P. DESIGN OF LIFTING AND TAILING LUGS

LIST OF SYMBOLS

A, d, G, H, J, K	=	shackle dimension
B	=	width of lug plate
C	=	centroidal distance
D	=	lug plate width at the pin.
D_B	=	bolt circle diameter
F	=	lift load at each lug
F_t	=	tailing lug load
f_v	=	vessel wall stress factor
L, L_1, L_2, L_3	=	dimensions of lug plate
N	=	number of bolts for the nozzle
S, P, E	=	dimensions of trunnion
T	=	lug thickness
t	=	lug plate thickness
t_1	=	collar plate thickness
t_v	=	thickness of vessel shell
t_w	=	weld size
W	=	weight of vessel
Z	=	section modulus
α	=	angle of vessel from horizontal
λ	=	stress ratio, σ_b/σ_t
σ_b	=	allowable bending stress
σ_p	=	allowable bearing stress
σ_s	=	allowable shear stress
σ_t	=	allowable tensile stress
σ_v	=	vessel stress
σ_w	=	allowable stress in the weld
$\sigma_{\theta 1}$, $\sigma_{\theta 2}$	=	bending stress in lug plate due to angle θ
θ	=	angle of lift cable from vert.

A. PURPOSE

This design guide describes methods of selecting and designing different types of lifting and tailing devices for use in the erection of pressure vessels.

B. CODES AND STANDARDS

Since lifting or tailing lugs are non pressure retaining structural members, vessel codes or standards are not applicable in the design of these lugs. Good engineering practice and the recommended procedures given in the AISC Steel Construction Manual are all that are needed.

C. TYPES OF LIFTING/TAILING LUGS

Besides slings and attachments improvised in the field, lifting lugs for towers, reactors and other pressure vessels can be broadly classified into (1) the ear type, (2) trunnions, and (3) the top nozzle blind type. See Fig. 1.

By far the most common are the ear type lifting lugs. These are usually installed at the top of the vessel and can be used for most vessels, especially large towers. Because of their location, interference with vessel appurtenances, such as platforms and ladders below the top head is not very critical. See Section 1.

When a tower is unusually tall, so that lifting it will require a large crane and/or a long boom, or will produce excessive bending stress in the vessel as a beam, the trunnion type lifting lugs may be more suitable. These are attached to the vessel shell some distance down from the top head. In using this type of lugs, the interference of the lifting cables with external ladders, platforms, nozzles, etc. attached to the vessel above the lugs, from the lying down to the up-right position of the tower should be carefully checked and avoided. See Section 2.

Most heavy wall vessels have a large and strong nozzle located at the center of their top head. With a special bolted attachment, this nozzle can be used for the erection of the vessel. This type of lifting blind is especially attractive when there are several vessels with the same size and rating top nozzles, so that one such lifting blind can be used on all of them. Even if the top nozzles were of different size, a special lifting blind can usually be designed to adapt it to more than one nozzle size. The advantages of the top nozzle lifting blind are economy, little interference with vessel appurtenances, and elimination of welding to the vessel which can cause stress risers that are undesirable in some critical vessels. See Section 3.

In uprighting a vessel from the horizontal to its vertical position, a pivot point, which is also capable of sliding toward the vessel foundation, is needed. This point can be provided simply and relatively inexpensively by a tailing lug. The design of tailing lugs is described in Section 4.

1. EAR TYPE LIFTING LUGS. (FIG. 2)

1.1 In designing a lifting lug, the first thing to do is to determine the lift load. Normally a vessel is lifted in the empty state, but in recent years, the trend has been to lift it fully dressed. That means the vessel will have insulation, some piping and all non-interfering ladders and platforms installed on it during lifting. Whatever the method of erection, determine the lift weight, W, multiply it by an impact factor, and then divide it by two to obtain the lift load, F, for each lug. The impact factor can vary from 1.25 to 2.0. The commonly used value is 1.50. Thus:

$$F=\frac{1.50W}{2}=0.75W \tag{1}$$

When the lift employs a tailing lug, the lift load is reduced in proportion to the relative distances of the lifting and tailing points from the center of gravity of the vessel. However, when the column is nearing its vertical position, the tailing device is going to be removed. Then the lifting load will be as calculated in Eq. (1).

1.2 In lifting, the field uses standard shackles with safe working load ratings determined from the minimum breaking loads with a safety factor of 5. Based on the lift load from Eq. (1), choose a shackle size from Table 1.

1.3 The dimensions of the shackle selected will determine the lug pin diameter, d, the lug thickness, T, at the pin joint, and the lug projection, L, to provide clearance between the shackle and the vessel. The lug pin diameter can also be determined based on its shearing strength as in §1.6 below.

1.4 Although the lug pin may be conservatively sized so that it can take some bending as well as shearing, it is best to keep the difference between the shackle dimension, G, and the lug thickness, T, to a minimum, usually from 3/32" to 3/16".

1.5 The width of the lug plate, D, at the pin is set equal to 3d minimum.

1.6 Lug Design (See Fig. 3 for different failure modes)

1. Determine lug pin diameter from either the commercial shackle size (1.3) or

$$\frac{2F}{\pi d^2}\leq\sigma_s \; ; \; d\geq\sqrt{\left(\frac{2F}{\pi\sigma_s}\right)} \tag{2}$$

2. Determine lug plate thickness t from

$$\frac{F}{3dt} \le \sigma_t \ ; \ t \ge \frac{F}{3d\sigma_t} \tag{3}$$

3. Find lug thickness. T = t + 2t_1, to suit the shackle .1.3) or from

$$\frac{F}{dT} \le \sigma_p \ ; \ T \ge \frac{F}{d\sigma_p} \tag{4}$$

4. If collar plates are not used, t_1 = 0, then t = T, and the lug pin may have to be checked for combined shearing and bending loads, especially where T< <G of the shackle.

5. Tensile and shearing modes of failure of the lug are not governing (see Fig. 3) unless the allowable tensile and shearing stresses are much smaller than the allowable bearing stress.

1.7 The minimum width, B at the lower end of the lug plate is given by

$$\frac{FL}{tB^2/6} \le \sigma_b \qquad \text{OR} \qquad B \ge \sqrt{\frac{6FL}{t\sigma_b}} \tag{5}$$

1.8 The combined tensile and bending stress in the lug plate should be checked when the vessel is lifted into a position where the axis is at an angle α from the horizontal. (See Fig. 4). If excessive, B should be increased to reduce this stress

$$\frac{F\sin\alpha}{tB\sigma_t} + \frac{6FL\cos\alpha}{tB^2\sigma_b} \le 1 \tag{6}$$

If σ_b is taken equal to $\lambda . \sigma_t$, then

$$\frac{F\sin\alpha}{tB\sigma_t} + \frac{6}{\lambda}\frac{FL\cos\alpha}{tB^2\sigma_t} \le 1$$

Combined stress, $$\sigma_{comb.} = \frac{F\sin\alpha}{tB} + \frac{6}{\lambda}\frac{FL\cos\alpha}{tB^2} \le \sigma_t$$

σ_{comb} is a maximum at a certain angle, α_m, such that

$$\frac{d}{d\alpha}(\sigma_{comb}) = \frac{F}{tB}\cos\alpha - \frac{6}{\lambda}\frac{FL}{tB^2}\sin\alpha = 0$$

Thus

$$\cos\alpha - \left(\frac{6L}{\lambda B}\right)\sin\alpha = 0$$

Therefore,

$$\alpha_m = \tan^{-1}\left(\frac{\lambda B}{6L}\right)$$

$$If \quad \sigma_B = \sigma_t \;, \lambda = 1.0 \quad then\ \alpha_m = \tan^{-1}\left(\frac{B}{6L}\right)$$

(7a)

$$If \quad \sigma_b = 1.5\sigma_t \;,\; \lambda = 1.5, \quad then\ \alpha_m = \tan^{-1}\left(\frac{B}{4L}\right)$$

(7b)

1.9 In calculating the lift load, F, the eccentricity of the tailing load has to be considered.

From Fig. 5

$$2F = \frac{W(l_2\cos\alpha + l_3\sin\alpha)}{l_1\cos\alpha + l_2\cos\alpha + l_3\sin\alpha} \tag{8}$$

$$F_{max} = \frac{W}{2}(Impact\ \ factor)$$

1.10 Stress in the weld for direct load, F, is

$$\frac{F}{0.7\,(B+2L_3)\,t_w} \le \sigma_w \tag{9}$$

Centroidal distance, C, of the weld is

$$C=\frac{L_3^2}{2L_3+B}$$

Moment about the weld centroid = F (L + L_3 -C)

Replacing the U shape weld conservatively with an approximate rectangular shape weld, where L_3 = 2C,

then: Polar moment of inertia ≈ 1.4 t_w BC (C+½B)

$$Polar\ section\ modulus = \frac{1.4t_w BC(C+\frac{1}{2}B)}{\sqrt{C^2+B^2/4}}$$

Stress due to moment (10)

$$\frac{F(L+L_3-C)\sqrt{C^2+B^2/4}}{1.4BC(C+B/2)t_w}\leq\sigma_w$$

From the combination of stresses (9) & (10), the weld size t_w can be determined.

If L_3 is made equal to B, the above expressions can be simplified:

$$C=\frac{B}{3}$$

$$Moment\ =F(L+B-\frac{B}{3})=F(L+0.667B)$$

$$I=1.4(B)(\frac{B}{3})(\frac{B}{3}+\frac{B}{2})t_w=0.389B^3t_w$$

$$Z=\frac{0.389B^3t_w}{\sqrt{B^2/9+B^2/4}}=0.647B^2t_w$$

$$\sigma=\frac{F}{0.7\,(B+2B)\,t_w}+\frac{F(L+0.667B)}{0.647B^2 t_w}=\frac{F}{B^2 t_w}\,(1.543L+1.507B)$$

Using an allowable stress for the weld σ_w = 13500 psi, we have:

$$t_w\geq\frac{F}{B^2}\,(0.114L+0.112B)\,10^{-3} \qquad (11)$$

Table 2 shows weld size, t_w, per 1000 lb. of lift load, F, for various ratios of L/B.

TABLE 2 - WELD SIZE t_w INCH PER 1000 LB. LIFT LOAD

L/B \ B	3"	6"	9"	12"	15"	18"	21"	24"
0.75	0.066	0.033	0.022	0.016	0.013	0.011	0.009	0.008
1.0	0.075	0.038	0.025	0.019	0.015	0.013	0.011	0.009
1.25	0.085	0.042	0.028	0.021	0.017	0.014	0.012	0.011
1.5	0.094	0.047	0.031	0.024	0.019	0.016	0.013	0.012
1.75	0.104	0.052	0.035	0.026	0.021	0.017	0.015	0.013
2.0	0.113	0.057	0.038	0.028	0.023	0.019	0.016	0.014
2.25	0.123	0.061	0.041	0.031	0.025	0.020	0.018	0.015
2.5	0.132	0.066	0.044	0.033	0.026	0.022	0.019	0.017
2.75	0.142	0.071	0.047	0.035	0.028	0.024	0.020	0.018
3.0	0.151	0.076	0.050	0.038	0.030	0.025	0.022	0.019

1.11 In order to keep the lifting cable vertical, a spreader bar is used. When the spreader bar is not available or not to be used, then the cables will make an angle with the axis of the vessel. This will produce out-of-plane bending of the lug plate. If the cable angle is Θ°, the corresponding bending stress in the lug plate is

$$\sigma_{\Theta_1}=\frac{6FL\sin\Theta 1}{Bt^2} \qquad (12)$$

The maximum allowable angle to ensure $\sigma_{\Theta 1} \leq \sigma_b$ is

$$\Theta_m=\sin^{-1}\left(\frac{Bt^2\sigma_b}{6FL}\right) \qquad (13)$$

To reduce this bending stress, a bracing plate may be used tying the lug plate to the vessel head at a distance L_1 from the lug pin. The reduced bending stress is now

$$\sigma_{\theta_2} = \left(\frac{L_1}{L}\right) \sigma_{\theta_1} \tag{14}$$

The load on this bracing plate, F sin θ, is rather small for small angles, θ. Therefore its thickness, t_2, and the weld sizes usually are minimal. For very large angles of θ, this plate should be properly designed.

2. TRUNNIONS (Fig. 6)

Trunnions are used to erect vessels that are too tall to be lifted by lugs located at the top head. There are generally 3 different types of trunnions. The most common is the fixed pipe, fixed plate type. Sometimes, for clearance purposes, the lug plate has to be made very long. In order to prevent the high twisting moment at the pipe-vessel attachment, the lug plate is allowed to turn over the lug pipe. When the lug pipe projection has to be large, again for clearance purposes, the bending moment at the vessel attachment may be very high. This will produce excessive stress and distortion in the shell, especially in large diameter, thin wall vessels. In this case, a turning axle type trunnion can be used.

2.1 As for the ear type lifting lug in Sectior :1, determine the lift load, shackle size, pin diameter and lug plate thickness.

2.2 For the turning plate trunnion, the dimension, s, is determined by

$$\frac{F}{2st} \le \sigma_t \quad \text{or} \quad s = \frac{F}{2t\sigma_t} \tag{15}$$

2.3 The trunnion pipe size is determined by either its bending strength:

$$\frac{FE}{Z} \le \sigma_b \tag{16}$$

or its torsional resistance:

$$\frac{FL}{2Z} \le \sigma_s \tag{17}$$

or its shear resistance:

$$\frac{F}{A_p} \le \sigma_s \tag{18}$$

where A_p = cross-sectional area of the lug pipe

In addition:

$$\frac{FL}{2Z}+\frac{F}{A_p}\leq\sigma_s \tag{19}$$

2.4 For the rotating axle type trunnion, the pipe diameter, P, is determined from:

$$\frac{F}{Pt_v}\leq\sigma_p \tag{20}$$

2.5 Weld size, t_w, is determined from

$$\frac{F}{0.7\pi Pt_w}=\frac{0.455F}{Pt_w}\leq\sigma_w \tag{21}$$

or from

$$\frac{FL}{0.7t_w(\pi\frac{P^2}{2})}=\frac{0.91FL}{P^2t_w}\leq\sigma_w \tag{22}$$

or from

$$\frac{FE}{\pi(\frac{P^2}{4})(0.7t_w)}=\frac{1.82FE}{P^2t_w}\leq\sigma_w$$

$$\frac{0.455F}{Pt_w}+\frac{0.91FL}{P^2t_w}\leq\sigma_w \tag{23}$$

2.6 Except in the case of the turning axle trunnion, the lift load on the trunnion produces local stresses in the vessel shell. These need to be evaluated using WRC Bulletin 107 or other similar methods. A simplified way to check the vessel stresses is: (Fig. 7 Solid Lines).

$$f_v(\frac{FE}{t_v^3})\leq\sigma_v \tag{24}$$

In checking the stress of the vessel in the vertical position, when the load F represents the entire vessel weight not reduced by any tailing load, use the stress factor given by the dashed lines in Fig. 7.

When the vessel wall stress is excessive, use a reinforcing pad. The stress is dependent on the combined vessel and pad thicknesses at the edge of the trunnion pipe and on the vessel thickness at the edge of the pad.

2.7 After a vessel has been erected, the trunnion lifting lugs are usually cut off. In the case of the turning axle type, the lug is cut apart and the axle withdrawn. The openings in the vessel are covered by welded plates or bolted flanges.

3. TOP LIFTING BLINDS

Heavy wall vessels have sturdy nozzles. If there is a large central nozzle in the top head, it can usually be utilized to lift the vessel into place. A specially designed lifting blind is bolted to the top nozzle, and lug plates are welded to the blind for attachment of the lifting shackle. When there are several vessels with the same size top nozzles, one lifting blind can be made to lift all of them. Even if the top nozzles are of different size and/or rating, the lifting blind can be designed with different sets of bolt holes to fit more than one size of nozzles.

3.1 In lifting, usually not all the bolts for the top nozzle are used. If only half of the bolts are used, the load on each bolt due to a moment is approximately 22% more than if all the bolts are utilized. In checking the adequacy of the botls, the following conditions have to be satisfied:

$$\frac{4.9F_1(E+t_b)}{ND_B} \leq \quad \text{Bolt capacity in tension} \qquad (25)$$

$$\frac{2F_1}{N} \leq \quad \text{Bolt capacity in shear} \qquad (26)$$

$$\frac{2F_2}{N} \leq \quad \text{Bolt capacity in tension} \qquad (27)$$

Where F_1 = max. load on the lug when lifting angle $\alpha = 0$ (vessel in horizontal position)

F_2 = lifting load when vessel is in the vertical position = W. impact factor.

E = distance from lug pin center line to blind flange back face. See Fig.8.

t_b = thickness of blind flange.

N = total number of bolts for the nozzle.

3.2 Design of the lug plate, that is to be welded to the blind, follows the same procedure as for the ear type lug.

3.3 A conservative way to determine the thickness of the blind is to design it for the pressure equivalent to the max. moment acting on it at the initial lifting condition. This moment equals $F_1(E+t_b)$ and the equivalent design pressure is:

$$P_{eq}=\frac{16F_1(E+t_b)}{\pi G_k^3}$$

where G_k = gasket diameter of the flange.
and the blind thickness is:

$$t_b=G_k\sqrt{\left(\frac{0.3P_{eq}}{\sigma_b}\right)}=G_k\sqrt{\left(\frac{4.8F_1(E+t_b)}{\pi G_k^3\sigma_b}\right)}$$

By assuming an initial value of 2" for t_b and using the bolt circle diameter D_B as G_k, we have

$$t_b=\sqrt{\left(\frac{1.5F_1(E+2)}{D_B\sigma_b}\right)} \tag{28}$$

If the calculated t_b is much different from 2", another iteration may be done. In any case, do not use t_b less than the lug plate thickness determined in 3.3.2 above. The equivalent pressure due to the force F_2 on the lug is relatively insignificant.

3.4 Examples of one lifting blind fitting two sizes of nozzles are shown in Fig. 9.

4. TAILING LUGS

In lifting a tall vessel, whether by lifting lugs attached near the top head, or by trunnions located lower down the shell, a tailing lug is usually required to lift the entire column off the ground in order to facilitate the uprighting of the vessel. Since most vertical vessels have skirt supports, the tailing lug is usually attached to the bottom of the vessel to take advantage of the stiffness of the base ring there. Unless the tailing load is unusually large, only one lug is required. See Fig. 10.

4.1 Design of the lug plate again follows the same procedure as for the ear type.

4.2 Check stresses in the base ring-skirt section

Max. moment = .2387 F_tR

where F_t = load on the tailing lug.

Generally the skirt plate is welded approximately to the middle of the base ring. It does not therefore contribute much to the section modulus of the section. Thus the bending stress in the base ring during the lift is

$$\sigma = \frac{1.44 F_t R}{t_{b_r} (B_{b_r})^2}$$ where t_{br} and B_{br} = thickness and width of the base ring.

When this stress is too high, a strut can be welded diametrally between the tailing lug and the other side of the skirt. The force on the strut is approximately half the tailing load, and the moment in the base ring section will be reduced to about a third of that without the strut. See the derivation in Appendix A.

In the case when the skirt plate is not centrally located on the base ring, a portion of the skirt plate will act to resist the bending moment in the ring section, will increase the section modulus of the base ring appreciably, and should therefore be taken into consideration. If a second ring also exists, then the section modulus will be further substantially increased. The length of the skirt plate that can be included in the base ring section, varies with the skirt diameter and its thickness, but for simplicity, use conservatively a length equal to 12 times its thickness. If a second ring exists, then this length shall be taken beyond the second ring.

D. MATERIALS

Lifting and tailing lugs are structural elements that are used only for a short time during the erection of vessels. For this reason, the most common structural steel A-36 is adequate. However, parts of the lugs that are welded directly to vessels which are constructed of alloy metals, such as Cr.Mo. steel, stainless steel, no-ferrous metal, etc., should preferably be of the same type of material as the vessel.

Forgings and pipes will also be carbon steel: A-105 and A-53-B respectively, except as noted above.

E. ALLOWABLE STRESSES

Unless otherwise specified, use the following allowable stresses

A-36, A-105, A-53	Tensile - 20,000 psi Bearing - 30,000 psi Shear- 13,500 Bending - 22,000 psi
C.S. Welds:	Full penetration - 20,000 psi Shear - 13,500 psi
Vessel Stress:	Membrane - 1.2S Bending - 1.5S where S = code allowable tensile design stress.

F. RECOMMENDED PROCEDURE

1 Determine from Construction, the type of lifting to be used: from the top of the vessel or lower down the shell; whether the vessel will be bare or insulated with platforms and ladders installed; whether a spreader beam will be used; whether a large sturdy nozzle will be available on the top head for lifting, etc. Then choose the appropriate lugs to be designed.

.2 Calculate the lift load at each lifting lug, and if tailing lug is going to be used, calculate the lift and tailing loads. See paragraph 1.1.

3 Design the lugs per sections ,1. ;2. .3, or .4.

.4 Forward the lug design to the fabricator of the vessel on which the lugs are to be installed.

5 When design of the lifting and tailing lugs is the responsibility of the vessel fabricator, use this Design Guide to check the latter's calculations.

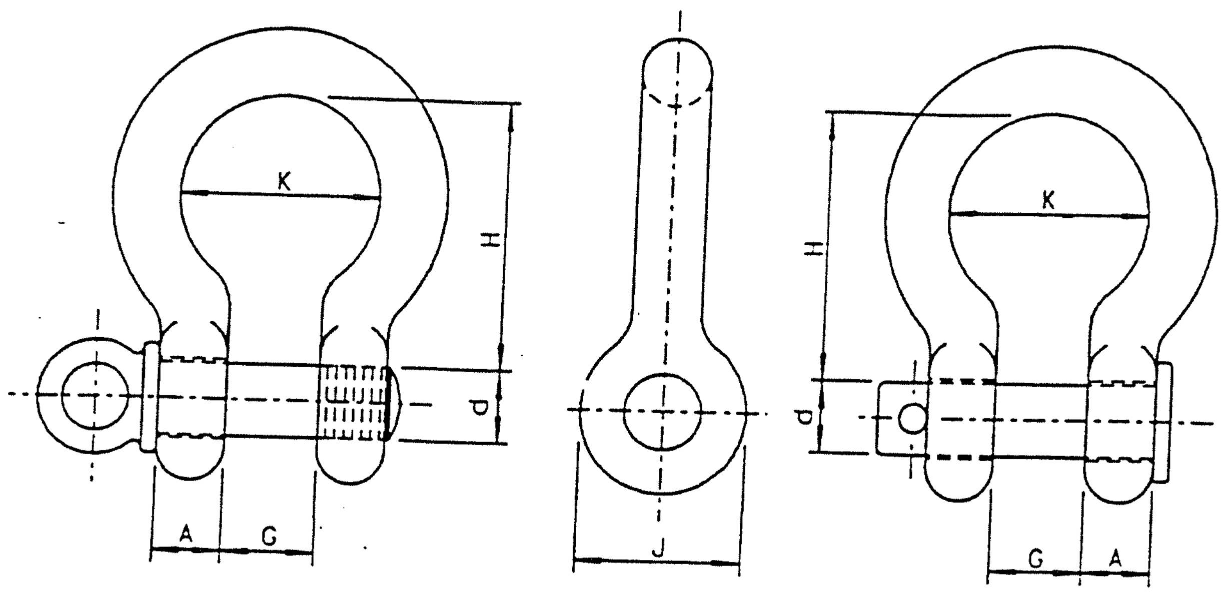
.6 After erection, remove the lugs from the vessel as necessary, and patch up the openings where the rotating axle type trunnion has been used.

G. REFERENCES

Formulas for Stress and Strain by R. J. Roark

Local Stresses in Spherical and Cylindrical Shells due to External Loadings, WRC Bulletin 107, by K. R. Wichman, A. G. Hopper and J. L. Mershon

TABLE 1



ANCHOR SHACKLES

(DROP FORGED STEEL, WELDLESS, BRIGHT, HEAT TREATED)

Size Inches	Min. Breaking Load, Tons	Safe Working Load, Tons	Shackle Dimensions, inches						Tolerance (±)		Est. Weight lbs.
			A	G	d	H	J	K	Length	Width	
3/8	3.97	.8	3/8	21/32	7/16	1-7/16	31/32	1-1/32	1/8	1/16	.3
7/16	5.42	1.08	7/16	23/32	1/2	1-11/16	1-1/16	1-5/32	1/8	1/16	.49
1/2	7.07	1.41	1/2	13/16	5/8	1-7/8	1-5/16	1-5/16	1/8	1/16	.74
5/8	11.05	2.21	5/8	1-1/16	3/4	2-13/32	1-6/19	1-11/16	1/8	1/16	1.44
3/4	15.9	3.18	3/4	1-1/4	7/8	1-17/32	1-7/8	2	1/4	1/16	2.16
7/8	21.62	4.32	7/8	1-7/16	1	3-5/16	2-1/8	2-9/16	1/4	1/16	3.37
1	28.27	5.65	1	1-11/16	1-1/8	3-3/4	2-3/8	2-11/16	1/4	1/16	5.26
1-1/4	41.25	8.25	1-1/4	2-1/32	1-3/8	4-11/16	3	3-1/4	1/4	1/8	9.55
1-3/8	49.9	10	1-3/8	2-1/4	1-1/2	5-1/4	3-5/16	3-5/8	1/4	1/8	12.57
1-1/2	59.35	11.87	1-1/2	2-3/8	1-5/8	5-3/4	3-5/8	3-7/8	1/4	1/8	17.25
1-3/4	80.8	16.17	1-3/4	2-7/8	2	7	4-1/8	5	1/4	1/8	27.75
2	105.55	21.25	2	3-1/4	2-1/4	7-3/4	5	5-3/4	1/4	1/8	41.12
2-1/4	135	27.	2-1/4	3-3/4	2-1/2	9-1/4	5-1/4	6-1/2	3/4	1/8	58.50
2-1/2	169	33.8	2-1/2	4-1/8	2-3/4	10-1/2	6	7-1/4	3/4	1/8	83.50
2-3/4	200	40.4	2-3/4	4-1/2	3	11-1/2	6	7-1/4	3/4	1/8	115.
3	230	48.5	3	5	3-1/4	13	6-1/2	7-7/8	3/4	1/8	145.

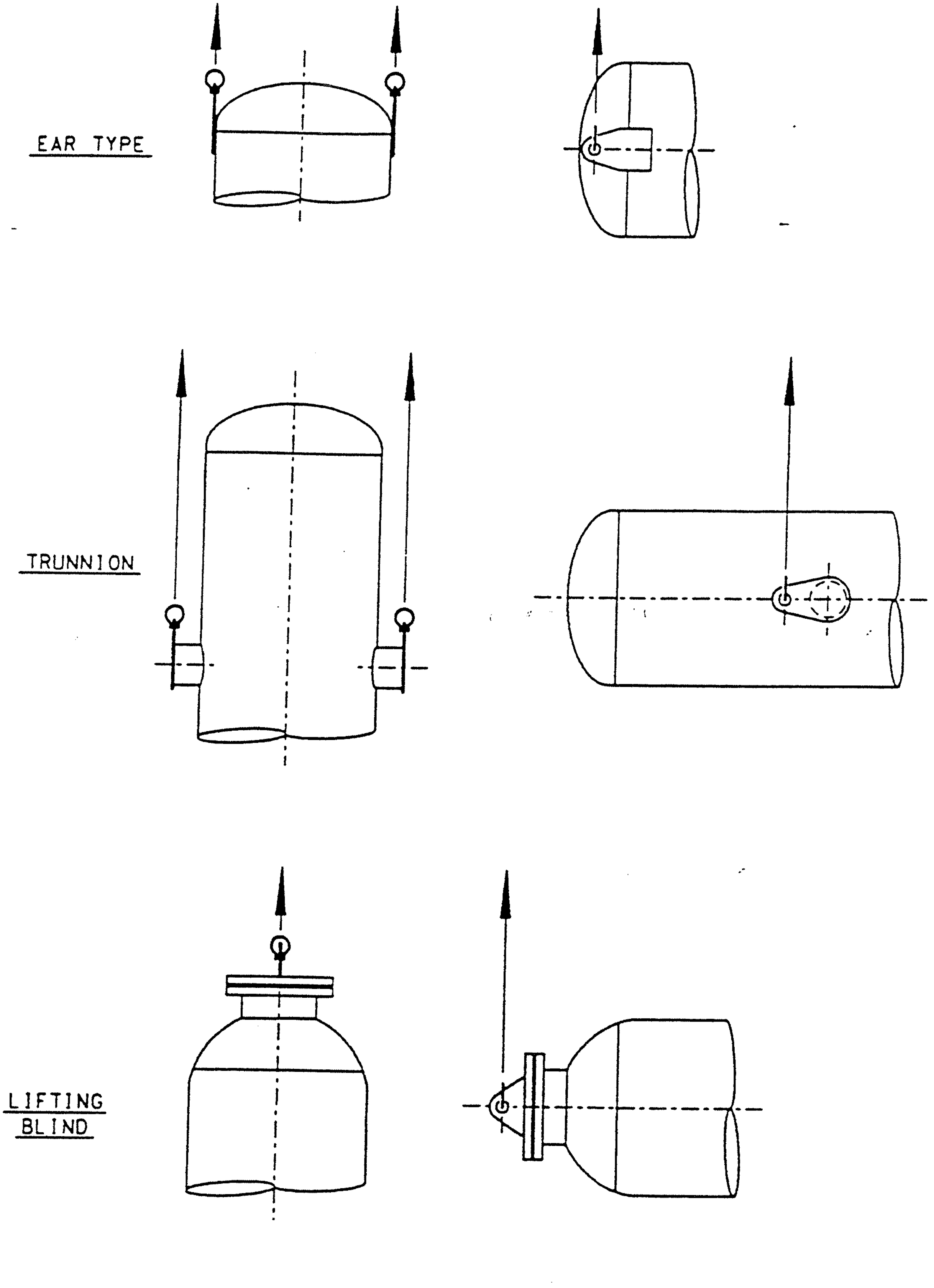


FIGURE 1——LIFTING LUGS

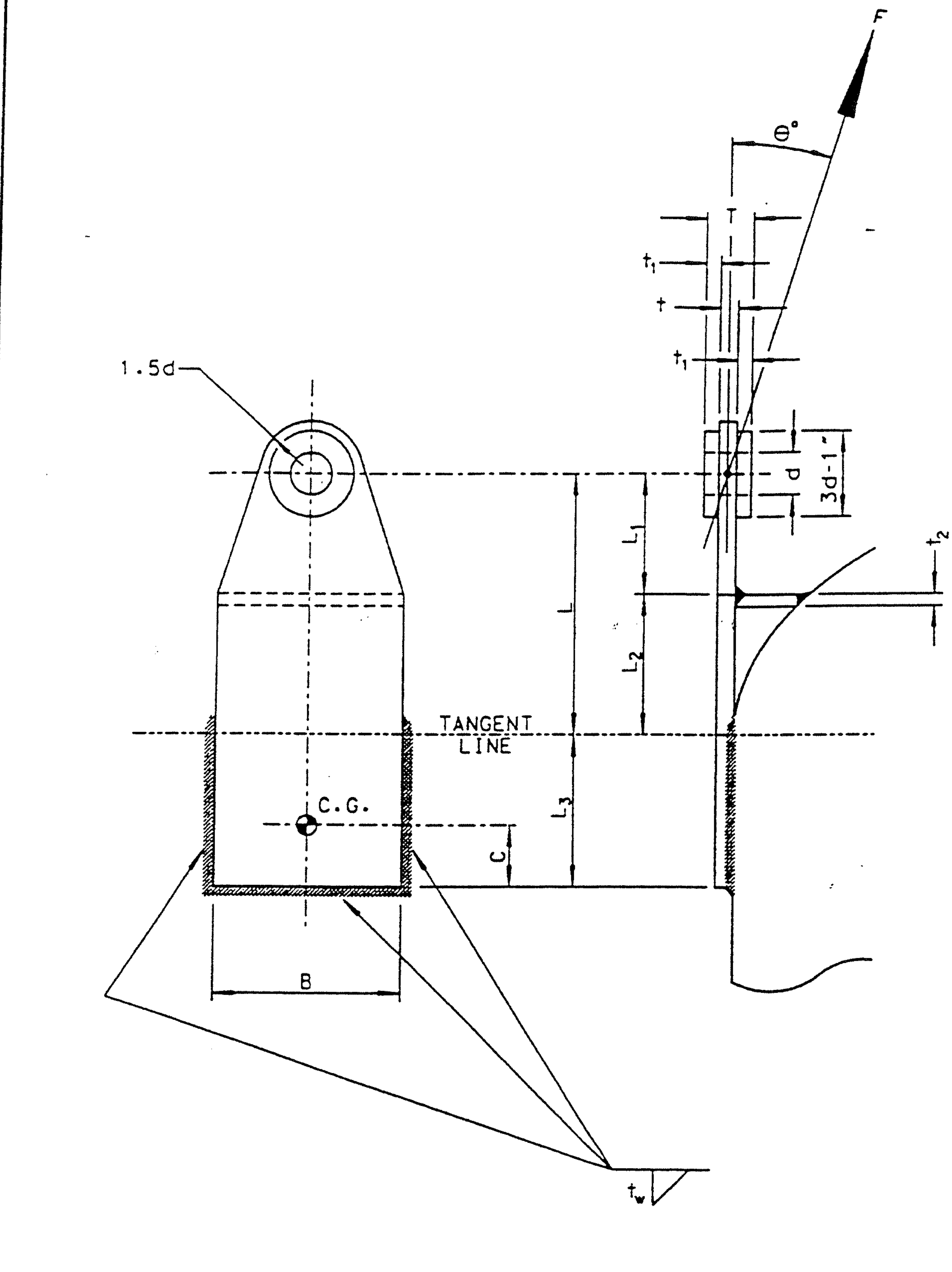
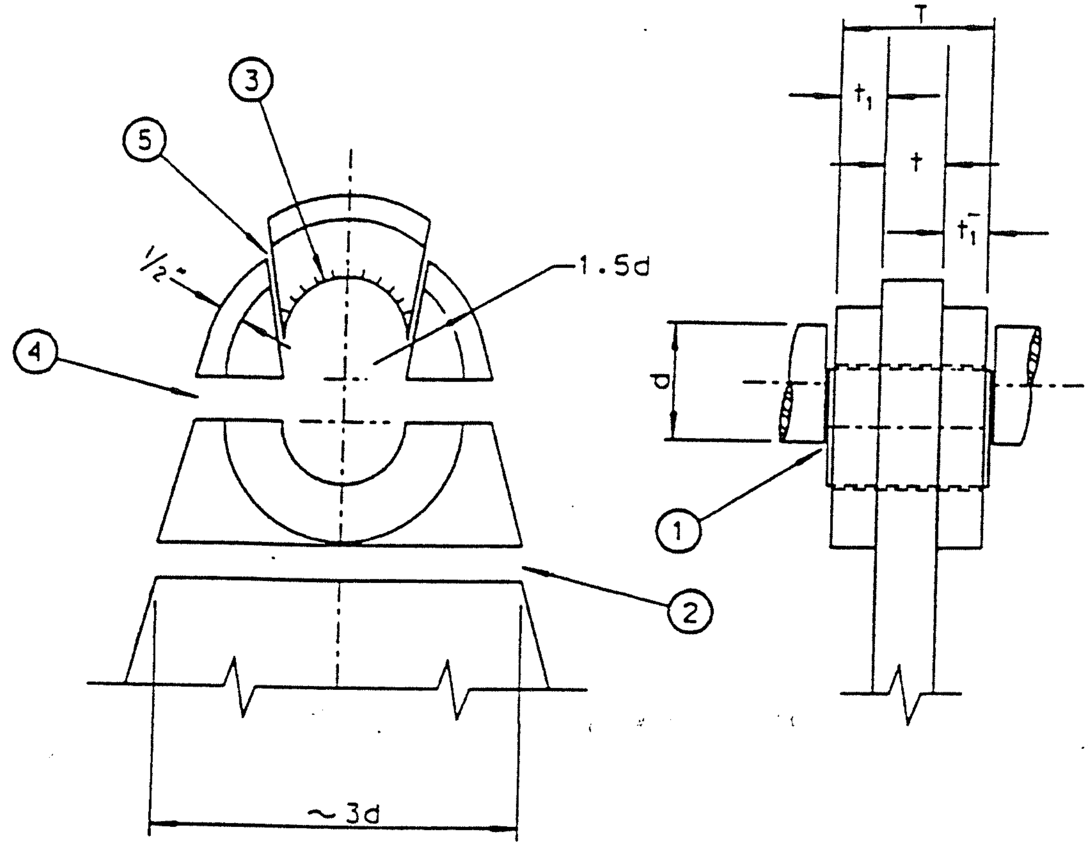


FIGURE 2——EAR TYPE LIFTING LUG



1. LUG PIN SHEARING $\frac{F}{2(\pi d^2/4)} = \frac{F}{d}\left(\frac{2}{\pi d}\right)$
2. LUG PLATE TENSILE $\frac{F}{3d(t)} = \frac{F}{d}\left(\frac{1}{3t}\right)$
3. LUG PIN BEARING $\frac{F}{d(t+2t_1)} = \frac{F}{d}\left(\frac{1}{t+2t_1}\right)$
4. LUG TENSILE $\frac{F}{2d(t+2t_1)} = \frac{F}{d}\left(\frac{1}{2(t+2t_1)}\right)$
5. LUG SHEARING $\frac{F}{2 \times 1.5d(t+2t_1)} = \frac{F}{d}\left(\frac{1}{3(t+2t_1)}\right)$

FIGURE 3——LIFTING LUG DESIGN

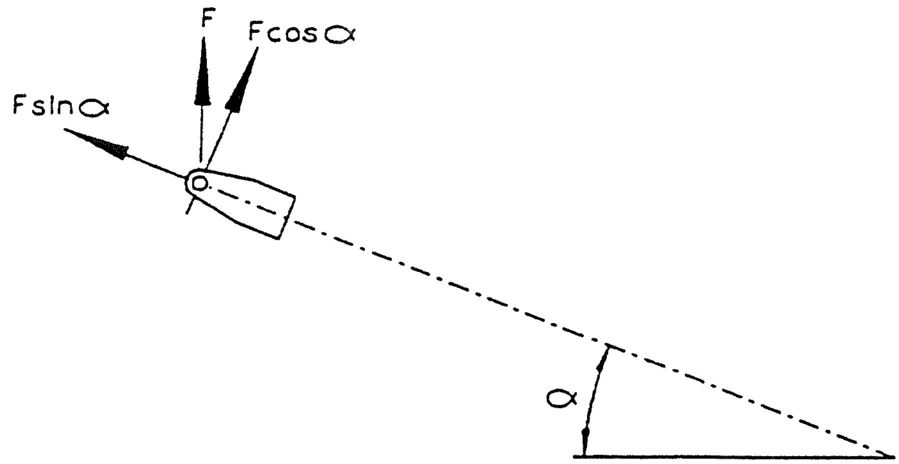


FIGURE 4——ANGLE OF LIFT

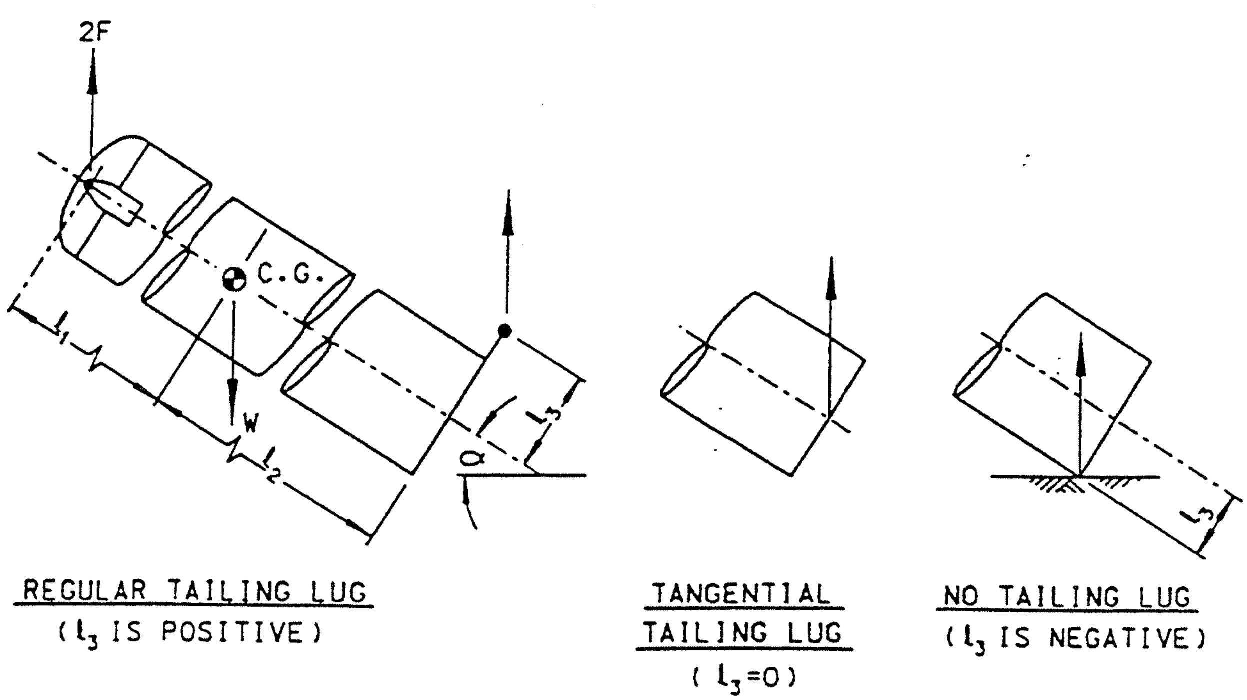


FIGURE 5——LOCATION OF TAILING LUG

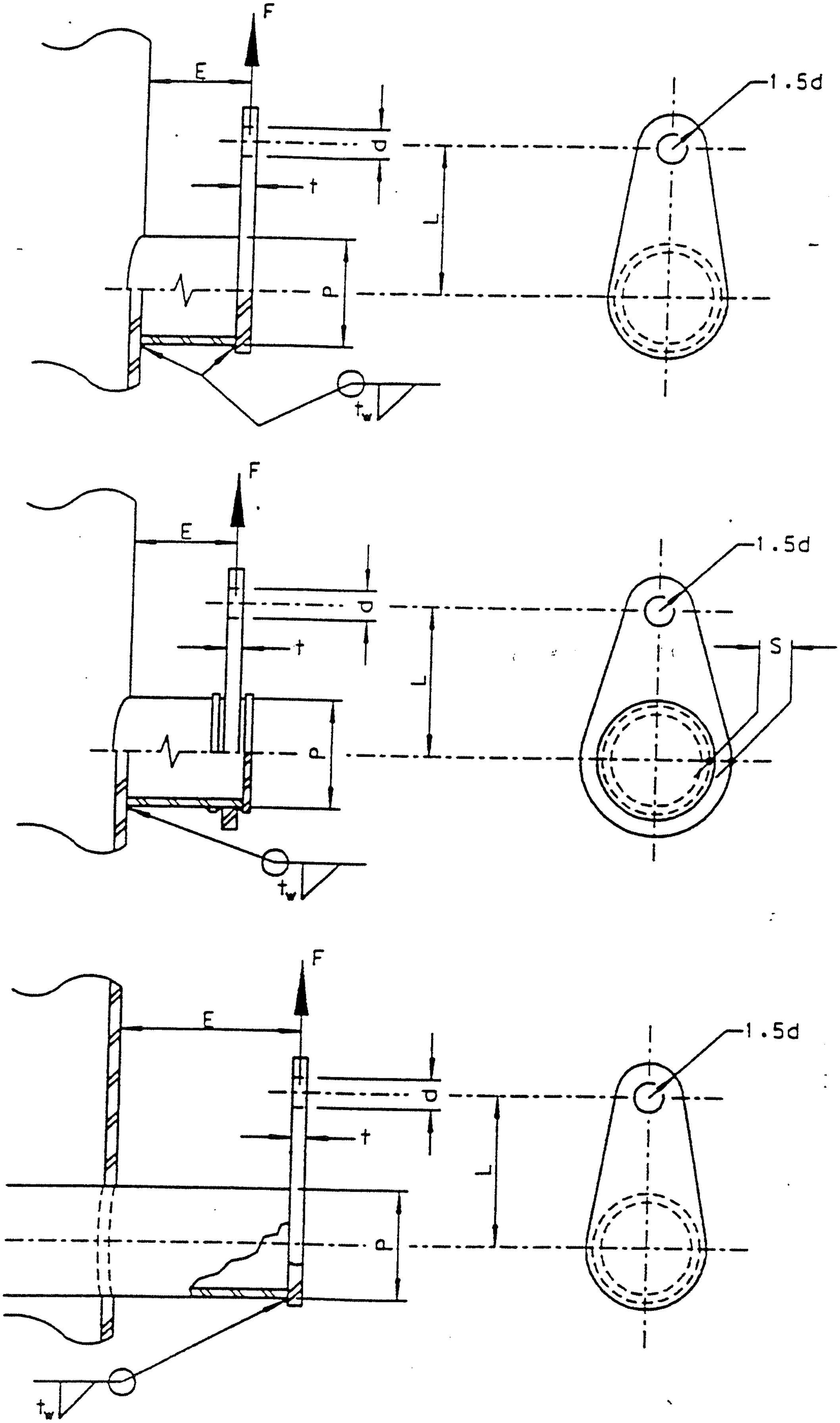


FIGURE 6——TRUNNIONS

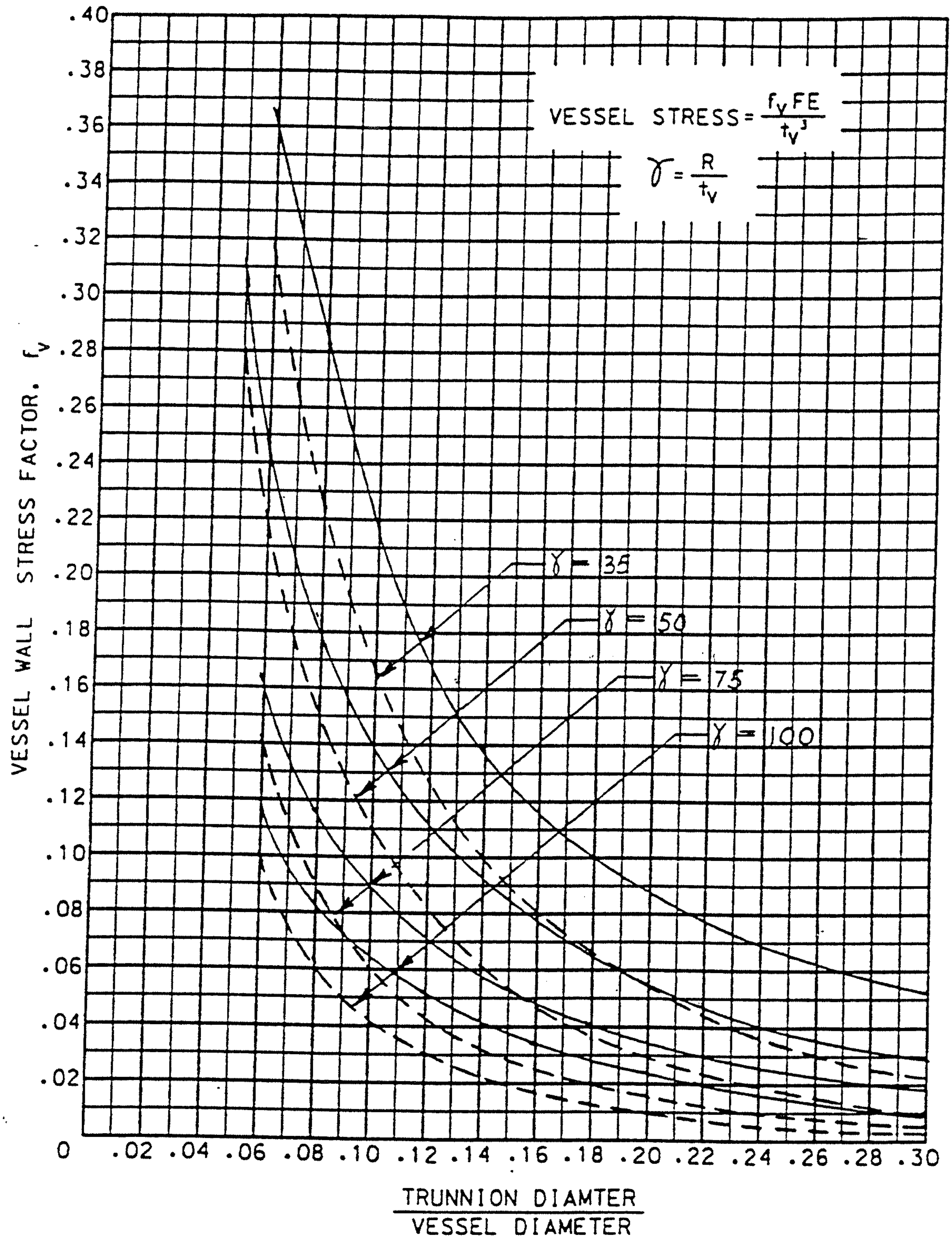


FIGURE 7——VESSEL STRESS FACTOR

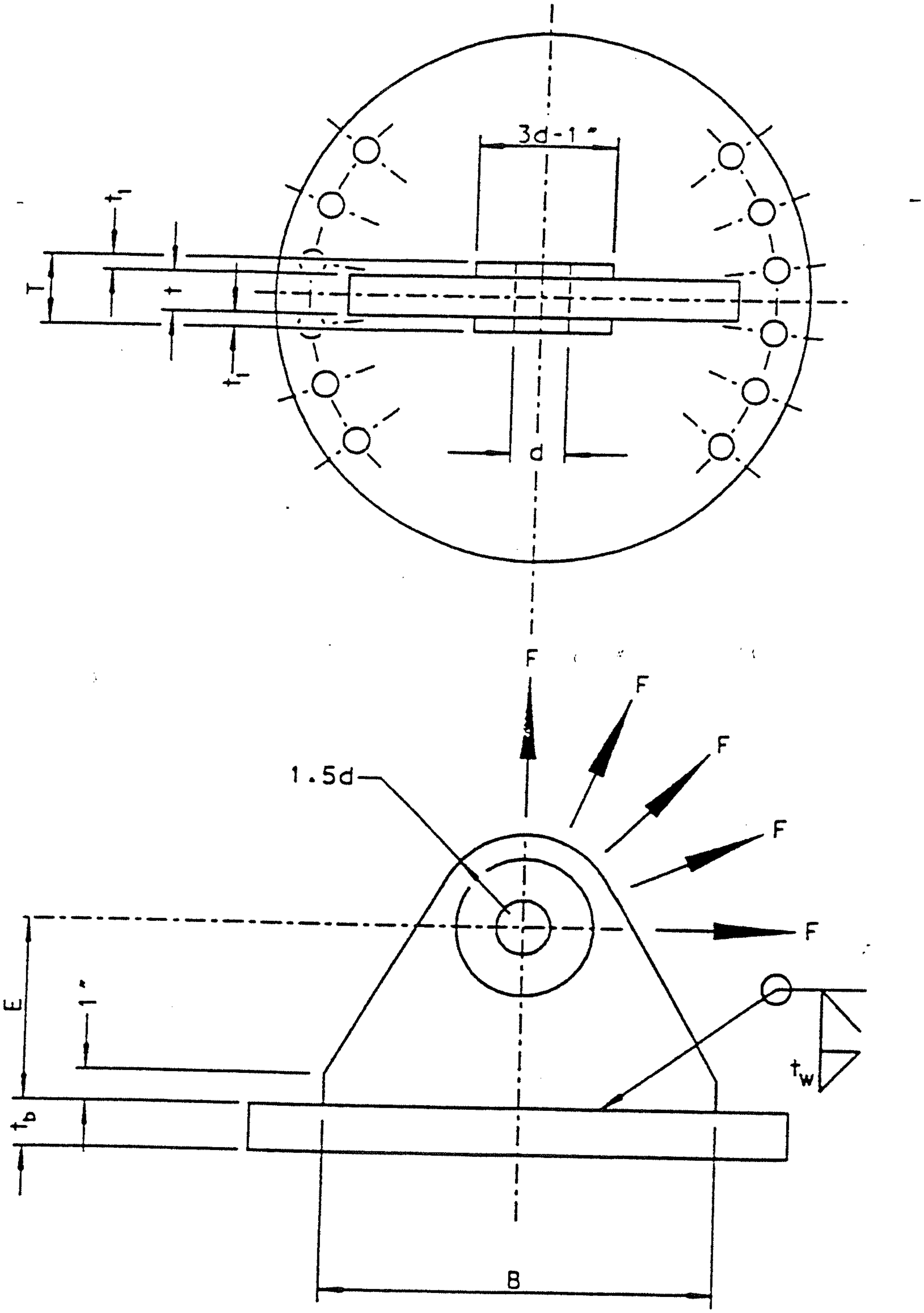
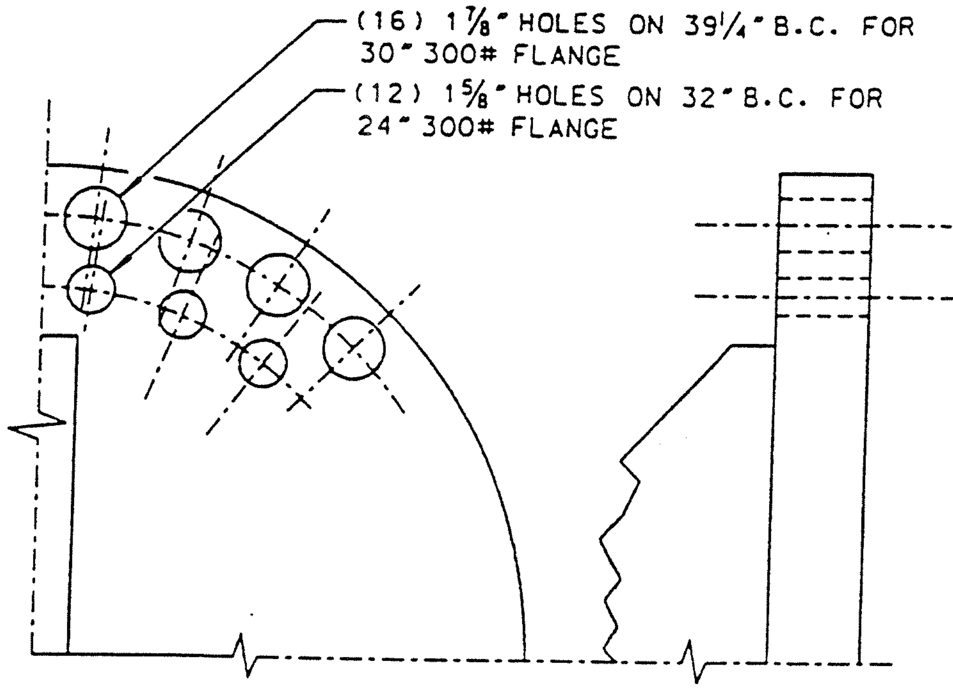


FIGURE 8——LIFTING BLIND



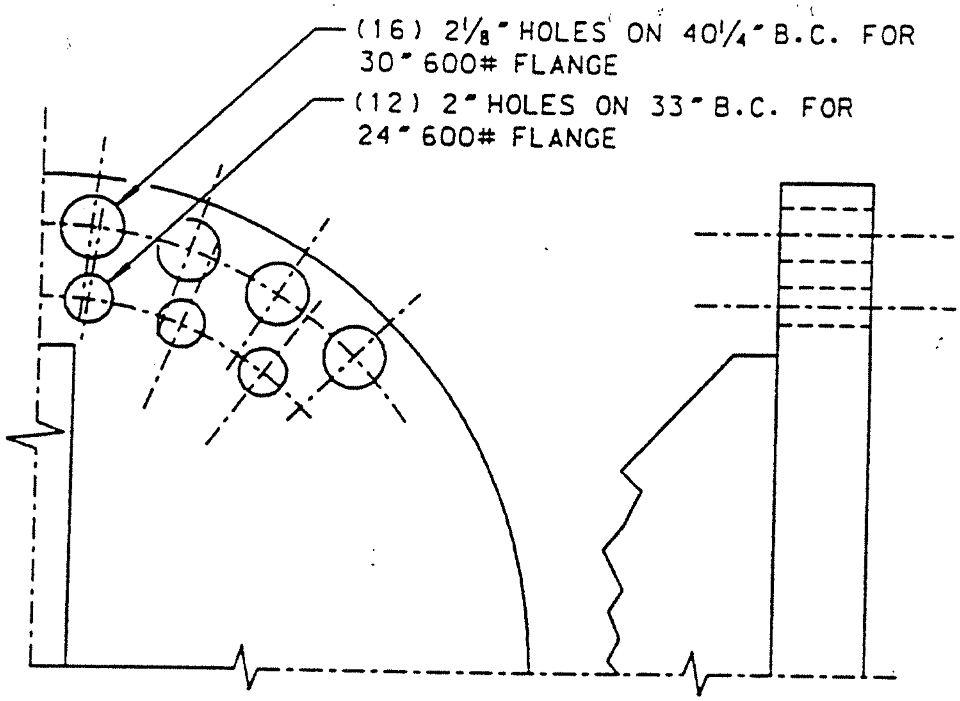


FIGURE 9——LIFTING BLIND BOLT PATTERNS

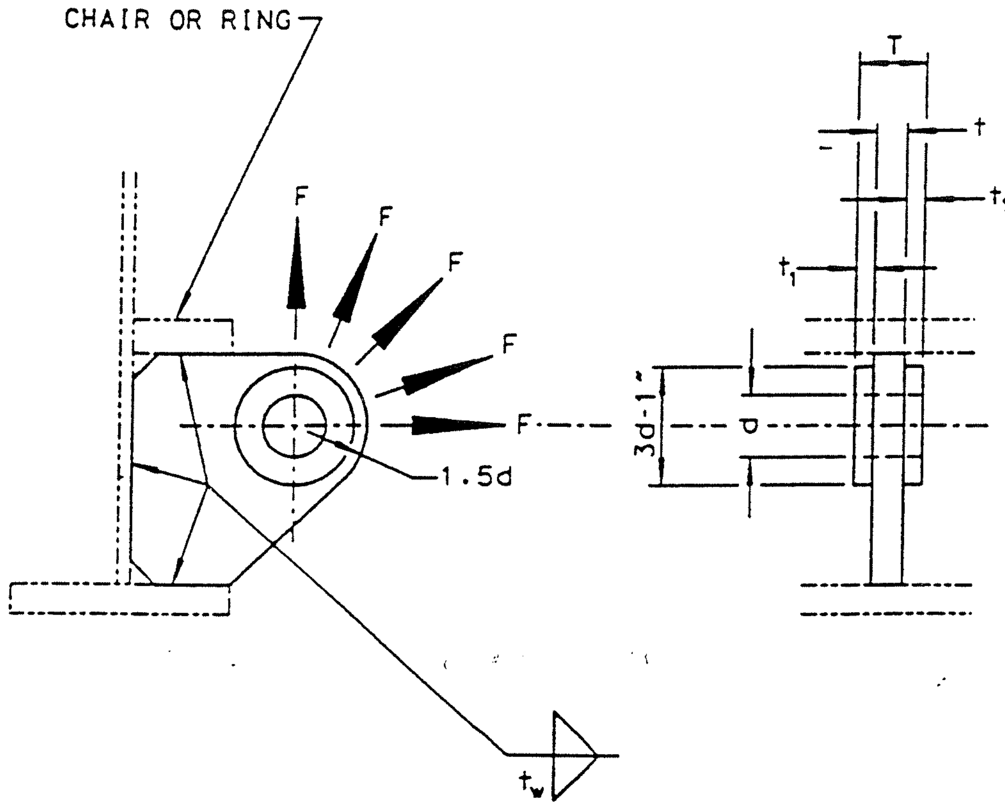
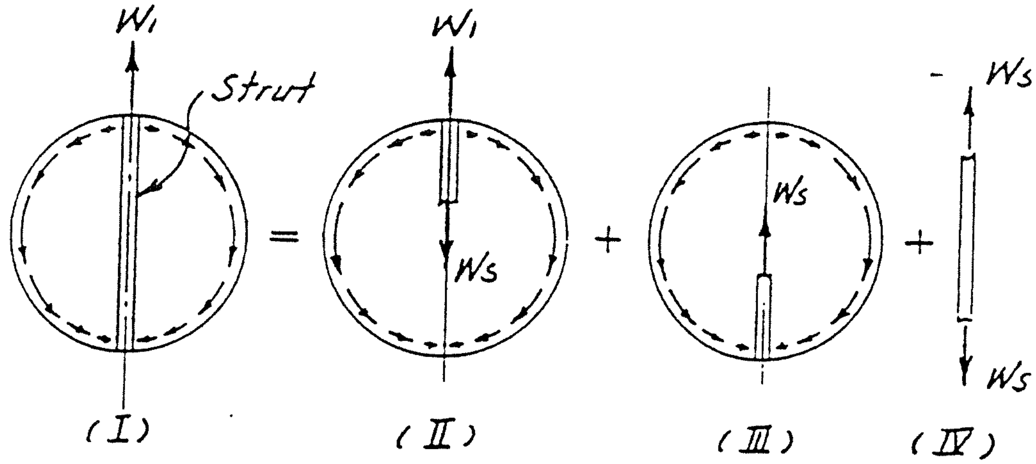


FIGURE 10——TAILING LUG

Derivation of Ring Moment



(II) Increase in diameter due to load $W_1 - W_s$ is:

$$D_y = \frac{R^2(W_1-W_s)}{EI}\left[.3183\,(s\theta + c) - \frac{\pi}{8}\right] = \frac{-.0744\,(W_1 - W_s)R^3}{EI}$$

(III) Decrease in diameter due to load W_s is:

$$D_y = \frac{-.0744\,W_s R^3}{EI}$$

(IV) Elongation of Strut due to load W_s is

$$\Delta l = \frac{2W_s R}{AE}$$

$$\frac{.0744\,(W_1 - W_s)R^3}{EI} - \frac{.0744\,W_s R^3}{EI} = \frac{2W_s R}{AE}$$

$$\frac{.0372\,W_1 R^2}{I} = \frac{W_s}{A} + \frac{.0744\,W_s R^2}{I}$$

The first term on the right side is small compared to the second term, and can be neglected.

$\therefore\ W_1 = 2W_s \quad$ or $\quad W_s = \frac{1}{2}W_1$

From Roark,

Moment in the ring due to outward load is.

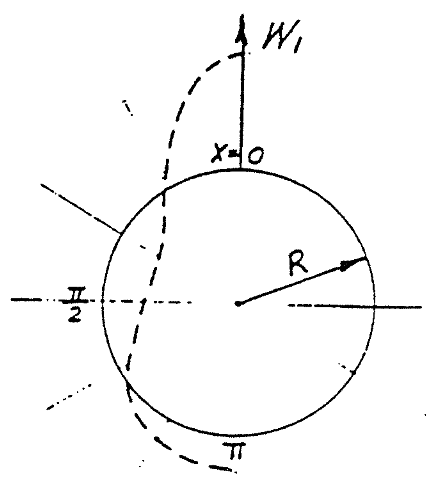
$$M_1 = (W_1 - W_5)R[.2387 \cos X - .5 \sin X + .1592(X \sin X + 1 - \cos X)]$$

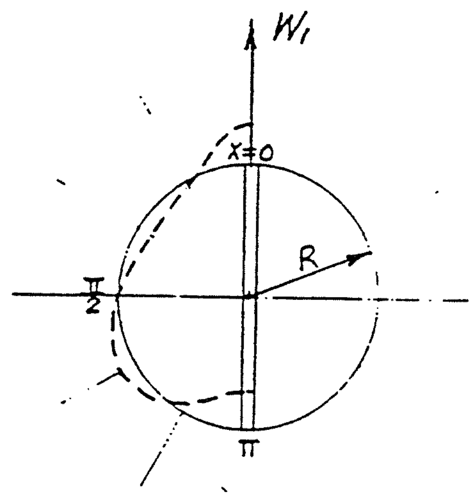
Moment due to inward load is:

$$M_2 = -W_5 R[.2387 \cos(\pi - X) - .5 \sin(\pi - X) + .1592\{(\pi - X)\sin(\pi - X) + 1 - \cos(\pi - X)\}]$$

$$= -W_5 R[-.2387 \cos X - .5 \sin X + .1592\{(\pi - X)\sin X + 1 + \cos X\}]$$

X	M_1	M_2	No Strut $M = 2M_1$	Strut $M = M_1 + M_2$
0	$+.1194\, W_1R$	$-.0398\, W_1R$	$+.2388\, W_1R$	$+.0796\, W_1R$
$\pi/6$	$+.0099\, W_1R$	$-.0243\, W_1R$	$+.0198\, W_1R$	$-.0144\, W_1R$
$\pi/3$	$-.0449\, W_1R$	$+.0124\, W_1R$	$-.0898\, W_1R$	$-.0325\, W_1R$
$\pi/2$	$-.0454\, W_1R$	$+.0454\, W_1R$	$-.0908\, W_1R$	0
$2\pi/3$	$-.0124\, W_1R$	$+.0449\, W_1R$	$-.0248\, W_1R$	$+.0325\, W_1R$
$5\pi/6$	$+.0243\, W_1R$	$-.0099\, W_1R$	$+.0486\, W_1R$	$-.0144\, W_1R$
π	$+.0398\, W_1R$	$-.1194\, W_1R$	$+.0796\, W_1R$	$-.0796\, W_1R$





Sample Problem

Given: Column 6'-0" I.D. × 100'-0" T/T plus 11 ft. skirt

Wt. with trays, insulation and platforms = 135 kips

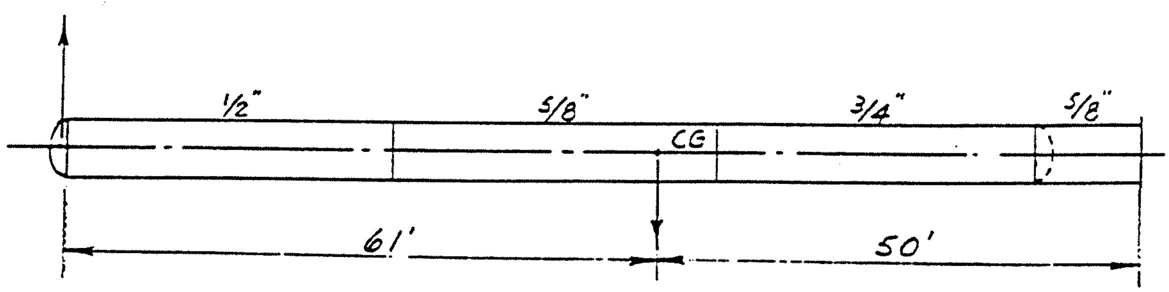
Center of gravity: 50 ft. from base.

Vessel thickness: ½" top third, 5/8" middle third, and ¾" bottom third. Skirt 5/8".

Lift point: (A) near the top head.

(B) 55'–60' from base.

Required: Lifting lug design.



(A) Ear Type Lifting Lug

Max. lift load (vertical, tailing load = 0)

$F = \frac{1}{2} \times 135 \times 1.5 = 101$ Kips/lug (46 tons/lug)

Horiz. lift load $= F = \frac{1}{2} \times 135 \times 1.5 \times \frac{50}{112} = 45$ Kips/lug

From Table 1, a size 3" anchor shackle is required:

$d = 3\frac{1}{4}$" (pin)

G (grip) = 5", ∴ Use lug thick. $T \cong 4\frac{7}{8}$"

From Eq. (3), lug plate thickness

$t = \frac{F}{3d\sigma_t} = \frac{101}{3 \times 3.25 \times 20} = 0.52"$. Use $1\frac{1}{4}$" plate

Collar plate thick. $t_1 = \frac{1}{2}(4\frac{7}{8} - 1\frac{1}{4}) = 1.81''$. Use $1\frac{3}{4}''$

Assume dimension $L = 12''$.

Then from Eq.(5),

Width of lug plate $= B \geq \sqrt{\frac{6FL}{t\sigma_b}} = 10.9''$ Use $12''$

Since σ_b is taken to be 22 ksi, and $\sigma_t = 20$ ksi

$\lambda = \sigma_b/\sigma_t = 1.1$

$\therefore$ Max. combined stress in the lug plate occurs at

$\alpha_m = \tan^{-1}\frac{\lambda B}{6L} = \tan^{-1}\frac{13.2}{72} = 10.4°$

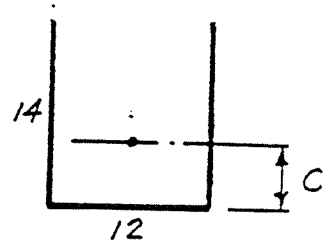
$\therefore$ Combined stress $= \frac{45 \sin 10.4°}{1.25 \times 12} + \frac{6}{1.1}\frac{45 \times 12 \cos 10.4°}{1.25 \times 12^2}$

$= 0.54 + 16.09 = 16.6$ ksi < 20 ksi $\therefore$ O.K.

Assume a weld length $L_3 = 14''$ and a weld size $t_w = \frac{1}{2}''$

From Eq.(9), for vessel in the horiz. position, shear stress in lug attachment weld

$= 45/[0.7(12 + 2 \times 14) \times 0.5] = 3.2$ ksi



Centroidal distance $C = \frac{14^2}{28 + 12} = 4.9''$

Moment about weld centroid

$= 45(12 + 14 - 4.9) = 950$ in.k.

Polar mom. of inertia of weld

$= 2\left[\frac{0.5 \times 14^3}{12} + 0.5 \times 14(7 - 4.9)^2\right] + 0.5 \times 12 \times 4.9^2$

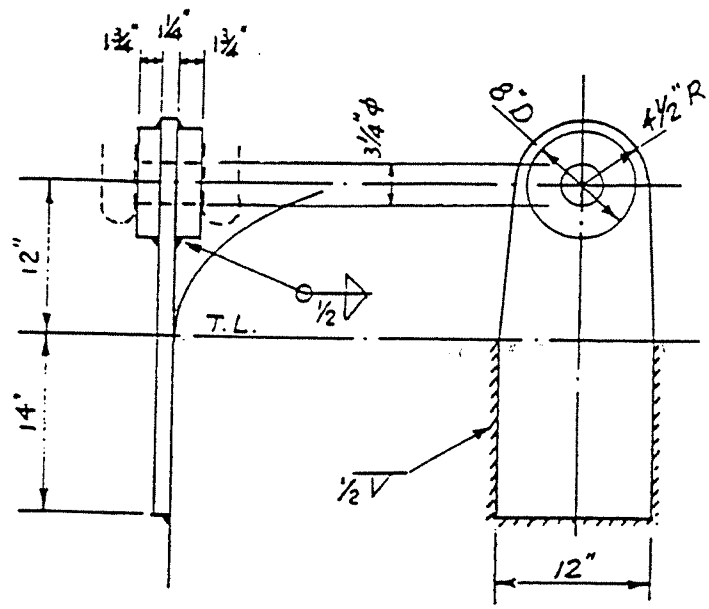
$+ 2 \times 0.5 \times 14 \times 6^2 + \frac{0.5 \times 12^3}{12} = 1010$ in.^{4}

Polar section modulus $= \frac{1010}{\sqrt{6^2 + 9.1^2}} = 92.7$ in.^{3}

$\therefore$ Torsional stress due to moment $= 950/92.7 = 10.2$ ksi

Combined shear stress for vessel in horiz. position
$= 3.2 + 10.2 = 13.4$ ksi O.K.

Shear stress in weld for vessel in vert. position
$= 101/[0.7(12+28)\times 0.5] = 7.2$ ksi O.K.



(B) Trunnion Type Lug (assume fixed pipe) (55' from base)

Max. lift load (vertical) = 101 k.

Max. lift load (horiz.) $= 0.5\times 1.5\times 135\times \frac{50}{55} = 92$ k.

As in example (A), 3" shackle is required:

$d = 3\frac{1}{4}"$, $t = 1\frac{1}{4}"$, $t_1 = 1\frac{3}{4}"$

Assume dimensions E and L to meet clearance requirements: $E = 8"$, $L = 15"$

From Eq. (16), Z of trunnion pipe $= 92\times 8/22 = 33.5$ in.^{3}

or Eq. (17), Z of trunnion pipe $= 92\times 15/(12\times 13.5) = 51.1$ in^3

From Eq. (18), X-sectional area of pipe $= 101/13.5 = 7.5$ in.^{2}

Try 12"XS pipe: $A \cong 19.2$ in.^{2}, $Z = 58.9$ in.^{3}

Combined shear & torsional stress $= \frac{92 \times 15}{2 \times 58.9} + \frac{92}{19.2} = 16.5\ ksi$

This stress exceeds $\sigma_s = 13.5\ ksi$

$\therefore$ Use 14" XS pipe : $Z = 71.5\ in.^3$, $A = 21.2\ in.^2$, $\sigma_{comb} = 14\ ksi$

Check vessel stress using Fig. 7.

$\gamma = \frac{36}{0.625} = 57.6$, $\frac{Trun.\ Diam}{Vess.\ Diam} = \frac{14}{72} = 0.19$

$\therefore f_r = 0.053$

Vessel Stress $= \frac{0.053 \times 92 \times 8}{0.625^3} = 160\ ksi$

By using a 3/4" thick pad , $t_v = 0.75 + 0.625 = 1.375"$

Stress around pipe $= (0.053 \times 92 \times 8)/1.375^3 = 15\ ksi$

Stress around the pad (assume reasonable size) will still be excessive.

$\therefore$ a turning axle trunnion is recommended.

Bearing stress at shell opening $= \frac{101}{14 \times 0.625} = 11.5\ ksi$ O.K.

If size of weld between lug plate and pipe end is 1/2", then weld stress $= \frac{101}{\pi \times 14 \times 0.5 \times 0.7} = 6.6\ ksi$ OK