3}{8(C_c^3)}} = \frac{[1 - \frac{(44.89)}{2(126.10)^2}](36{,}000\ \text{psi})}{1.667 + \frac{3(44.89)}{8(126.10)} - \frac{(44.89)^3}{8(126.10)^3}}$$

$$= \frac{35{,}949.18}{[1.667 + 0.1335 - .00564]} = 20{,}029\ \text{psi}$$

Since $\sigma_{axial} < F_a$ 1,409 psi< 20,029 psi

Therefore spreader beam satisfies AISC design code for axial Loading.

B. Allowable Bending Stress of Circular Tubes

Ref. Manual for Steel Construction:
Allowable Stress Design.
American Institute of Steel
Construction, 9th. Edition.
Pg.5-49 Chapter F

Allowable Bending Stress $F_b = 0.60\ F_y$ where F_y is yield strength of A36 steel

$$=(.66)(36{,}000\text{ psi}) = 21{,}600\text{ psi}$$

Actual bending stress:

$$\sigma_{\substack{\text{total}\\\text{bending}}} = \sum \frac{Mc}{I}$$

$$= \sigma_{\substack{\text{bending stress from}\\\text{own weightl}}} + \sigma_{\substack{\text{bending stress from eccentric}\\\text{loading due to deflection and}\\\text{inherit sagitta from manufacturing}\\\text{tolerance}}} + \sigma_{\substack{\text{bending stress from}\\\text{eccentric support}}}$$

$$= 237.5\text{ psi} + 80.5\text{ psi} + 4{,}637.5\text{ psi}$$

$$= 4{,}956\text{ psi}$$

$$4{,}956\text{ psi} < 21{,}600\text{ psi}$$

Therefore spreader beam satisfies AISC design code for allowable bending stress.

C. Combined Stresses of Axial Compression and Bending

Checking whether spreader beam will satisfy AISC Steel Construction allowable stress design code for axial compression and bending.

Ref. Manual for Steel Construction:
Allowable Stress Design.
American Institute of Steel
Construction, 9th. Edition.
Pg.5-54~56 Chapter H

Members subjected to both axial compression and bending stresses must satisfy Following conditions:

F_y = min yield strength of the material = 36 x 103 psi
F_a = allowable Axial Stress = 20,029 psi
F_b = allowable bending stress = 0.60 F_y = 23,760 psi
K = effective length factor in the plan of the bending = 1

L = effective length = 101 in
r_b = radius of gyration = 2.25 in
KL/r_b = largest effective slenderness ratio of unbraced beam
= 44.889
f_a = computed axial stress = 1,409 psi
f_b = computed compressive bending stress at the point under consideration
= 6,365 psi

F_e'= Euler stress divided by a factor of safety

$$= \frac{12\pi^2 E}{23(\frac{KL_b}{r_b})^2} = \frac{12\pi^2(29 \text{ x} 10^6 \text{ psi})}{23(44.889)^2} = 74.109 \text{ x } 10^3 \text{ psi}$$

From eq (H1-2)
For $f_a/F_a \leq 0.15$ since 0.070<0.15

$$\frac{f_a}{F_a} + \frac{f_{bx}}{F_{bx}} + \frac{f_{bz}}{F_{bz}} \leq 1.0$$

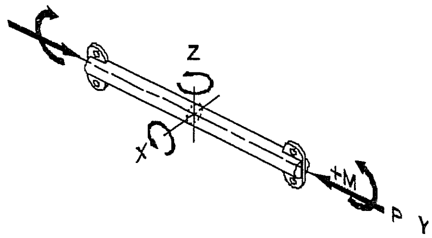
Since bending stress about Z - axis (lateral bending stress) is zero therefore equation (H1 - 2) becomes

$$\frac{(1{,}409 \text{ psi})}{(20{,}029 \text{ psi})} + \frac{(6{,}365 \text{ psi})}{(23{,}760 \text{ psi})} + \frac{0}{(23{,}760 \text{ psi})} \leq 1.0$$

$$0.070 + 0.27 + 0 \leq 1.0$$

$$0.34 < 1.0$$

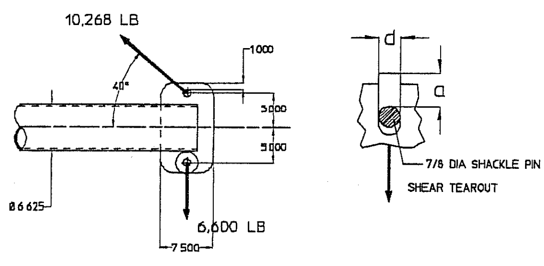
Since it is less than 1.0, therefore, it met AISC design code requirement Regarding combine stress of compressive axial and bending stress.



7. SPREADER BEAM FLANGE STRESS ANALYSIS

Spreader beam flange with suspension force acting.

Probable bearing failure of the flange occurs with two shear tearout surface areas described below. Therefore,



F = 10,268 lb
a = shear length =1.0 in
t = thickness of the plate =1.0 in
d = dia. Of shackle pin
= 0.875 in dia.

$$\text{Shear Stresss} = \tau_{\substack{\text{tear}\\ \text{out}}} = \frac{\text{force acting}}{\text{shear area}} = \frac{F}{2(at)} = \frac{10{,}268\ \text{lb}}{2(1)(1)} = 5{,}134\ \text{psi}$$

$$\text{Bearing Stress} = \sigma_{\text{bearing}} = \frac{\text{force acting}}{\text{bearing area}} = \frac{F}{dt} = \frac{10{,}268\ \text{lb}}{(0.875\ \text{in})(1\text{in})} = 11{,}734\ \text{psi}$$

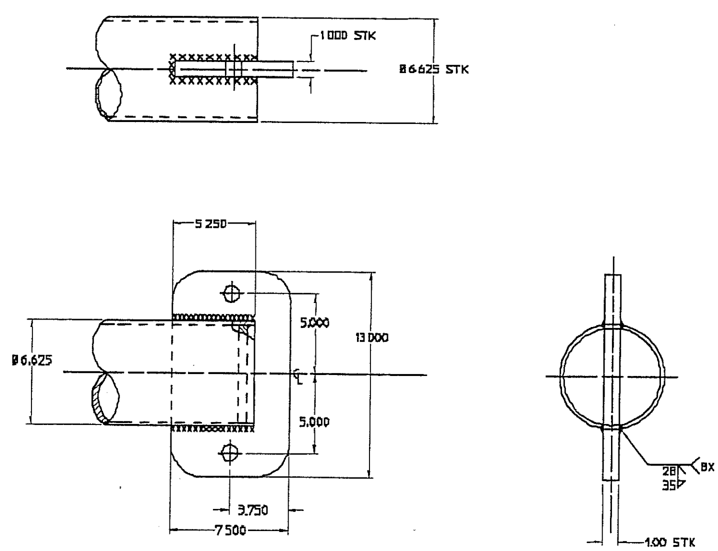
From AISC Allowable Stress Design manual:

Allowable shear stress = 0.40 S_y = 0.4(36 x 10^3 psi) = 14.40 x 10^3 psi
Allowable bearing stress = 0.90 S_y = 0.9(36 x 10^3 psi) = 32.40 x 10^3 psi

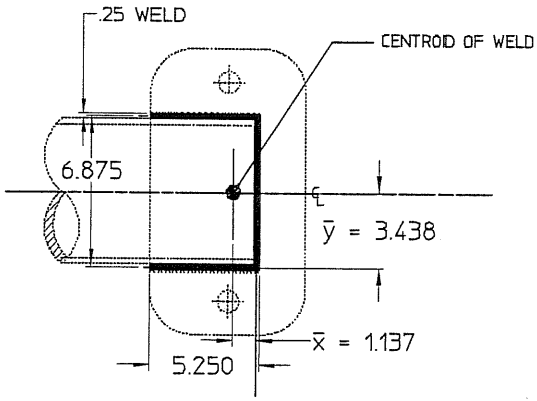
Therefore both shear and bearing stress at the flange meet AISC design requirement.

8. SIZING OF THE WELD AT SPREADER BEAM AND THE VERTICAL FLANGES

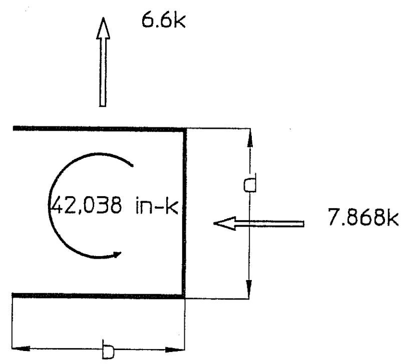
The "X" marks on the drawing below are the areas to be welded



The heavy black line below indicates the fillet weld



Assume ¼ in fillet weld is applied.
Then weld throat of ¼ in weld becomes (1/4om) sin 45°= (.25(.707) = .177 in.
Therefore, the weld cross sectional area becomes
A_w= 2(.177 in)(5.25 in + 5.25 in + 6.875 in) = 6.151 in^2



Ref. "Design of Welded Steel Structures" by Blodgett, section 7.4
Note: loads shown are total loads for welds on both sides of pipe: electrode to be E770-XX F_{allow} = 14.847 ksi

n = 2 welds b = 5.25 in d = 6.875 in

$$A_w = n\,(2b + d)$$
$$= 2[2(5.25\text{ in}) + 6.875\text{ in}]$$
$$= 34.75\text{ in}$$

$$J_w = \frac{(2b+d)^3}{12} - \frac{b^2(b+d)^2}{2b+d}$$

$$f = \left[\left(\frac{6.6k}{A_w} + \frac{42.038 \text{ x } k \text{ x } c_v}{J_w}\right)^2 + \left(\frac{7.868k}{A_w} + \frac{42.038\text{ in x } k\, c_h}{J_w}\right)\right]^{.5}$$

$$f = 1614.49\ \frac{\text{lb}}{\text{in}}$$

size of fillet weld required: $\omega = 0.11$ in min.

9. WIND LOAD:

Although we assumed that lifting of DFBX will not take place at outside of building 77A but there is a case when we are loading on to the truck for the shipment to CERN (Geneva Switzerland).

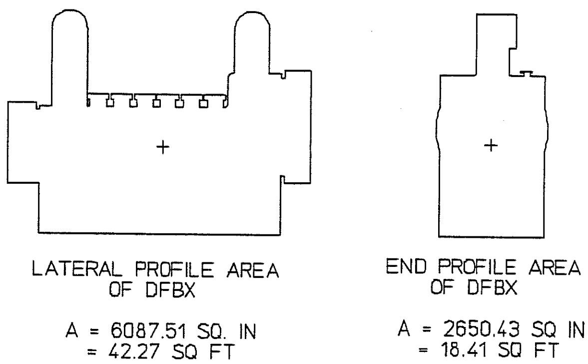
Ref: 1998 CALIFORNIA BUILDING CODE
Chapter 16, Div III sec1615- 1621.3,
Sec 1620, Table 16F, Table 16G, table 16H

DESIGN MANAGEMENT PROCEDURES MANUAL
RD3.22
Lateral Force Design Criteria
Pg.2 sec. 6

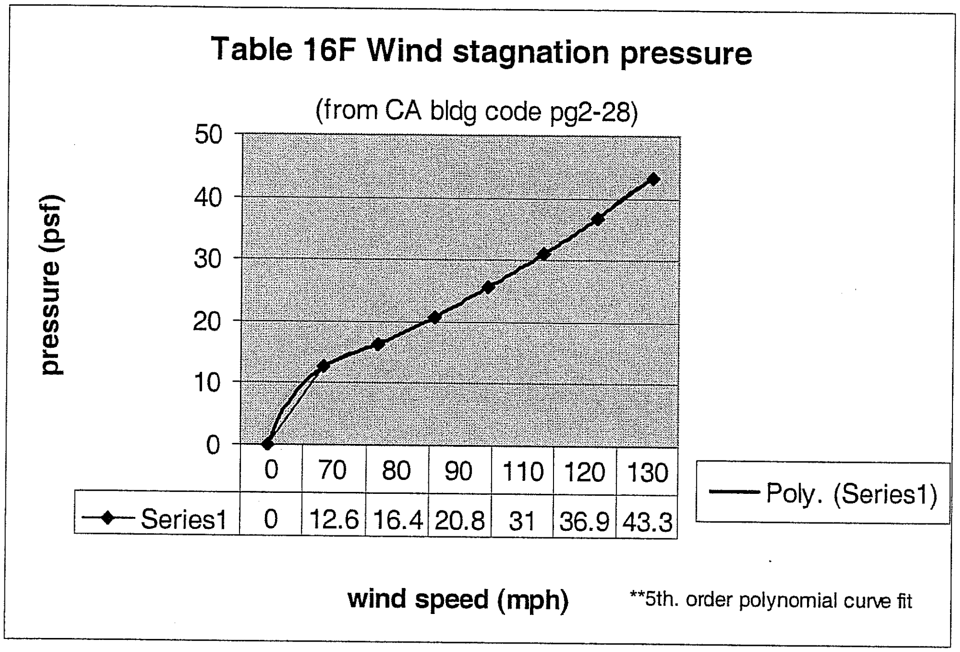
Sec 1620 – DESIGN WIND PRESSURES

Design wind pressures for buildings and structures and elements therein shall be determined for any height in accordance with the following formula.

a) Projected Lateral Surface Area of DFBX
From ME10 CAD Area measurement
LATERAL PROFILE AREA $A = 6087.51\ in^2 = 42.27\ ft^2$
END PROFILE AREA $A = 2650.43\ in^2 = 18.41\ ft^2$



Nomenclature:
P = Design Wind Pressures (psf)
A = Projected Lateral Surface area of DFBX (ft^2) = $5948.8\ in^2 = 41.31\ ft^2$
C_e= Combined height, exposure and gust factor coefficient = 1.39
C_q= Pressure Coefficient for Tanks (any directions) = 1.4
q_s= Wind Stagnation Pressure = $14.5\ lb/ft^2$ minimum value set by Facilities Dept.
(any structure must withstand minimum of 75 mph)
I_w= Wind Importance factor for function of the structure = 1.0
from table 16-k pg2-30



Lateral Design wind Force = $P_{wind} = C_e C_q q_s I_w A$

$= (1.39)(1.4)(14.5 lb/ft^2)(1)(42.27\ ft^2)$

$= (28.17\ lb/ft^2)(42.27\ ft^2)$

= 1,191 lb

End Design wind Force = $P_{wind} = C_e C_q q_s I_w A$

$= (1.39)(1.4)(14.5 lb/ft^2)(1)(18.41\ ft^2)$

$= (28.17\ lb/ft^2)(18.41\ ft^2)$
= 518.6 lb

Since any structure at the LBNL must withstand 75 mph wind load.
As result lateral wind load acting on DFBX= 1,191 lb and
End wind load acting on DFBX= 518.6 lb

This means that if 75 mph wind speed condition did occur without any gradient wind force through the side or end of the DFBX (otherwise it start spinning),

then suspended DFBX will move by:

From lateral wind:

$$\tan\theta = \frac{\text{wind load}}{\text{weight of DFBX}} = \frac{1{,}191\,\text{lb}}{13{,}000\,\text{lb}} = 0.092$$

$$\theta = 5.3^\circ \text{ tilt}$$

From end wind:

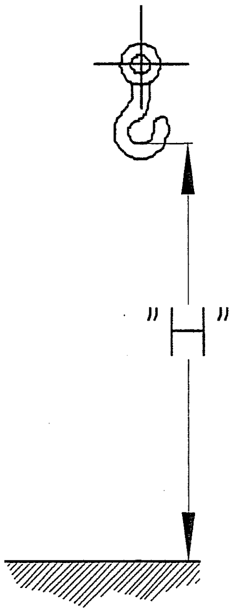
$$\tan\theta = \frac{\text{wind load}}{\text{weight of DFBX}} = \frac{518.6\,\text{lb}}{13{,}000\,\text{lb}} = 0.040$$

$$\theta = 2.3^\circ \text{ tilt}$$

Conclusion: If wind load plant engineering specified did occur hoist rings will not making contact with the DFBX vacuum tank to cause a damage.

10. LBNL BLDG 77A CRANE CAPAPACITY AND MAXIMUM HEIGHT

AUTHOR Yoichi Kajiyama | DEPARTMENT Mechanical Engineering | DATE 10/10/2001 | checked by Carol Corradi



LBNL BLDG 77A CRANE CAPACITY

NEASUREMENT WAS TAKEN BY DAVID ANDERSON 9/28/2001

BLDG 77	CAPACITY	CRANE HEIGHT (GROUND TO INNER RADIUS OF THE HOOK)	ROLLED UP DOOR
WEST END	15 TON	17 FT 1 IN = 205 IN = 5207 M	YES AT WEST END
CENTER	10 TON	12 FT 10.5 IN = 154.5 IN = 3.924 M	YES FACING THE ST
EAST END	10 TON	13 FT 9 IN = 165 IN = 4.191 M	NONE**

** TO MOVE OBJECT FROM EAST END
TO CENTER OR WEST END YOU
CAN NOT USE THE CRANE TO ACCOMPLISH
THE TASK YOU MUST MOVE BY GROUND
METHOD