

10

Transportation and Erection of Pressure Vessels

Procedure 10-1: Transportation of Pressure Vessels	632	Procedure 10-7: Design of Flange Lugs	695
Procedure 10-2: Erection of Pressure Vessels.....	660	Procedure 10-8: Design of Trunnions	706
Procedure 10-3: Lifting Attachments and Terminology	666	Procedure 10-9: Local Loads in Shell Due to Erection Forces.....	710
Procedure 10-4: Lifting Loads and Forces.....	675	Procedure 10-10: Miscellaneous	713
Procedure 10-5: Design of Tail Beams, Lugs, and Base Ring Details	681		
Procedure 10-6: Design of Top Head and Cone Lifting Lugs.....	691		

Procedure 10-1: Transportation of Pressure Vessels

The transportation of a pressure vessel by ship, barge, road, or rail will subject the vessel to one-time-only stresses that can bend or permanently deform the vessel if it is not adequately supported or tied down in the right locations. The shipping forces must be accounted for to ensure that the vessel arrives at its destination without damage.

It is very frustrating for all the parties involved to have a load damaged in transit and to have to return it to the factory for repairs. The cost and schedule impacts can be devastating if a vessel is damaged in transit. Certain minimal precautions can avoid the costly mistakes that often lead to problems. Even when all precautions are made, however, there is still the potential for damage due to unforeseen circumstances involved in the shipping and handling process.

Care should be taken to ensure that the size and location of the shipping saddles, tie-downs, or lashing are adequate to hold the vessel but not deform the vessel. Long, thin-walled vessels, such as trayed columns, are especially vulnerable to these shipping forces. The important thing to remember is that someone must take the responsibility. The barge and rail people have their own concerns with regard to loading and lashing. These may or may not coincide with the concerns of the vessel designer.

The shipping forces for ships, barges, trucks, and rail are contained in this procedure. Each method of transportation has its own unique load schemes and resulting forces. Barge shipping forces will differ from rail due to the rocking motion of the seas. Rail shipments, however, go around corners at high speed. In addition, rail forces must allow for the "humping" of rail cars when they are joined with the rest of the train. Ocean shipments have to resist storms and waves without breaking free of their lashings.

Whereas horizontal vessels on saddles are designed for some degree of loading in that position, vertical vessels are not. The forces and moments that are used for the design of a vertical vessel assume the vessel is in its operating position. Vertical vessels should generally be designed to be put on two saddles, in a horizontal position, and transported by various means. That is the purpose of this procedure. Too often the details of transportation and erection are left in the hands of people who, though well versed in their particular field, are not pressure vessel specialists.

Often vessels are transported by multiple means. Thus there will be handling operations between each successive mode of transportation. Often a vessel must be moved by road to the harbor and then transferred to a barge or ship. Once it reaches its destination, it must be reloaded onto road or rail transport to the job site. There it will be off-loaded and either stored or immediately erected. A final transport may be necessary to move the vessel to the location where it will be finally erected. At each handling and transport phase there are different sets of forces exerted on the vessel that must be accounted for.

Shipping Saddles

The primary concern of the vessel designer is the location and construction of the shipping saddles to take these forces without overstressing or damaging the vessel. If saddles are to be relocated by the transporter, it is important that the new locations be reviewed. Generally only two shipping saddles should be used. However, this may not always be possible. Remember that the reason for using two saddles is that more than two saddles creates a statically indeterminate structure. You are never assured that any given saddle is going to take more than its apportioned load.

Here are some circumstances that would allow for more than two saddles to be used or for a special location of two saddles:

- Transporter objects due to load on tires.
- Transporter objects due to load on barge or ship.
- Very thin, long vessel.
- Heavy-walled vessels for spreading load on ship or transporters.

Shipping saddles can be constructed of wood or steel or combinations. The saddles should be attached to the vessel with straps or bolts so that the vessel can be moved without having to reattach the saddle. Horizontal vessels may be moved on their permanent saddles but should be checked for the loadings due to shipping forces and clearances for boots and nozzles. Shipping saddles should have a minimum contact angle of 120° , just like permanent saddles. Provisions for jacking can be incorporated into the design of the saddles to allow loading and handling operations without a crane(s).

Shipping saddles should be designed with the vessel and not left up to the transport company. In general, transportation and erection contractors do not have the capability to design shipping saddles or to check the corresponding vessel stresses for the various load cases.

Whenever possible, shipping saddles should be located adjacent to some major stiffening element. Some common stiffening elements include stiffening rings, heads (both internal and external), or cones. If necessary, temporary internal spiders can be used and removed after shipment is complete.

Key factors for shipping saddles to consider:

- Included angle.
- Saddle width.
- Type of construction.
- Lashing lugs.
- Jacking pockets.
- Method of attachment to the vessel.
- Overall shipping height allowable—check with shipper.

Recommended contact angle and saddle width:

Vessel Diameter	Contact Angle	Minimum Saddle Width
D < 13 ft-0 in.	120°	11 in.
13 ft-0 in. < D < 24 ft-0 in.	140°	17 in.
D > 24 ft-0 in.	160°	23 in.

Vessel Stresses

The stresses in the vessel shell should be determined by standard Zick’s analysis. The location of shipping saddles should be determined such that the bending at the midspan and saddles is not excessive. Also, the stresses due to bending at the horn of the saddle is critical. If this stress is exceeded, the saddle angle and width of saddle should be increased. Also, move the saddle closer to the head or a major stiffening element.

Lashing

Vessels are lashed to the deck of ships and barges. In like manner they must be temporarily fixed to rail-cars, trailers, and transporters. Lashing should be restricted to the area of the saddle locations. Vessels

are held in place with longitudinal and transverse lashings. Lashings should never be attached to small nozzles or ladder or platform clips. In some cases, lashing may be attached to lifting lugs and base rings. Lashings should not exceed 45° from the horizontal plane.

Other Key Factors to Consider

- Shipping clearances.
- Shipping orientation—pay close attention to lift lugs and nozzles.
- Shipping route.
- Lifting orientation.
- Type of transport.
- Watertight shipment for all water transportation.
- Escorts and permits.
- Abnormal loads—size and weight restrictions.
- Vessels shipped with a nitrogen purge.
- Shipping/handling plan.

Organizations That Have a Part in the Transportation and Handling of Pressure Vessels

- Vessel fabricator.
- Transport company.
- Engineering contractor.
- Railway authorities.
- Port authorities.
- Erection/construction company.
- Trailer/transporter manufacturer.
- Ship or barge captain.
- Crane company/operator.

Special Considerations for Rail Shipments

1. Any shipment may be subject to advance railroad approval.
2. Any shipment over 10 ft-6 in. wide must have railroad approval.
3. A shipping arrangement drawing is required for the following:
 - a. All multiple carloads (pivot bolster required).
 - b. All single carloads over 10 ft-6 in. wide.
 - c. All single carloads over 15 ft-0 in. ATR (above top of rail).
 - d. All single carloads that overhang the end(s) of the car and are over 8 ft-0 in. ATR.

4. Clearances must be checked for the following:
 - a. Vessels greater than 9 feet in width.
 - b. Vessels greater than 40 feet overall length.
 - c. Vessels greater than 50 tons.
5. The railroad will need the following specific data as a minimum:
 - a. Weight.
 - b. Overall length.
 - c. Method of loading.
 - d. Loadpoint locations.
 - e. Overhang lengths.
 - f. Width.
 - g. Height.
 - h. Routing/route surveys.
 - i. Center of gravity.
6. A swivel (pivot) bolster is required whenever the following conditions exist:
 - a. Two or more cars are required.
 - b. The capacity for a single car is exceeded.
 - c. The overhang of a single car exceeds 15 feet.
7. Rated capacities of railcars are based on a uniformly distributed load over the entire length of the car. The capacity of a car for a concentrated load will only be a percentage of the rated capacity.
8. Rules for loads, loading, and capacities vary by carrier. Other variables include the types of cars the carrier runs, the availability, and the ultimate destination. Verify all information with the specific carrier before proceeding with the design of shipping saddles or locations.
9. For vessels that require pivot bolsters, the shipping saddles shall be adequately braced by diagonal tension/compression rods between the vessel and the saddle. The rods and clips attached to the vessel shell should be designed by the vessel fabricator to suit the specific requirements of the carrier.
10. If requested, rail bolsters can be returned to the manufacturer.
11. Loading arrangement and tie-downs will have to pass inspection by a representative of the railways and sometimes by an insurance underwriter prior to shipment.

12. Accelerometers can be installed on the vessel to monitor shipping forces during transit.
13. A rail expeditor who accompanies the load should be considered for critical shipments.
14. The railroad will allow a fixed time for the cars to be offloaded, cleaned, and returned. Demurrage charges for late return can be substantial.

Outline of Methods of Vessel Shipping and Transportation

1. Road.
 - a. Truck/tractor and trailer.
 - b. Transporters—single or multiple, self-propelled or towed.
 - c. Special—bulldozer.
 - d. Frame adapters.
 - e. Beams to span trailers or transporters.
 - f. Rollers.
 - g. Special.
2. Rail.
 - a. Single car.
 - b. Multiple cars.
 - c. Special cars.
 - d. Types of cars.
 - Flatcar.
 - Fishbelly flatcar.
 - Well car.
 - Heavy-duty car.
 - Gondola car.
3. Barge.
 - a. River barge.
 - b. Ocean-going barge.
 - c. Lakes and canals.
4. Ships.
 - a. Roll-on, roll-off type.
 - b. Loading and off-loading capabilities.
 - c. In-hull or on-deck.
 - d. Floating cranes.
5. Other.
 - a. Plane.
 - b. Helicopter.
 - c. Bulldozer.

Table 10-1
Overland shipping limits in the US

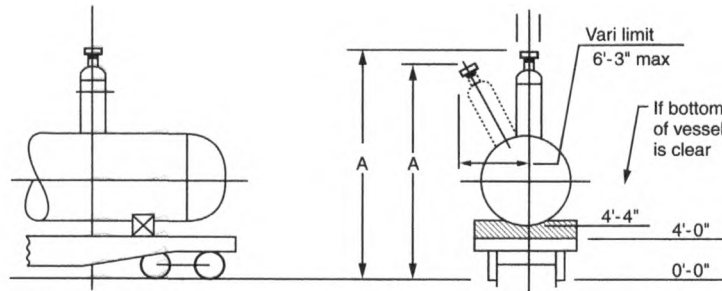
State	Length, ft	Width, ft	Height, ft	Gross Weight, lb
Alabama	150	16	16	180,000
Arizona	120	14	16	250,000
Arkansas	100	14	14	120,000
California	120	14	16	220,000
Colorado	130	17	16	228,000
Connecticut	120	15	16	250,000
Delaware	120	15	15	250,000
Florida	150	16	16	250,000
Georgia	100	14	14	120,000
Idaho	110	16	15'6"	200,000
Illinois	145	14'6"	15	250,000
Indiana	110	16	15	250,000
Iowa	120	18	16	250,000
Kansas	126	16'6"	16	250,000
Kentucky	110	16	15	250,000
Louisiana	125	18	16	250,000
Maine	125	16	16	250,000
Maryland	100	15'11	15'11	220,000
Massachusetts	115	14	14	240,000
Michigan	150	16	15	230,000
Minnesota	95	14'6"	14	250,000
Mississippi	100	14	14	120,000
Missouri	100	14	14	120,000
Montana	110	18	17	240,000
Nebraska	120	14	15'6"	212,000
Nevada	105	17	16	240,000
New Hampshire	120	15	16	250,000
New Jersey	120	18	16	220,000
New Mexico	120	14	16	250,000
New York	120	14	14	160,000
North Carolina	100	14	14	120,000
North Dakota	120	14'6"	15'6"	150,000
Ohio	100	14	14'10"	120,000
Oklahoma	100	16	16	212,000
Oregon	105	14	16	220,000
Pennsylvania	120	16	15'6"	201,000
Rhode Island	90	14	13'6"	120,000
South Carolina	125	16	14	250,000
South Dakota	120	14'6"	15'6"	150,000
Tennessee	120	16	15	250,000
Texas	125	20	18'11"	252,000
Utah	125	15	16'6"	250,000
Vermont	100	15	14	150,000
Virginia	150	14	15	150,000
Washington	150	14	16	200,000
West Virginia	150	16	15	212,000
Wisconsin	150	16	16	250,000
Wyoming	110	18	17	252,000

Note: This information in this Table is for general, reference information only and should not be relied upon for any given application. These values change regularly and local, state and national regulations should be checked for any given haul application.

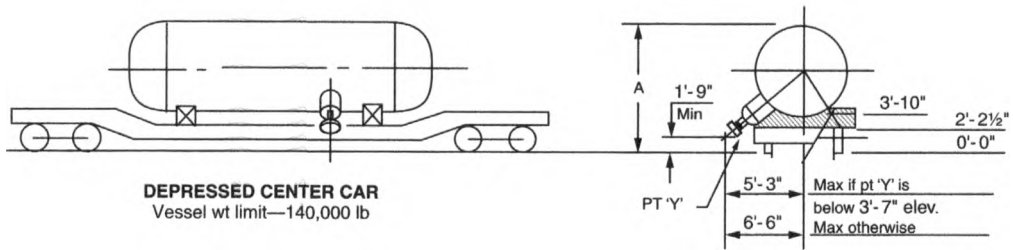
Rail—Types of Cars

Notes:

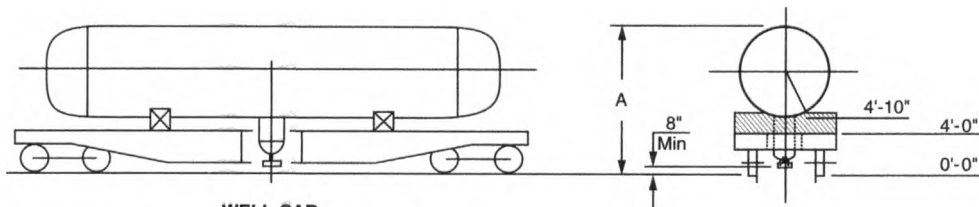
1. Allowable vessel weight ranges and limits are subject to reductions under certain conditions and as noted herein.
2. Dimension A = ATR, above top of rail.



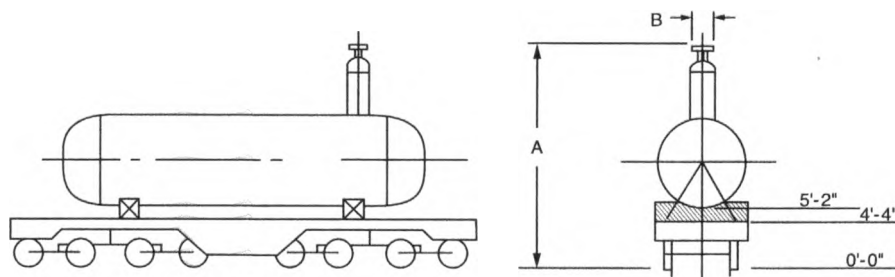
FISHBELLY FLATCAR
Vessel wt limit—140,000 lb



DEPRESSED CENTER CAR
Vessel wt limit—140,000 lb

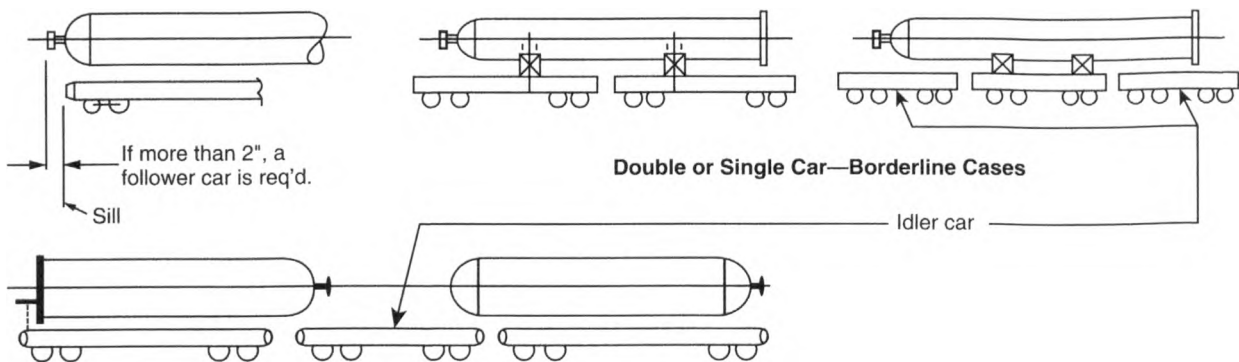


WELL-CAR
Vessel wt range (140,000 – 250,000 lb)



HEAVY-DUTY CAR
Vessel wt range (140,000 – 400,000 lb)

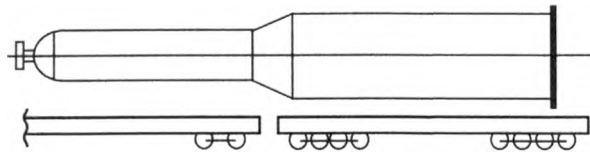
Rail—Multiple Car Loading Details



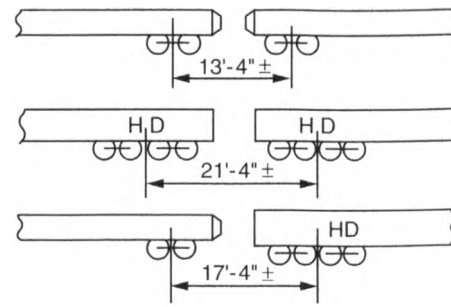
Double or Single Car—Borderline Cases

Idler car

Bolster load with overhang over fourth car is acceptable. A brakewheel must be used in at least 1 of every 3 cars.

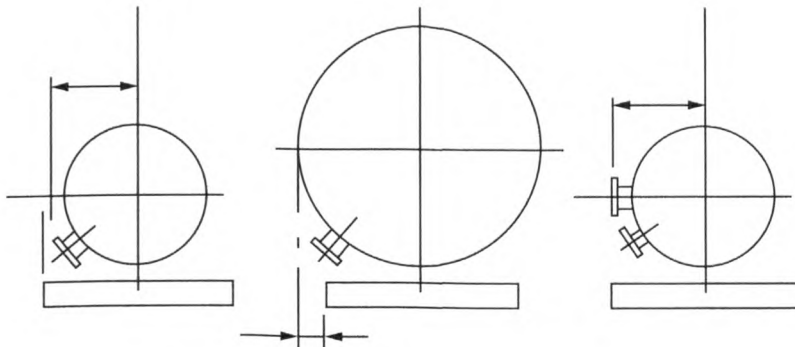
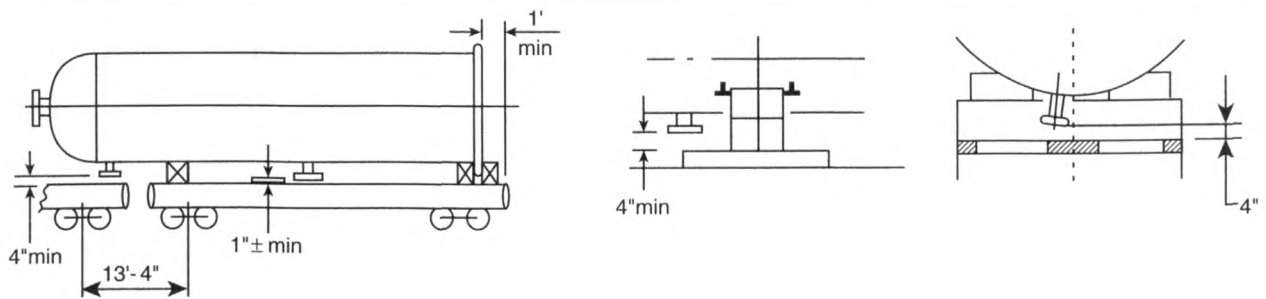


Avoid combination of dissimilar cars.



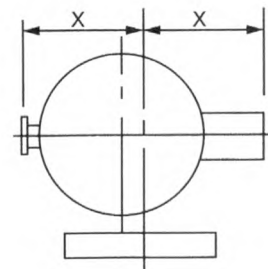
Truck Centers for Bolster Loads

Rail—Clearances



Clearance of Projections

Note: Minimum clearance to any moving part. This includes nozzles, shipping covers, or clips.



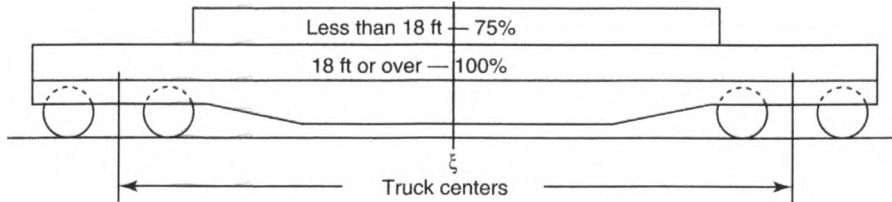
Offset Loads

Ballast may be required to offset heavy loads. "Depressed center cars" are favored for these applications.

Rail—Capacity Ratios for Concentrated Loads

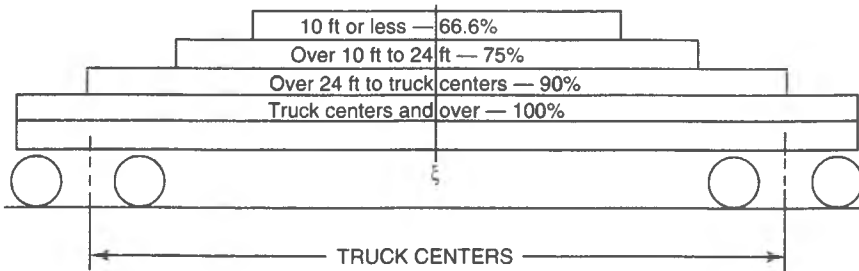
1. Flatcars with both fish belly center and fishbelly side sills and all flatcars built after January 1, 1965.

Less than 18 ft	75%
18 ft or over	100%



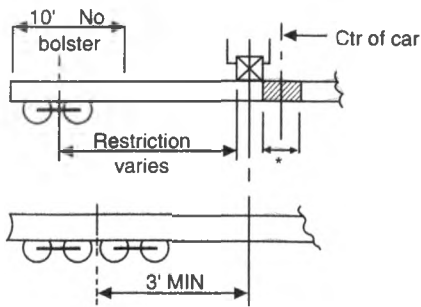
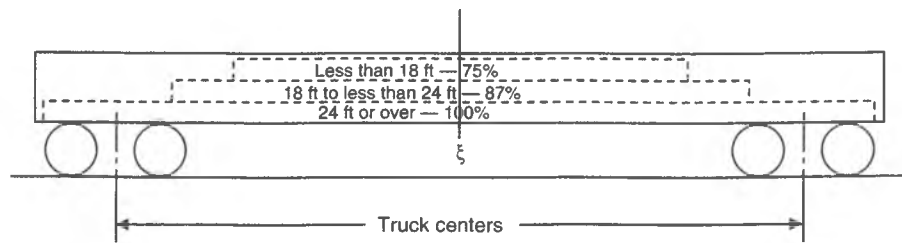
2. Flatcars not equipped with both fishbelly center and fishbelly side sills built prior to January 1, 1965.

10 ft or less	66.6%
Over 10 ft to 21 ft	75%
Over 21 ft to truck centers	90%
Truck centers and over	100%



3. Gondola cars.

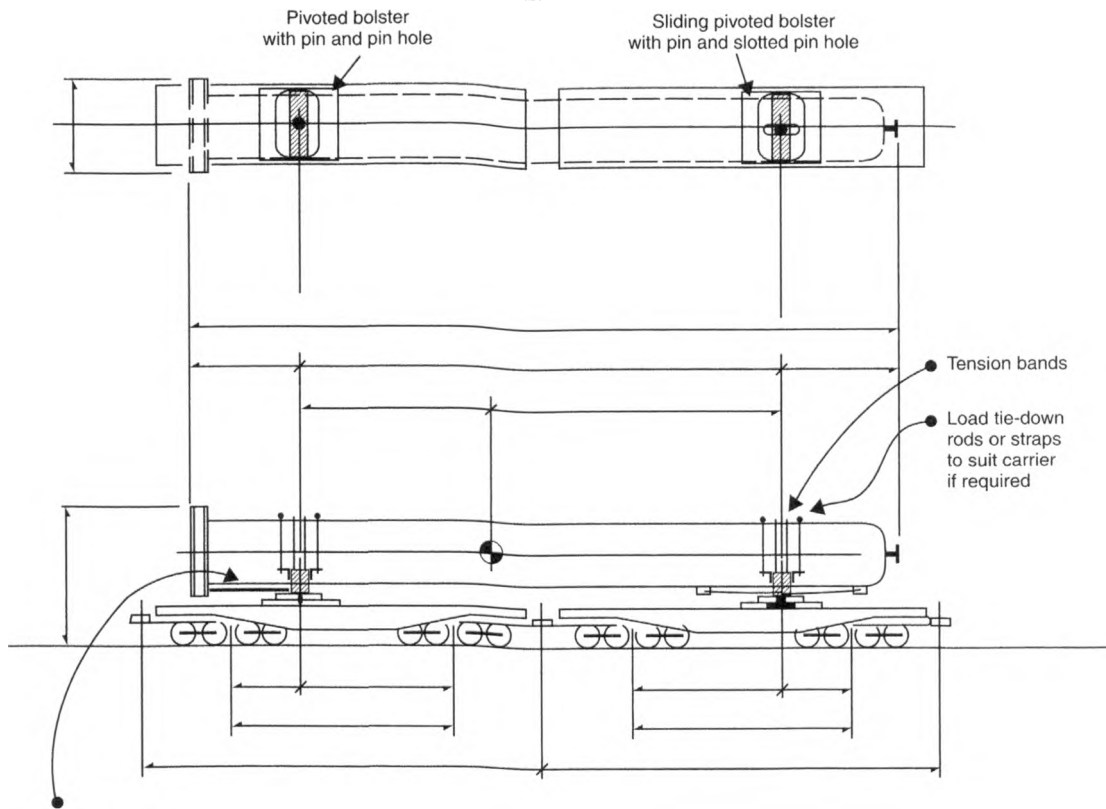
Less than 18 ft	75%
18 ft or less than 24 ft	87%
24 ft or over	100%



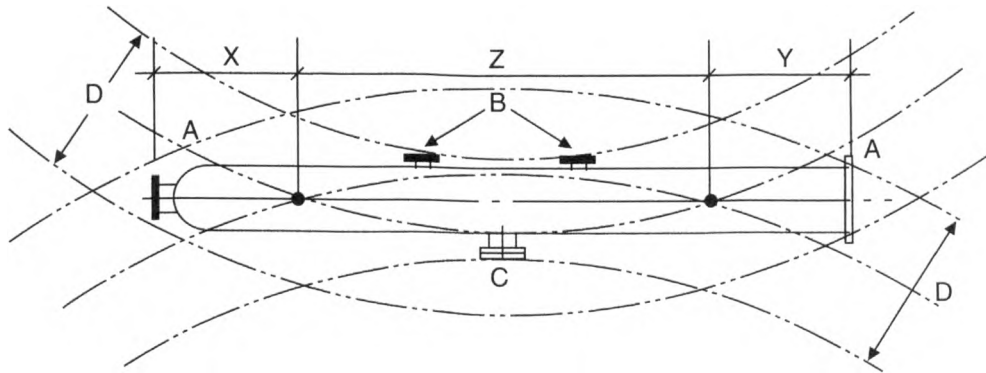
*3' restricted @ center of car to 50,000 lb except for heavy-duty cars

Bolster Locations

Rail—Details of Pivoted Bolster Loads



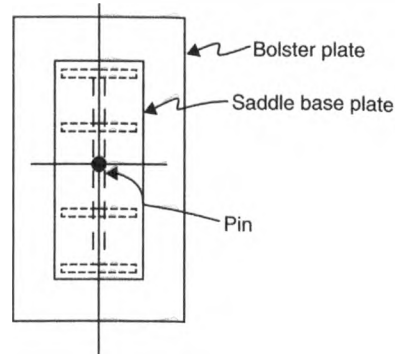
Longitudinal tie-downs are required at each saddle to suit the individual carrier. Tie-downs may consist of two brace rods, steel cables, and turnbuckles or a brace frame against the vessel base plate to take the longitudinal loads. The vessel fabricator should provide adequate clips or like attachment to the vessel for securing this bracing to the vessel shell. It is imperative that any welding to the vessel be done in the shop!



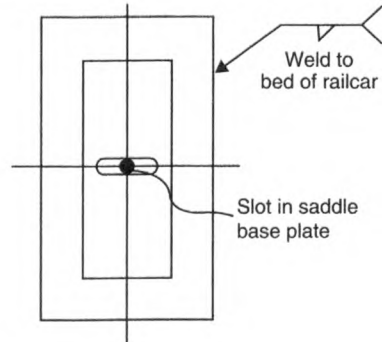
BOLSTER SETTING & CLEARANCES

1. Set X, Y, and Z so that clearance at points A, B, and C are adequate.
2. Watch relationship between bolsters and car trucks and car ends.
3. Add a minimum of 1 in. to all lateral dimensions to allow for shipping covers and small projections.
4. Dimension "D" shall be a maximum of 15 ft-5 in. of occupied space based on a 10° curve.

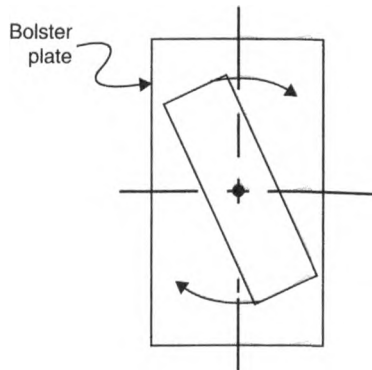
Details of Bolsters



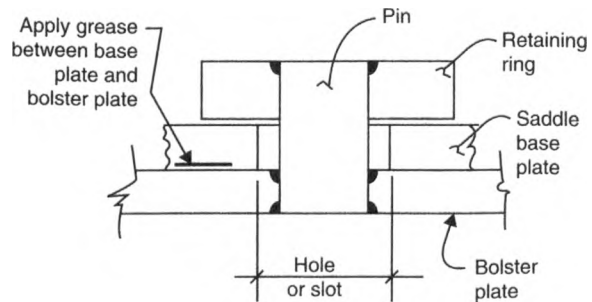
PIVOT BOLSTER
Pivots only



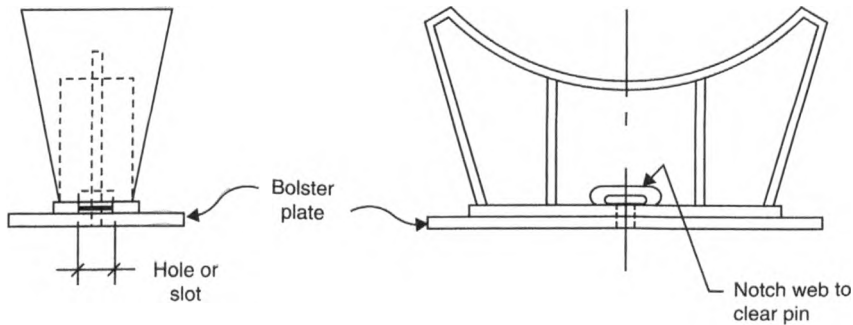
SLIDING PIVOTED BOLSTER
Allows for longitudinal and angular movement



Saddle plate rotates during turns



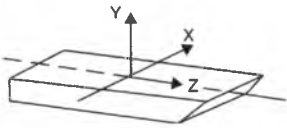
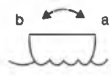

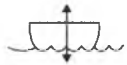
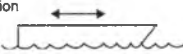
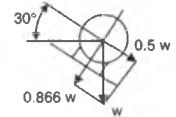
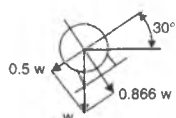
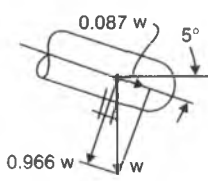
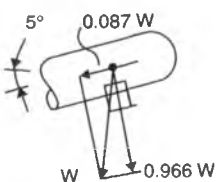
Design pin for shear based on load F_z



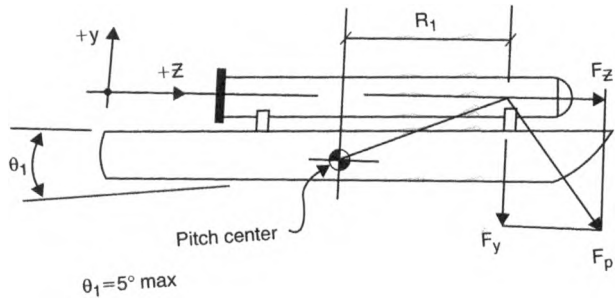
Notes:

1. Pivoting bolsters must be used for all rail shipments.
2. Pivoting bolsters must be utilized for all vessels spanning two or more railcars.
3. Design pin for shear based on full load of F_z .
4. Do not anchor the saddle plate to the bolster plate or the railway bed. The saddle plate must be free to rotate on the bolster plate. Only the bolster plate is anchored to the railway bed. The most common means of anchoring the bolster plate to the railway bed is welding. Design anchorage for a load of $\frac{1}{2}F_z$.
5. Apply grease generously between saddle base plate and bolster plate.
6. In general all clips or welds on the railcar will have to be removed, ground, and cleaned to the satisfaction of the railways prior to return.

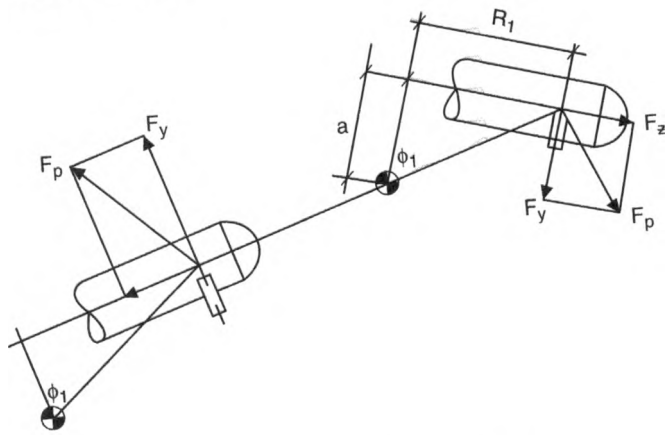
Table 10-2
Barge shipping forces

Case			F = force due to barge motion, lb W = shipping weight, lb T = period of vibration of barge, secs		
	Condition	F_x	F_y	F_z	Diagram
1	Gravity	—	-1.0w	—	—
2a	Roll 	$\pm 0.45w$	-0.4w	—	—
2b		$\pm 0.45w$	+0.4w	—	—
3a	Pitch 	—	-1.5w	1.0w	—
3b		—	+1.5w	1.0w	—
4	Heave 	—	1.2w	—	—
5	Collision 	—	—	$\pm 1.5w$	—
6a	Roll + Gravity	+0.95w +0.05w	-1.266w	—	
6b		-0.95w -0.05w	-0.466w	—	
7a	Pitch + Gravity	—	-2.5w	$\pm 0.5w$	
7b		—	+0.5w	$\pm 0.5w$	

Pitch



Cases 3a and 3b



Forces in Vessel Due to Pitch

General:

$$F = ma = \left(\frac{W}{g}\right) \left(\frac{2\pi}{T}\right)^2 \left(\frac{R\theta\pi}{180}\right)$$

$$F = 0.0214 \frac{WR\theta}{T^2}$$

$$\phi_1 = \tan^{-1}\left(\frac{a}{R_1}\right)$$

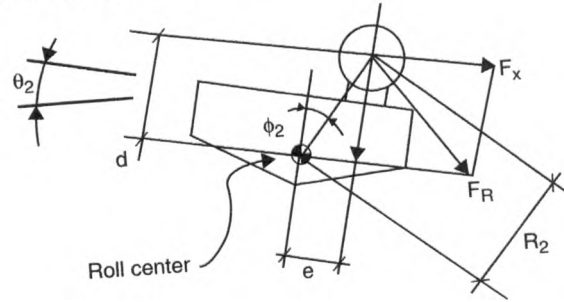
$$F_p = \frac{0.0214WR_1\theta_1}{T_1^2}$$

Case 3a: $F_y = -F_p \sin \phi_1$
 $F_z = F_p \cos \phi_1$

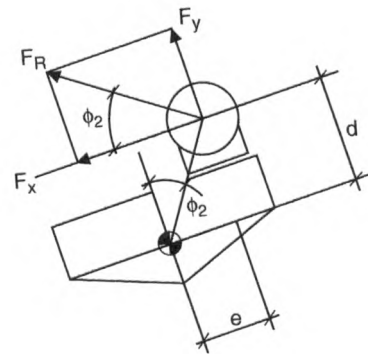
Case 3b: $F_y = F_p \sin \phi_1$
 $F_z = -F_p \cos \phi_1$

Roll

Case 2a: $\theta_2 = 30^\circ$ max



Case 2b



Forces in Vessel Due to Roll

$$\phi_2 = \tan^{-1}\left(\frac{e}{d}\right)$$

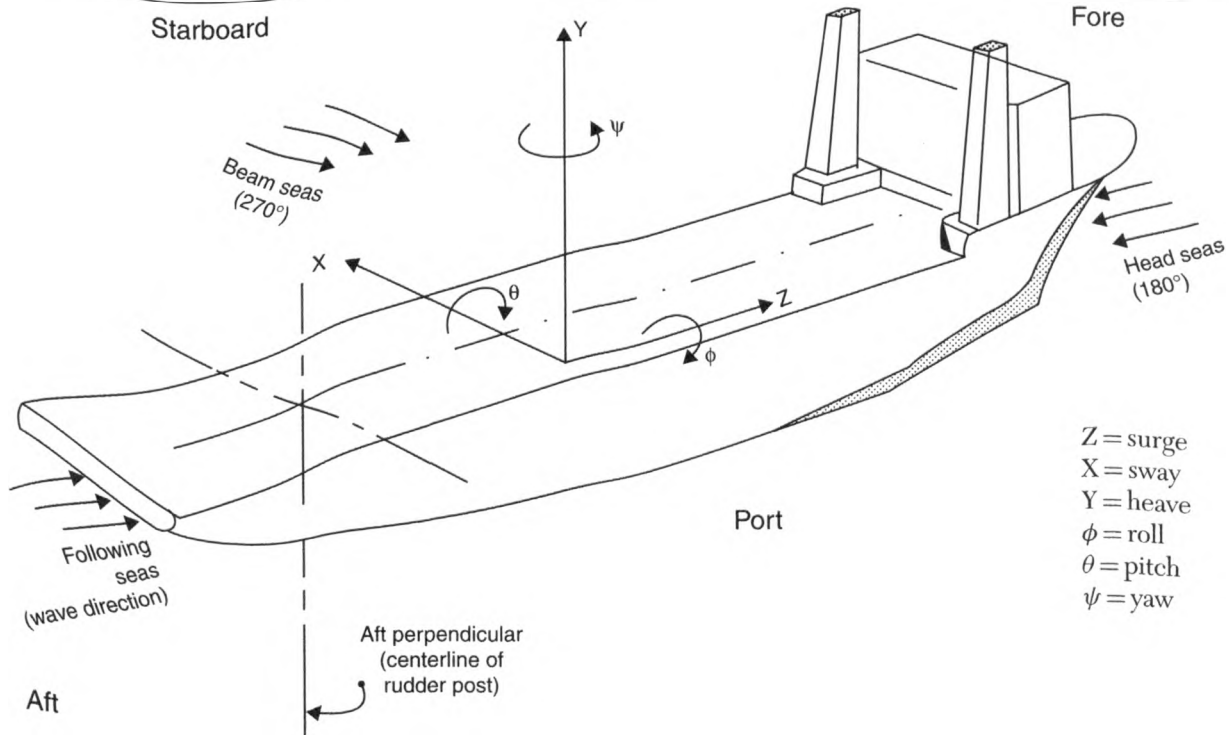
$$R_2 = \frac{e}{\sin \phi_2}$$

$$F_R = \frac{0.0214WR_2\theta_2}{T_2^2}$$

Case 2a: $F_y = -F_R \sin \phi_2$
 $F_x = F_R \cos \phi_2$

Case 2b: $F_y = F_R \sin \phi_2$
 $F_x = -F_R \cos \phi_2$

Directions of Ship Motions



The job of the designer is to translate the loads resulting from the movement of the ship into loads applied to the pressure vessel that is stored either at or below decks. The ship itself will rotate about its own center of buoyancy (CB) depending on the direction of the sea and the ship's orientation to that direction of sea. The vessel strapped to its deck is in turn affected by its location in relation to the CB of the ship. For example, if the CG of the vessel is located near the CB of the ship, the forces are minimized. The farther apart the two are in relation to each other, the more pronounced the effect on the vessel.

The ship's movement translates into loads on the three principal axes of the vessel. Saddles and lashings must be strong enough to resist these external forces without exceeding some allowable stress point in the vessel. The point of application of the load is at the CG of the vessel. These loads affect the vessel in the same manner as

seismic forces do. In fact, the best way to think of these loads is as vertical and horizontal seismic forces. Vertical seismic forces either add or subtract to the weight of the vessel. Horizontal seismic forces are either transverse or longitudinal.

The X, Y, and Z axes translate into and are equivalent to the following loadings in the vessel:

X axis: horizontal transverse.

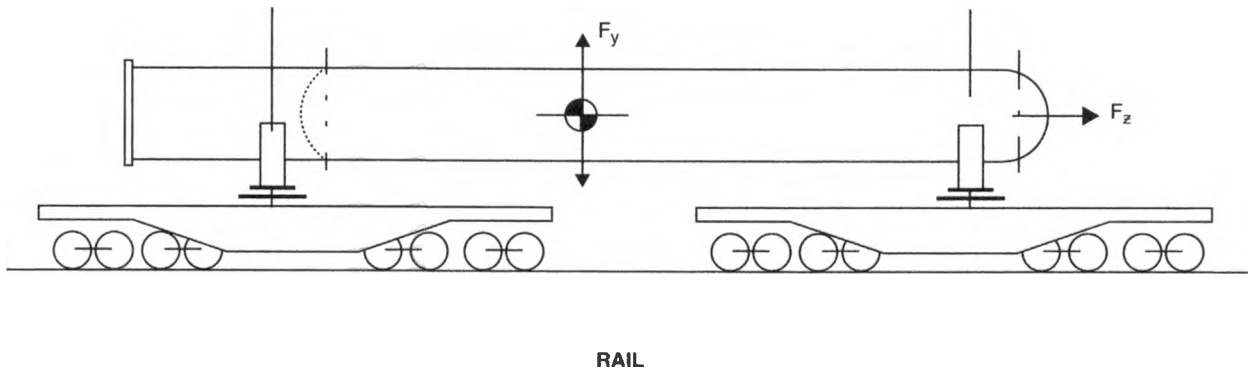
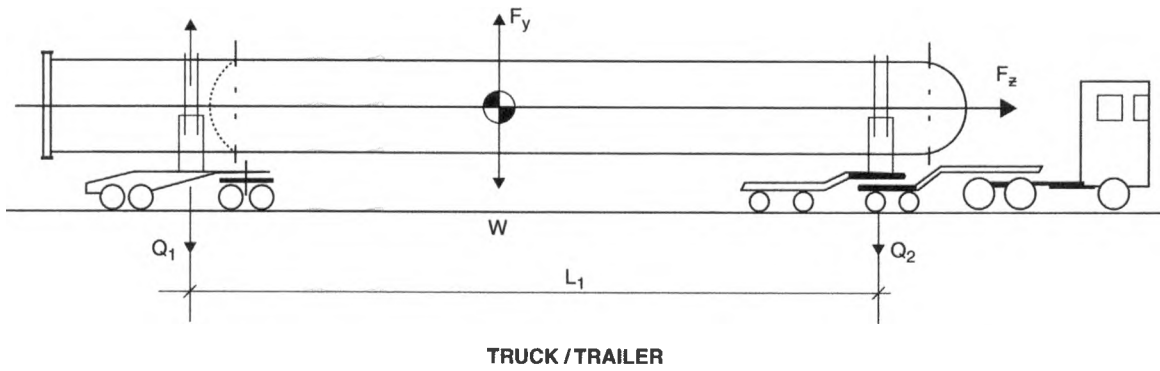
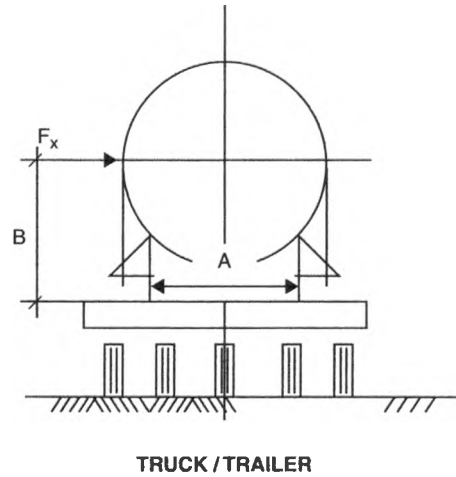
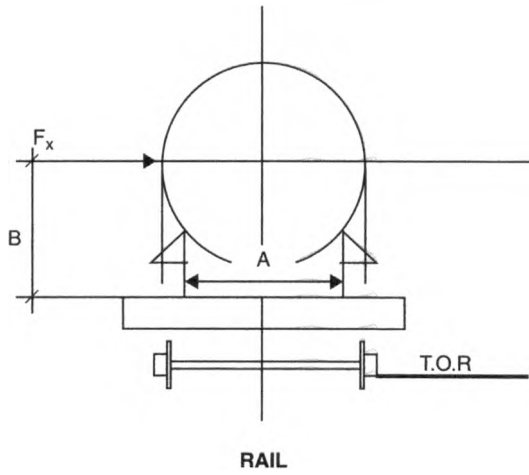
Y axis: corresponds to vertical loads by either adding or subtracting from the weight of the vessel.

Z axis: longitudinal axis of the vessel. All Z axis loads are longitudinal loadings.

Load Combinations for Sea Forces

1. dead load + sway + heave + wind
2. dead load + surge + heave + wind

Forces on Truck and Rail Shipments



Axle Loads

The number of axles that must be placed under a load is determined by analyzing the weight restrictions and allowable bearing load from local, state or national regulations. The transportation contractor is responsible for determining the axle loads based on the equipment used and the weight and distribution of the loads. The authorities that permit the load will require an analysis of the axle loads to ensure that the roadbed is not overloaded. Axle loads include the weight of the vessel, transport saddles, beams, hauler (tractor), dollies, etc.

There are three different methods used to distribute the loads to the axles;

1. Flatbed: This method uses conventional tractor-trailer assembly with various numbers of axles and wheels under the trailer bed to distribute the weight to the road surface.
2. Bolstered: This method is used for abnormally long loads in which the sets of axles are attached directly to the load via the transport saddle. Both sets of axles will have steering capabilities.
3. Bolstered loads using equalizing transporting beams: This method is much the same as the bolstered, long vessel load. In this case the load is too heavy for a flatbed, yet too short for bolstered axles. The solution is to utilize beams between bolsters to suspend the load.

Transporter Stability

There are two types of stability checks that should be performed on each load. The first has to do with the tipping point of the load relative to the roadway as the load shifts due to the camber of the road. The second has to do with the turning radius of bolstered loads. As the load goes around curves, the C.G. shifts from being in line with the dollies, to an eccentric condition. In tight curves the

eccentricity of the load can overload the outer set of wheels to the point of rollover. The two cases are;

1. Rollover stability due to road camber
2. Turning stability due to turning radius

Case 1:

Due to the camber in roads, the load will be subject to various angles, θ , that will change the location of the center of gravity of the load. On a flat surface, the center of gravity is in line with the centerline of the trailer. As the road camber increases, the C.G. is steadily moved toward the outer set of wheels. At some point the wheels are overloaded on one side and the entire assembly reaches a tipping point. Beyond this, the trailer turns over and the load is lost. This condition has resulted in numerous rollovers.

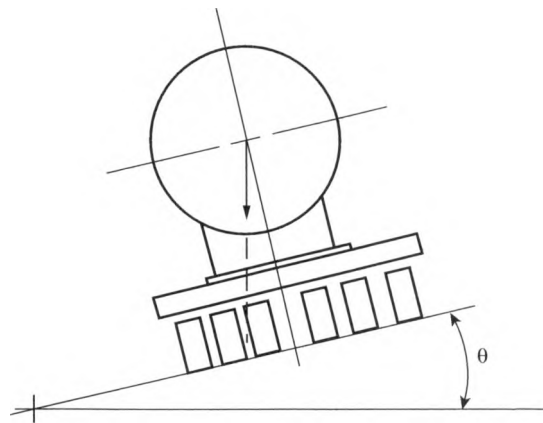


Figure 10.1. Rollover stability.

Case 2:

For bolstered loads, the vessel must swivel on the deck of the trailer in order to accommodate curves in the road and corners. As the curve or corner is negotiated, the actual CG gets further away from the projected load point. This is true whether you have a single pivoting bolster or two, however the situation is more pronounced with the double pivoting case. There have been a number of rollovers as a result of the eccentricity, ie, shifting the load to the outer row of wheels until the load becomes unstable.

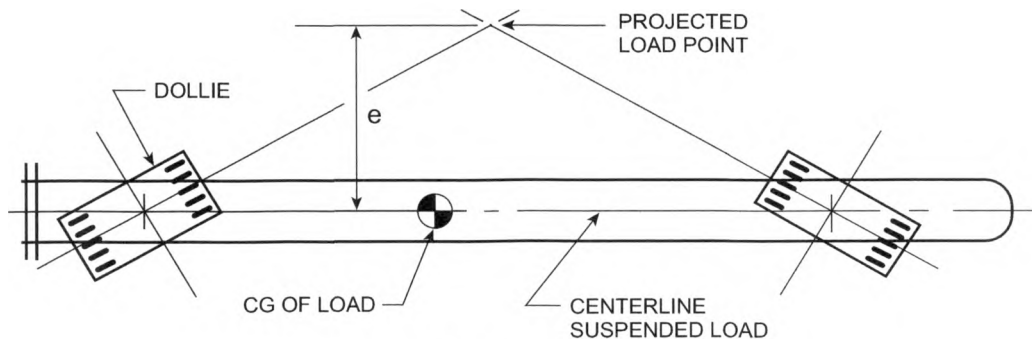
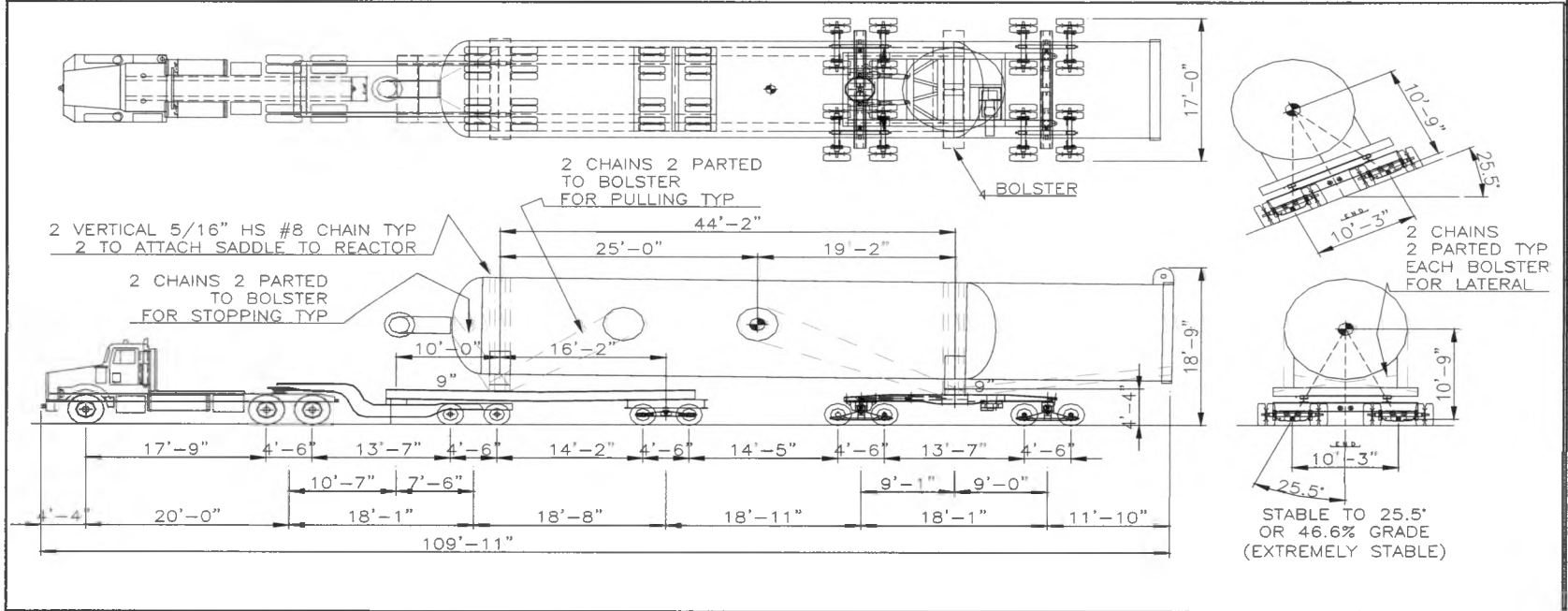


Figure 10.2. Turning stability.

AXLE LOADS - TYPICAL EXAMPLE - 11 AXLE, DUAL LANE TRANSPORTER



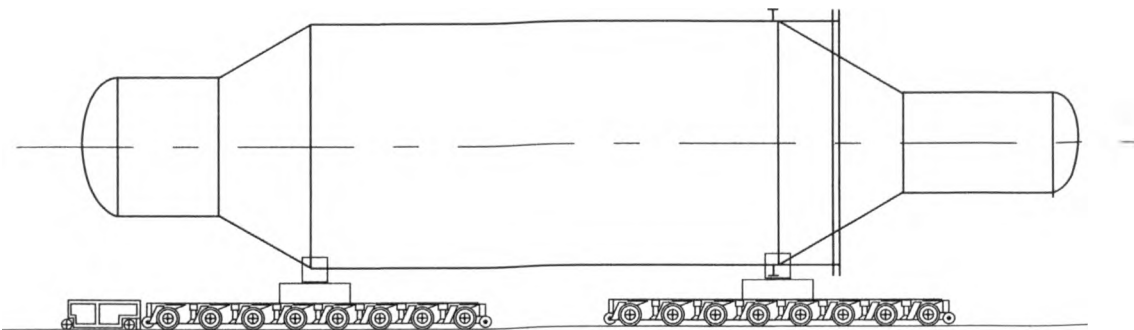
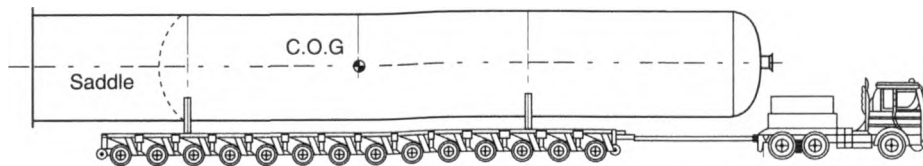
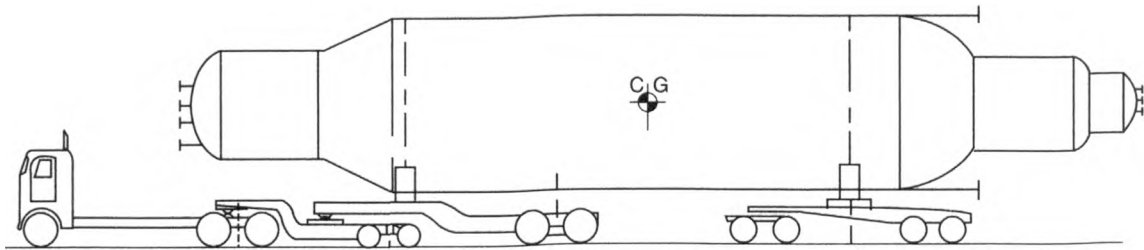
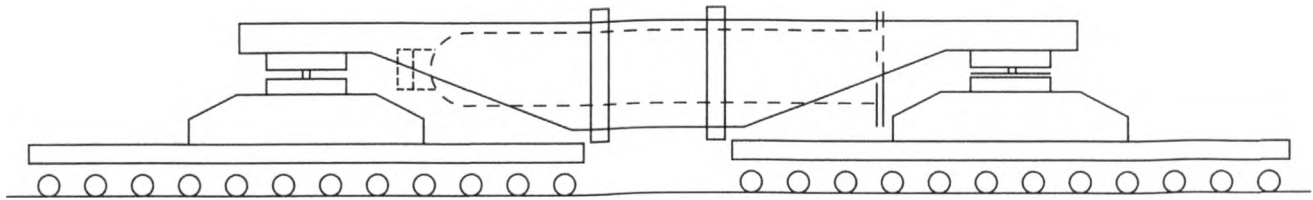
AXLE LOADS

AXLE NO.	1	2-3	4-5	6-7	8	9	10	11	TOTALS
WIDTH (ft)	8	8	10	10	17	17	17	17	
TARE WEIGHT(lbs)	12200	15360	9640	12000	5450	5450	5450	5450	71000
MAX PAYLOAD(lbs)	1510	28610	46510	43680	39498	39498	39498	39498	278300
MAX LOAD (lbs)	13710	43970	56150	55680	44948	44948	44948	44948	349300
NO. OF TIRES	2	8	16	16	8	8	8	8	74
TIRE LOADING	6855	5496	3509	3480	5618	5618	5618	5618	
ALLOW LOAD	14000	45000	58000	58000	45000	45000	45000	45000	

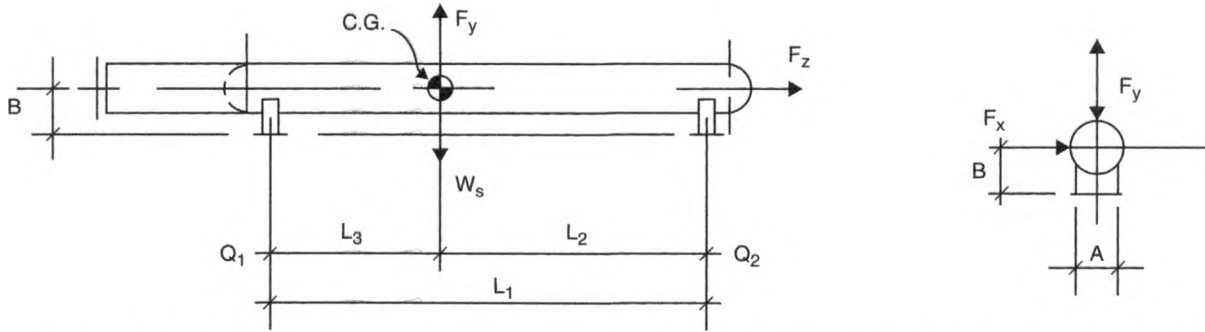
EQUIPMENT WEIGHT		DESCRIPTION	
TRACTOR	22000	ITEM	REACTOR
JEEP	9200	FOR	
BEAMS	12000	WEIGHT	278,300
DOLLY	6000	SIZE	76'-6" LX 15'-9" W X 14'-9" H
DBL DOLLY	21800	ORIGIN	
TOTAL	71000	DESTINATION	

Examples of Road Transport

If a vessel is too heavy for one trailer and too short to span two trailers, then a pair of outrigger beams can be used to span the trailers and still distribute the load to the trailers. A wide variety of trailers, self-propelled transporters, and beam configurations have been utilized for these applications. Short, squat, heavy vessels are the most common.



Summary of Loads/Forces on Vessels During Transportation



Loads $F_x, F_y, F_z = KW_s$

Verify coefficients with transport contractor/shipper.

Table 10-3
Transportation load coefficients, K

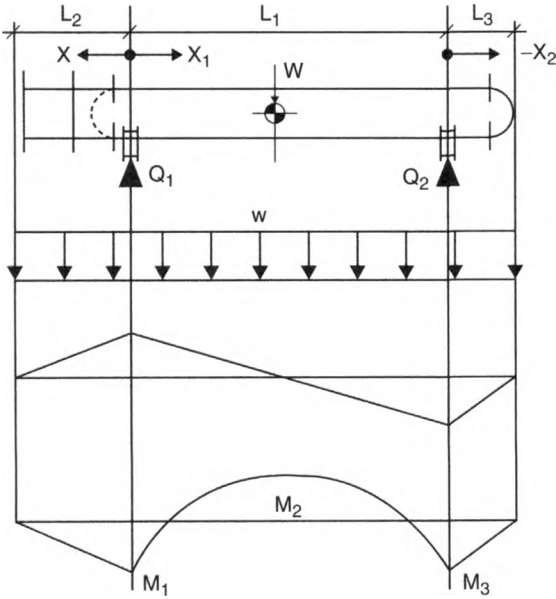
Forces	Road	Rail	Barge	Ocean
F_x	0.5	1.0	0.95	1.0
F_y	1.5	2.0	1.3	1.5
F_z	1.0	1.5	1.5	1.5

Table 10-4
Load per saddle due to transport forces

Due to ...	Load per Saddle	Diagram
F_x	$Q_1 = \frac{W_s L_2}{L_1} + \frac{F_x B}{2A}$ $Q_2 = \frac{W_s L_3}{L_1} + \frac{F_x B}{2A}$	
F_y	$Q_1 = \frac{(W_s + F_y)L_2}{L_1}$ $Q_2 = \frac{(W_s + F_y)L_3}{L_1}$	
F_z	$Q_1 = \frac{W_s L_2}{L_1} + \frac{F_z B}{L_1}$ $Q_2 = \frac{W_s L_3}{L_1} + \frac{F_z B}{L_1}$	

Load Diagrams for Moments and Forces

Case 1



Note: W = weight of vessel plus any impact factors.

$$OAL = L_1 + L_2 + L_3 \quad w = \frac{W}{OAL}$$

$$Q_1 = \frac{w[(L_1 + L_2)^2 - L_3^2]}{2L_1}$$

$$Q_2 = W - Q_1$$

$$M_1 = \frac{wL_2^2}{2}$$

$$M_2 = Q_1 \left(\frac{Q_1}{2w} - L_2 \right)$$

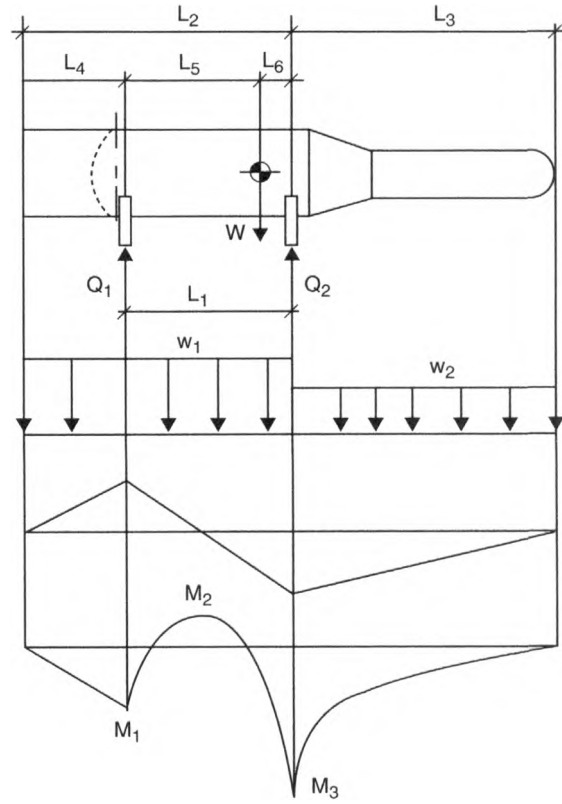
$$M_3 = \frac{wL_3^2}{2}$$

$$M_x = \frac{w(L_2 - X)^2}{2}$$

$$M_{x1} = \frac{w(L_2 + X_1)^2}{2} - Q_1 X_1$$

$$M_{x2} = \frac{w(L_3 - X_2)^2}{2}$$

Case 2



$$w_1 = \frac{W_1}{L_2}$$

$$w_2 = \frac{W_2}{L_3}$$

$$Q_1 = \frac{WL_6}{L_1}$$

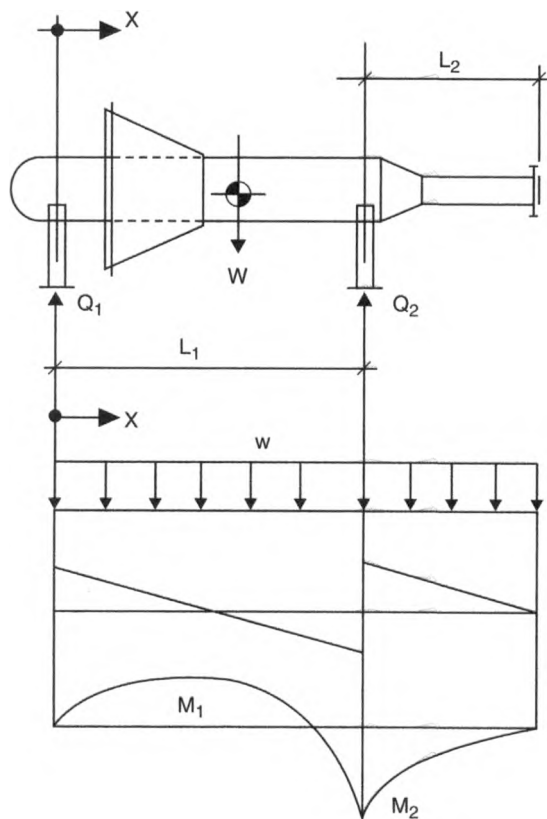
$$Q_2 = W - Q_1$$

$$M_1 = \frac{w_1 L_4^2}{2}$$

$$M_2 = \frac{M_1 + M_3}{2} - \frac{w_1 L_1^2}{8}$$

$$M_3 = \frac{w_2 L_3^2}{2}$$

Case 3



$$Q_1 = \frac{WL_1}{2(L_1 + L_2)} - \frac{WL_2^2}{2L_1(L_1 + L_2)}$$

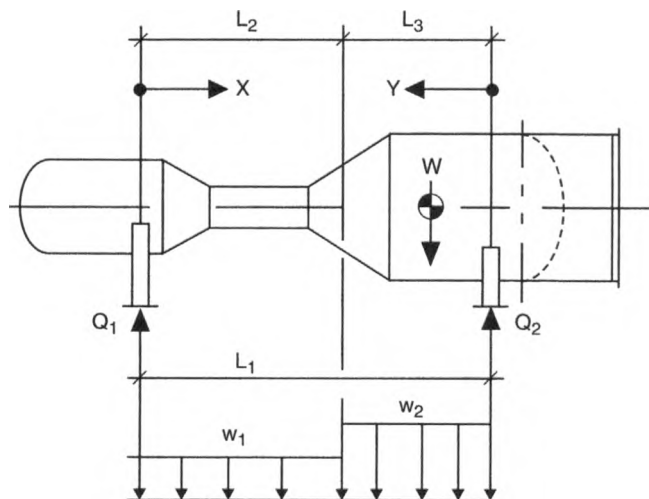
$$Q_2 = \frac{WL_1}{2(L_1 + L_2)} + \frac{WL_2}{L_1 + L_2} + \frac{WL_2^2}{2L_1(L_1 + L_2)}$$

$$M_1 = \frac{Q_1^2(L_1 + L_2)}{2W}$$

$$M_2 = Q_1L_1 - \frac{WL_1^2}{2(L_1 + L_2)}$$

$$M_x = Q_1X - \frac{(WX^2)}{2(L_1 + L_2)}$$

Case 4



$$w_1 = \frac{W_1}{L_2}$$

$$w_2 = \frac{W_2}{L_3}$$

$$Q_1 = \frac{w_1L_2(2L_1 - L_2) + w_2L_3^2}{2L_1}$$

$$Q_2 = \frac{w_2L_3(2L_1 - L_3) + w_1L_2^2}{2L_1}$$

Moment at any point X from Q₁:

$$M_x = Q_1X - \frac{w_1X^2}{2}$$

Moment at any point Y from Q₂:

$$M_y = Q_2(L_1 - Y) - \frac{w_2(L_1 - Y)^2}{2}$$

Transportation-Vertical Vessel on Two Saddles, Uniform Load Case, With Incorporation of Shipping Factors

Notation

- F_2 = Additional load on Q_2 , Lbs
- F_Z = Longitudinal loading due to shipping forces, Lbs
- F_Y = Vertical Loading due to shipping forces, Lbs
- K_Z = Longitudinal impact factor
- K_Y = Vertical impact factor
- Q_1, Q_2 = Saddle loads without impact factors, Lbs
- Q_1', Q_2' = Saddle loads with impact factors, Lbs
- W = Shipping weight of vessel without impact factors, Lbs
- W_T = Shipping weight with impact factors, Lbs
- w_1 = Uniform load, without F_Y , Lbs/Ft
- w_2 = Uniform load including F_Y , Lbs/Ft

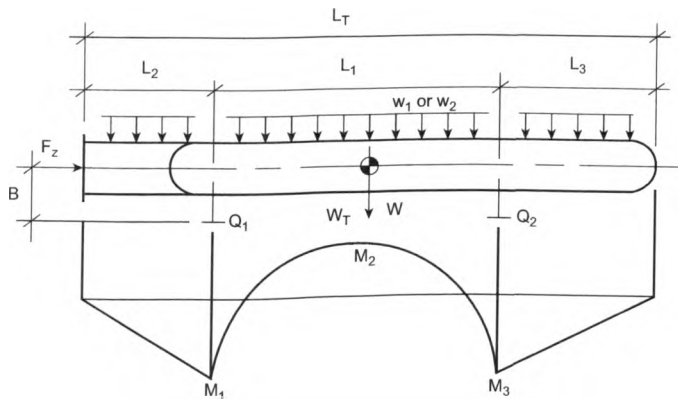


Figure 10.3. Data for uniform load case

NOTE: Assume that F_Z & F_Y do not occur at the same time

Case 1: Adding Load for F_Z

- Longitudinal load, F_Z
- $F_Z = K_Z W$
- Uniform load, w_1
- $w_1 = W/L_T$
- Additional load on saddle, F_2

$$F_2 = (F_Z B)/L_1$$

- Saddle loads, Q_1, Q_2, Q_1', Q_2'

$$Q_1 = w_1 [(L_1 + L_2)^2 - L_3^2] / 2 L_1$$

$$Q_1' = Q_1 - F_2$$

$$Q_2 = W - Q_1$$

$$Q_2' = Q_2 + F_2$$

Case 2: Adding Loads for F_Y

- Longitudinal load, F_Y

$$F_Y = K_Y W$$

- Vertical load, F_Y

$$W_T = W + F_Y$$

- Uniform load, w_2

$$w_2 = W_T/L_T$$

- Saddle loads, $Q_1' & Q_2'$

$$Q_1' = w_2 [(L_1 + L_2)^2 - L_3^2] / 2L_1$$

$$Q_2' = W_T - Q_1'$$

Select worst case and calculate moments;

$$M_1 = w_n L_2^2 / 2$$

$$M_2 = Q_1' (Q_1' / (2 w_n) - L_2)$$

$$M_3 = w_n L_3^2 / 2$$

Sample Problem

Given;

$$B = 15.75 \text{ ft}$$

$$L_1 = 124 \text{ ft}$$

$$L_2 = 24 \text{ ft}$$

$$L_3 = 21 \text{ ft}$$

$$L_T = 169 \text{ ft}$$

$$W = 741 \text{ kips}$$

$$K_Y = .5$$

$$K_Z = .6$$

Calculation

$$F_Z = K_Z W = .6 (741) = 444.6 \text{ kips}$$

$$F_Y = K_Y W = .5 (741) = 370.5 \text{ kips}$$

$$W_T = W + F_Y$$

$$= 741 + 370.5 = 1,111.5 \text{ kips}$$

$$w_1 = W/L_T = 741/169 = 4.38 \text{ kips/ft}$$

$$w_2 = W_T/L_T = 1111.5/169 = 6.58 \text{ kips/ft}$$

$$F_2 = (F_Z B)/L_1 = [444.6 (15.75)]/124$$

$$= \pm 56.47 \text{ kips}$$

Case 1: Adding Load for F_Z

- Saddle loads, Q_1 , Q_2 , Q_1' , Q_2'

$$Q_1 = w_1 \left[(L_1 + L_2)^2 - L_3^2 \right] / 2 L_1$$

$$= 4.38 \left[(124 + 24)^2 - 21^2 \right] / 2(124)$$

$$= 379 \text{ kips}$$

$$Q_1' = Q_1 - F_2 = 379 - 56.5 = 322.5 \text{ kips}$$

$$Q_2 = W - Q_1 = 741 - 379 = 362 \text{ kips}$$

$$Q_2' = Q_2 + F_2 = 362 + 56.5 = 418.5 \text{ kips}$$

Case 2: Adding Load for F_Y

- Saddle loads, Q_1' & Q_2'

$$Q_1' = w_2 \left[(L_1 + L_2)^2 - L_3^2 \right] / 2 L_1$$

$$= 6.58 \left[(124 + 24)^2 - 21^2 \right] / 2 (124)$$

$$= 569 \text{ kips}$$

$$Q_1' = W_T - Q_1' = 1111.5 - 569$$

$$= 542.5 \text{ kips}$$

Worst case is Case 2;
Determine moments...

$$M_1 = (w_2 L_2^2) / 2 = (6.58(24^2)) / 2$$

$$= 1,895 \text{ ft - kips}$$

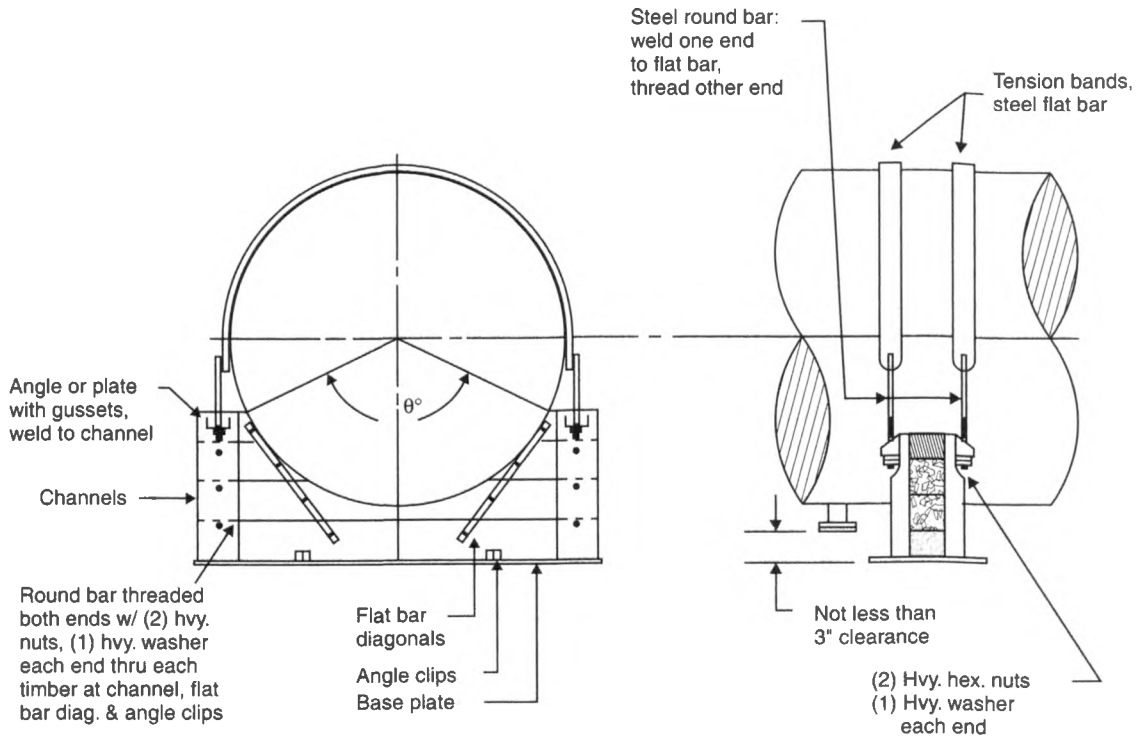
$$M_2 = Q_1'(Q_1' / (2 w_2) - L_2)$$

$$= 569 (569 / (2 \cdot 6.58) - 24) = 10,946 \text{ ft - kips}$$

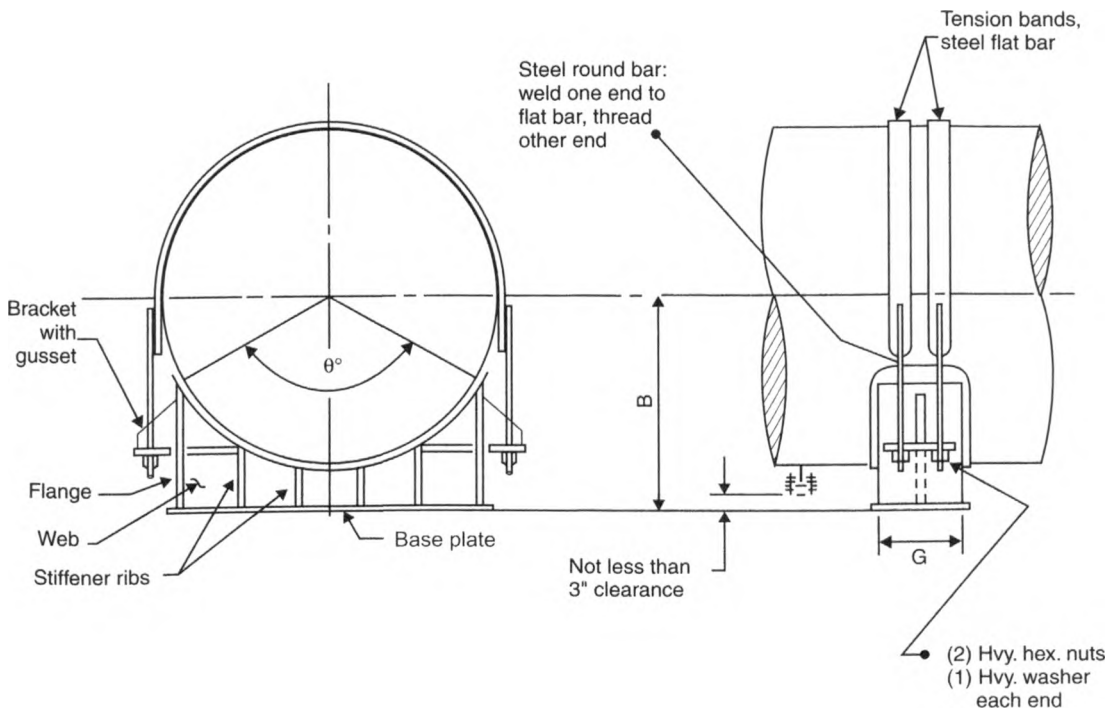
$$M_3 = (w_2 L_3^2) / 2 = (6.58(21^2)) / 2 = 1,450 \text{ ft - kips}$$

Use these moments and loads to determine stresses in shell.

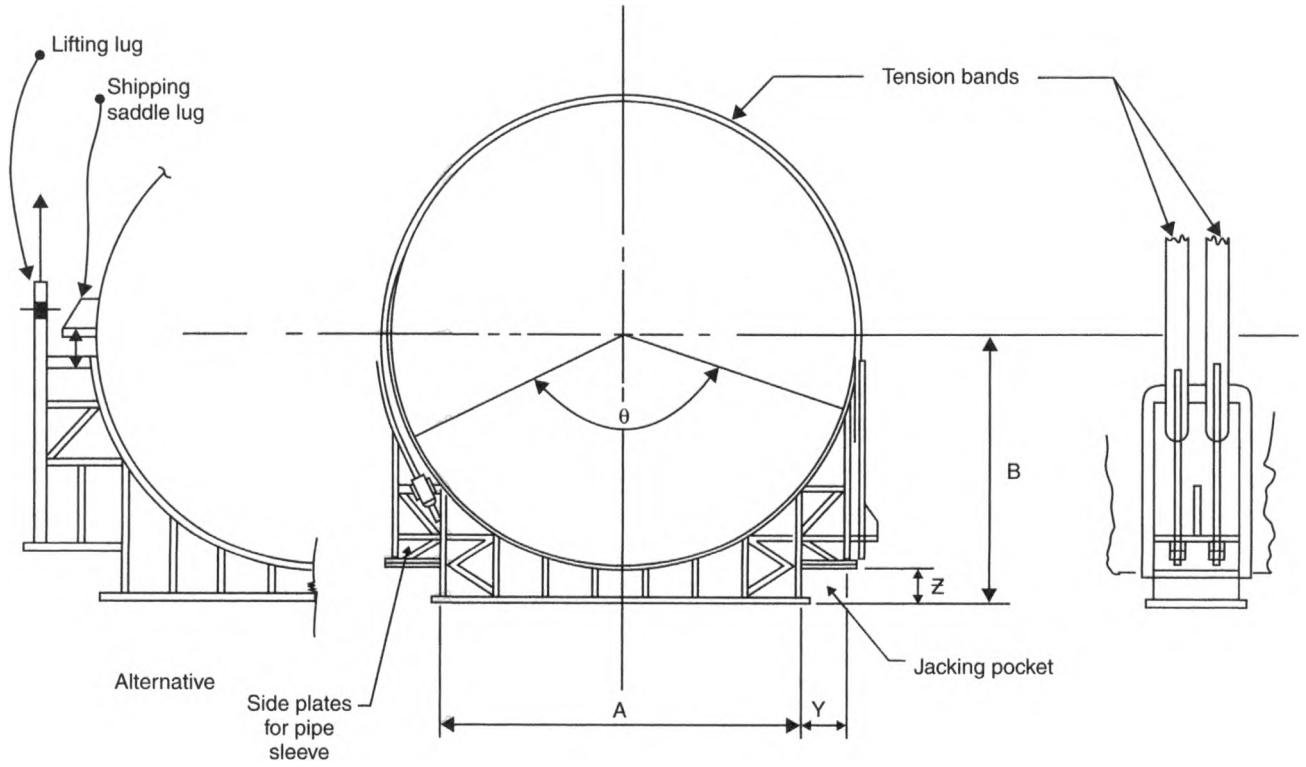
Shipping Saddles



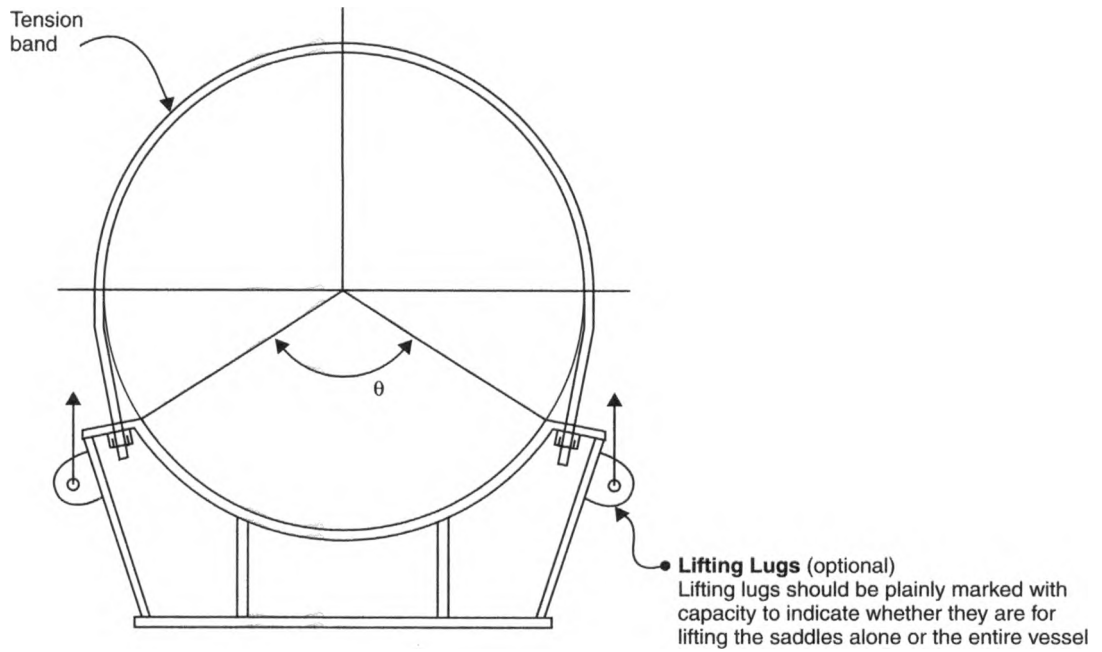
TIMBER CONSTRUCTION



STEEL CONSTRUCTION



Shipping Saddle
Steel Construction with Jacking Pocket

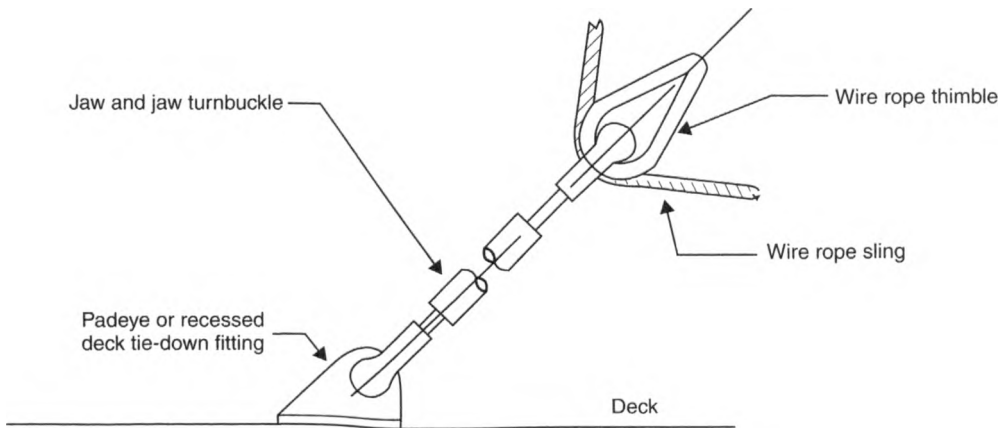
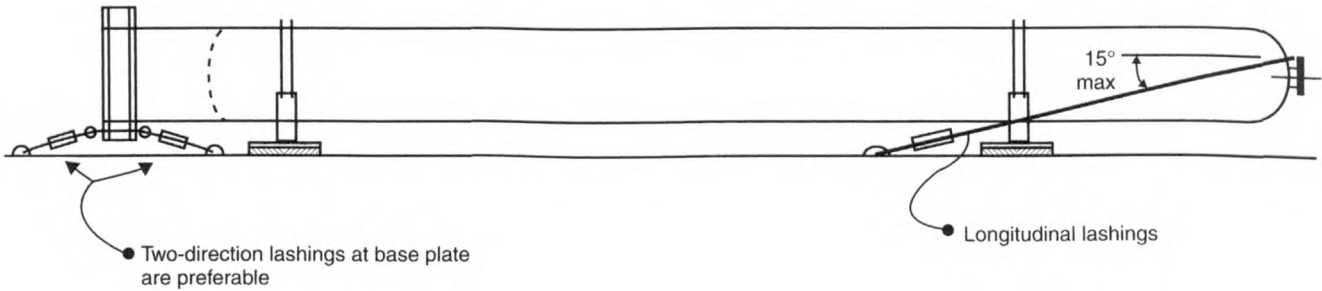
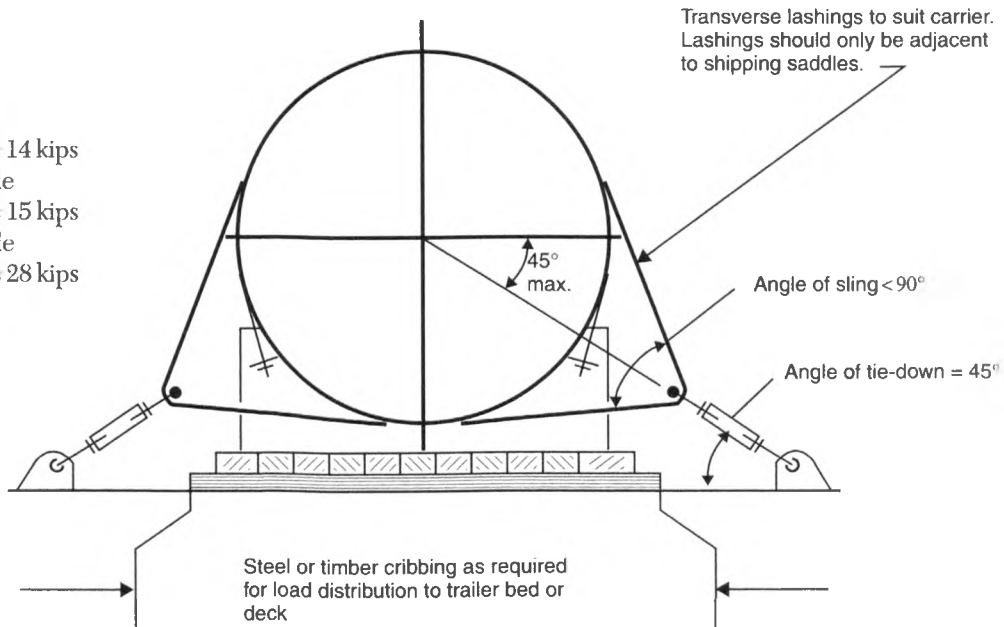


Alternate Construction

Lashing

Reference

- 1-in. wire rope
Allowable load = 14 kips
- 1¼-in. turnbuckle
Allowable load = 15 kips
- 1¾-in. turnbuckle
Allowable load = 28 kips
- 1-in. Shackle
SWL = 17 kips
- 1½-in. Shackle
SWL = 34 kips

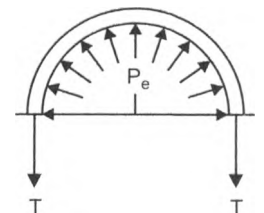
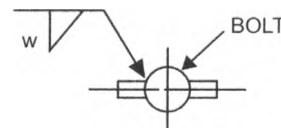
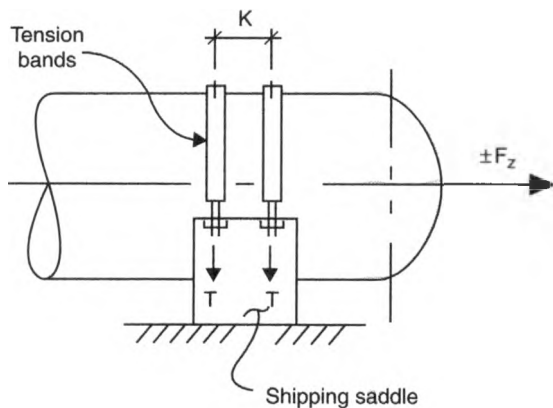
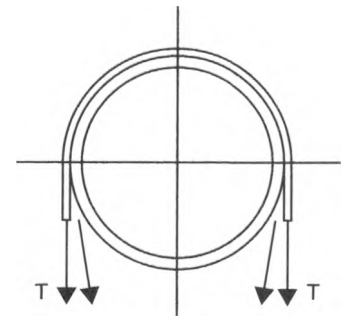
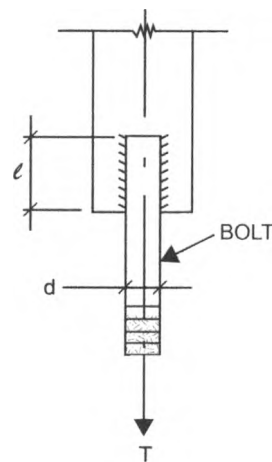
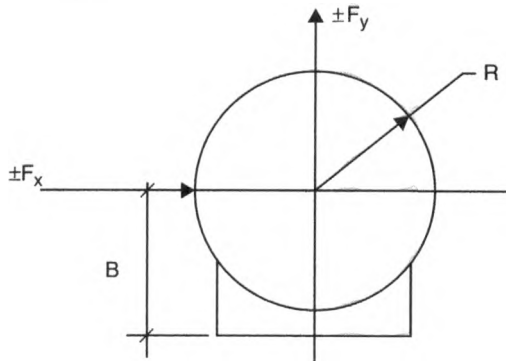
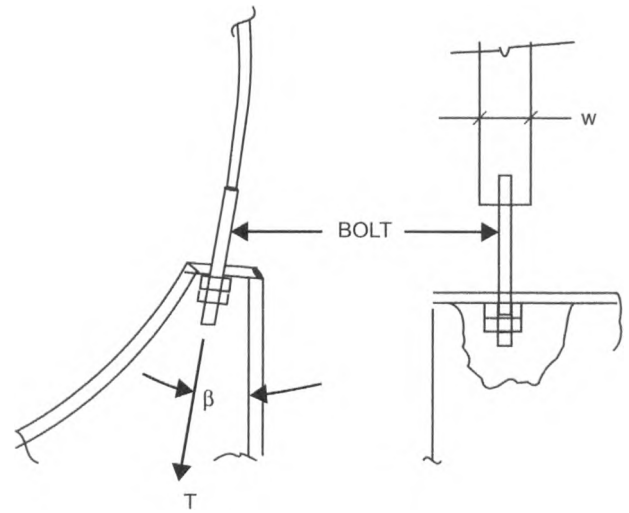


Detail of Lashing to Deck

Tension Bands on Saddles

Notation

- A_r = area required, in.²
- A_s = area of bolt, in.²
- A_b = area of band required, in.²
- A_w = allowable load on weld, lb/in.
- B = saddle height, in.
- d = bolt diameter, in.
- f = load on weld, kips/in.
- F_t = allowable stress, tension, psi
- F_x, F_y, F_z = shipping, external forces, lb
- K = maximum band spacing, in.
- N = number of bands on one saddle
- P_e = equivalent external pressure, psi
- R = outside vessel radius, in.
- T = tension load in band, lb
- $T_{1,2,3}$ = load cases in bolt and band, lb
- T_b = tension load in bolt, lb
- W_s = weight of one saddle, lb
- β = angle of tension bands, degrees
- σ_a = stress in bolt, psi
- σ_b = stress in band, psi



- Find tension in band, T_1 , due to shipping forces on saddle, F_x and F_y .

$$T_1 = \cos \beta \left(\frac{F_x B}{4RN} + \frac{F_y - W_s}{4N} \right)$$

- Area required for bolt.

$$A_r = \frac{T_1}{F_t}$$

- Find bolt diameter, d .

$$d = \sqrt{\frac{4A_r}{\pi}}$$

Select nominal bolt diameter:

$$A_s =$$

- Find maximum stress in bolt due to manual wrenching, σ_a .

$$\sigma_a = \frac{45,000}{\sqrt{d}}$$

Table 10-5
Allowable load, weld

Weld Size, w	E60XX*	E70XX*
3/16 in.	2.39	2.78
1/4 in.	3.18	3.71
5/16 in.	3.98	4.64
3/8 in.	4.77	5.57
7/16 in.	5.56	6.50

*Kips/in. of weld.

- Maximum tension load in bolts, T_2 .

$$T_2 = \sigma_a A_s$$

- Load due to saddle weight, T_3 .

$$T_3 = \frac{W_s}{2N}$$

Note: Include impact factor in weight of saddle.

- Find maximum load, T .

$$T = \text{greater of } T_1, T_2, \text{ or } T_3.$$

- Load on weld, f .

$$f = \frac{T}{4\ell}$$

- Determine size of weld from table based on load, f .

$$\text{Use } w =$$

- Maximum band spacing, K .

$$K = \frac{4\sqrt{Rt}}{1.285}$$

- Find area required for tension band, A_r .

$$A_r = \frac{T}{F_t}$$

Use:

- Check shell stresses due to force T , P_e .

$$P_e = \frac{4T}{\pi RK} < \text{ASME factor "B"}$$

Alternate Procedure

Tension Band Notation

- N = Number of bands on one saddle
- Q = Total load on one saddle, Lbs
- R = Outside vessel radius, in
- T = Tension load in band, Lbs
- β = Angle of tension bands, degrees
- K_1 = Transverse shipping coefficient
- K_2 = Vertical shipping coefficient
- X = Horizontal distance to centroid of saddle reaction, in
- Y = Vertical distance to centroid of saddle, in

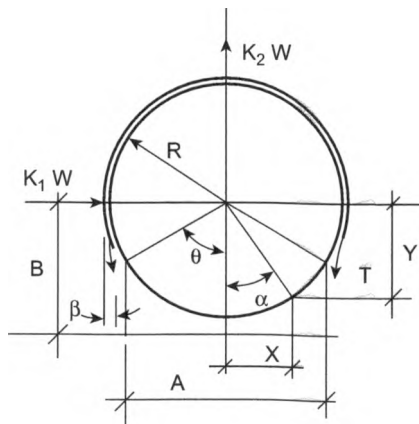


Figure 10-4. Dimensions of shipping saddle for alternate case.

Calculation

- Vertical distance to centroid of saddle, Y
 $Y = R \sin \theta / \theta$
- Find angle, α
 $\alpha = \cos^{-1} (Y/R)$

- Horizontal distance to centroid of saddle, X
 $X = R \sin \alpha$
- Tension load in band, T
 $T = [Q [(K_1 Y/X) + K_2]] / [2N \cos \beta]$

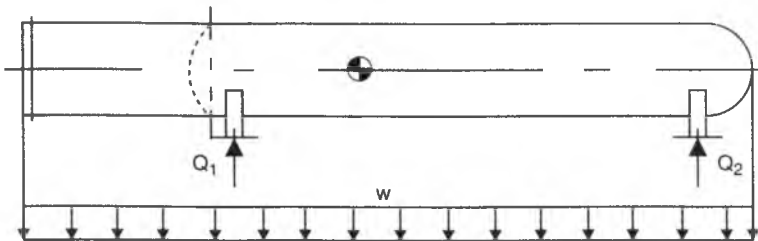
Notes

1. Vertical reaction can be a result of longitudinal load. Use largest value
2. Use $K_2 = 0$ for transverse case
3. Use $K_1 = 0$ for longitudinal case
4. Use worst case of T_1 or T_2 and design the balance of components per previous method

Example

- $K_1 = .25$
- $K_2 = .5$
- Q = 500 kips
- R = 92.5 in
- $\theta = 75^\circ = 1.308$ rad
- $\beta = 7.5^\circ$
- N = 2
- $Y = R \sin \theta / \theta$
 $Y = 92.5 \sin 75 / 1.308 = 68.3$ in
- $\alpha = \cos^{-1} (Y / R)$
 $\alpha = \cos^{-1} (68.3 / 92.5) = 42.4^\circ$
- $X = R \sin \alpha$
 $X = 92.5 (\sin 42.4) = 62.4$ in
- Transverse ($K_2 = 0$)
 $T_1 = [Q [(K_1 Y / X) + K_2]] / [2N \cos \beta]$
 $T_1 = [500 [(0.25 \cdot 68.3 / 62.4) + 0]] / [2 \cdot 2 \cos 7.5] = 34.35$ kips
- Longitudinal ($K_1 = 0$)
 $T_2 = [Q [(K_1 Y / X) + K_2]] / [2N \cos \beta]$
 $T_2 = [500 [0 + 0.5]] / [2 \cdot 2 \cos 7.5] = 63.04$ kips

Check Vessel Shell Stresses



LOAD DIAGRAM

Notation

$Z = \pi R^2 t$
 r = radius of vessel, in.
 R = radius of vessel, ft.
 b = width of saddle, in.
 $d = b + 1.56\sqrt{r}$

Stress Type	General	At Saddle 1	At Saddle 2
Longitudinal bending at saddles	—	$S_1 = \frac{M}{K_1 r^2 t}$ $S_2 = -\left(\frac{M}{K_7 r^2 t}\right)$	$S_1 = \frac{M}{K_1 r^2 t}$ $S_2 = -\left(\frac{M}{K_7 r^2 t}\right)$
Longitudinal bending at midspan	$S_3 = \frac{M}{Z}$		
Tangential shear	—	$S_7 = \frac{K_3 Q_1}{rt}$	$S_7 = \frac{K_3 Q_2}{rt}$
Circumferential stress at horn of saddle $L_1 > 8R$ $L_1 < 8R$	—	$S_9 = -\left(\frac{Q_1}{4td}\right) - \frac{3K_6 Q_1}{2t^2}$ $S_{10} = -\left(\frac{Q_1}{4td}\right) - \frac{12K_6 Q_1 R}{L_1 t^2}$	$S_9 = -\left(\frac{Q_2}{4td}\right) - \frac{3K_6 Q_2}{2t^2}$ $S_{10} = -\left(\frac{Q_2}{4td}\right) - \frac{3K_6 Q_2 R}{L_1 t^2}$
Circumferential compression	—	$S_{12} = -\left(\frac{K_5 Q_1}{td}\right)$	$S_{12} = -\left(\frac{K_5 Q_2}{td}\right)$

- Notes:
1. Also check shell stresses at each change of thickness and diameter.
 2. See procedure for the design of saddles for horizontal vessels for a detailed description of shell stresses and for values of coefficients K_1 through K_7 .
 3. Values of M and Q should be determined from the previous pages at the applicable location.
 4. Allowable stresses:

Tension: $0.9F_y$
 Compression: $1.2 \times$ Factor "B" from ASME Code

Procedure 10-2: Erection of Pressure Vessels

The designer of pressure vessels and similar equipment will ultimately become involved in the movement, transportation, and erection of that equipment. The degree of that involvement will vary due to the separation of duties and responsibilities of the parties concerned. It is prudent, however, for the designer to plan for the eventuality of these events and to integrate these activities into the original design. If this planning is done properly, there is seldom a problem when the equipment gets to its final destination. Conversely, there have been numerous problems encountered when proper planning has not been done.

There is also an economic benefit in including the lifting attachments in the base vessel bid and design. These lifting attachments are relatively inexpensive in comparison to the overall cost of the vessel and minuscule compared to the cost of the erection of the equipment. The erection alone for a major vessel can run into millions of dollars. If these attachments are added after the purchase order is awarded, they can become expensive extras.

There are also the consequences to life, property, and schedules if this activity is not carried out to a successful conclusion. Compared to the fabricated cost of the lifting attachments, the consequences to life, property, and schedule are too important to leave the design of these components and their effect on the vessel to those not fully versed in the design and analysis of pressure vessels.

In addition, it is important that the designer of the lifting attachments be in contact with the construction organization that will be executing the lift. This ensures that all lifting attachments meet the requirements imposed by the lifting equipment. There are so many different methods and techniques for the erection of vessels and the related costs of each that a coordinated effort between the designer and erector is mandatory. To avoid surprises, neither the designer nor the erector can afford to work in a vacuum. To this end, it is not advisable for the vessel fabricator to be responsible for the design if the fabricator is not the chief coordinator of the transport and erection of the vessel.

Vessels and related equipment can be erected in a variety of ways. Vessels are erected by means of single cranes, multiple cranes, gin poles, jacking towers, and other means. The designer of the lifting attachments should not attempt to dictate the erection method by the types of attachments that are designed for the vessel. The

selection of one type of attachment versus another could very well do just that.

Not every vessel needs to be designed for erection or have lifting attachments. Obviously the larger the vessel, the more complex the vessel, the more expensive the vessel, the more care and concern that should be taken into account when designing the attachments and coordinating the lift. The following listing will provide some guidelines for the provision of special lifting attachments and a lifting analysis to be done. In general, provide lifting attachments for the following cases:

- Vessels over 50,000 lb (25 tons).
- Vessels with L/D ratios greater than 5.
- Vertical vessels greater than 8 ft in diameter or 50 ft in length.
- Vessels located in a structure or supported by a structure.
- High-alloy or heat-treated vessels (since it would not be advisable for the field to be doing welding on these vessels after they arrive on site, and wire rope slings could contaminate the vessel material).
- Flare stacks.
- Vessels with special transportation requirements.

At the initial pick point, when the vessel is still horizontal, the load is shared between the lifting lugs and the tail beam or lug, based on their respective distances to the vessel center of gravity. As the lift proceeds, a greater percentage of the load is shifted to the top lugs or trunnions until the vessel is vertical and all of the load is then on the top lugs. At this point the tail beam or shackle can be removed.

During each degree of rotation, the load on the lugs, trunnions, tailing device, base ring, and vessel shell are continually varying. The loads on the welds attaching these devices will also change. The designer should evaluate these loadings at the various lift angles to determine the worst coincident case.

The worst case is dependent on the type of vessel and the type of attachments. For example, there are three types of trunnions described in this procedure. There is the bare trunnion (Type 3), where the wire rope slides around the trunnion itself. While the vessel is in the horizontal position (initial pick point), the load produces a circumferential moment on the shell. Once the vessel is in the upright position, the same load produces

a longitudinal moment in the shell. At all the intermediate angles of lift there is a combination of circumferential and longitudinal moments. The designer should check the two worst cases at 0° and 90° and several combinations in between.

The same trunnion could have a lifting lug welded to the end of the trunnion (Type 1). This lug also produces circumferential and longitudinal moments in the shell. However, in addition this type of lug will produce a torsional moment on the shell that is maximum at 0° and zero at 90° of angular rotation. The rotating lug (Type 2) eliminates any torsional moment.

There is one single lift angle that will produce the maximum stress in the vessel shell but no lift angle that is the worst for all vessels. The worst case is dependent on the type of lift attachments, distances, weights, and position relative to the center of gravity.

The minimum lift location is the lowest pick point that does not overstress the overhanging portion of the vessel. The maximum lift location is the highest pick point that does not overstress the vessel between the tail and pick points. These points become significant when locating the lift points to balance the stress at the top lug, the overhang, and the midspan stress.

The use of side lugs can sometimes provide an advantage by reducing the buckling stress at midspan and the required lift height. Side lugs allow for shorter boom lengths on a two-crane lift or gin poles. A shorter boom length, in turn, allows a higher lift capacity for the cranes. The lower the lug location on the shell, the shorter the lift and the higher the allowable crane capacity. This can translate into dollars as crane capacity is affected. The challenge from the vessel side is the longitudinal bending due to the overhang and increased local shell stresses. All of these factors must be balanced to determine the lowest overall cost of an erected vessel.

Requirement for Erection and Setting of Vertical Vessels

The following is a brief synopsis of general recommendations regarding the setting, leveling and shimming of vertical vessels. The following should be considered as guidelines only. There are no codes or standards that are applied. In general, company specifications contain contract requirements for the contractors scope of supply or duties. The following lists help to clarify general construction practices with regard to the setting of vertical vessels and towers.

Contractor Duties

1. Prepare tops of foundations (bush hammer if required)
2. Perform surveying as required to establish centerlines, sole plate or shim elevations at bottom of base of equipment
3. Shimming
4. Erect equipment
5. Level/plumbing
6. Final alignment
7. Grouting
8. Bolting

Tolerances

Out of vertical tolerance for vertical vessels, unless specified otherwise, shall be 0.1% of the vessel height, or about ¼ inch for every 20 feet to a maximum of ¾ inches.

Soleplates (also called bearing pads, leveling plates or embedments)

Soleplates are stainless steel plates, 0.5 inches to 0.75 inches thick, set in grout, on top of the foundation at the exact height of the underside of the base plate. As a rule, two soleplates should be installed per anchor bolt, one on each side of the bolt. Depending on the tower diameter, and the distance between the anchor bolts, another soleplate may be installed between adjacent anchor bolts. The dimensions of the soleplates will vary according to the width of the vessel base ring and vessel weight. Soleplates are supported in place by a mixture of Portland cement and sand in proportions 1:3. The vessel should not be erected until the soleplates have been in place for 28 days to allow for concrete curing. Shims and soleplates will remain in place after the grouting operation.

Shims

Shims are used to provide precise leveling of the vessel. Shim packs may be grouted into the foundation in lieu of sole plates but this practice is unusual. Typically, shims are used on top of the sole plates for the leveling operation. Special shims may be required for unique applications such as a large vessel supported on a braced frame structure with minimal contact/bearing at each support

point. The following are some guidelines for the use of shims.

1. Shims. If left in place, shall be stainless steel
2. Shims must have rounded corners
3. Shims will be fixed in place
4. Shims shall be deburred
5. Shims shall be full bearing
6. Shims may be horseshoe type
7. Shims thinner than 0.001 inches are not allowed
8. Shims with holes are not allowed
9. Shims should be the full width of the base plate

Leveling/Straightness/Plumbness

After the vessel has been placed on its foundation it must be checked to be certain it is vertical and plumb. Leveling is normally checked by use of two theodolites, 90 degrees apart. The theodolites shall be spaced an adequate distance from the vessel to allow visual field of the entire height of the vessel. Adjustments can be made to the vessel alignment by means of wedges, either powered or not, and then shimmed. The wedges should not be left in place after shimming.

The vessel may be heated by the sun to a higher temperature on one side than the other. This can create a slight "banana" effect which should be taken into account when checking levelness. The equation for calculating the deflection from this effect is as follows:

$$\zeta = [\pi D^2 t H^2 \alpha \Delta T] / 8 I$$

where;

ζ = Deflection, in

D = Diameter, ft

T = Thickness, in

H = Height, ft

α = Coefficient of thermal expansion, in/in/°F

ΔT = Temperature difference from one side of the column to the other, °F

I = Moment of inertia of vessel cross section, ft⁴

Bolting

After the vessel is aligned and shimmed, the nuts on the anchor bolts must be tightened. The vessel should not be

left standing without the crane attached unless all anchor bolts have been tightened.

The anchor bolts should not be tightened to their maximum load until the drypacking under the base plate is complete. At this stage, the base plate is suspended between the soleplates until the drypack is installed. Since the soleplates straddle the anchor bolts, there is a chance of deforming the baseplate prior to the installation of the drypack, if the anchor bolts are over tightened.

After drypacking, the anchor bolts should be tightened to the correct torque to produce the maximum allowable bolt stress. The anchor bolts should not be tightened beyond the point of maximum allowable bolt stress.

Note that the initial anchor bolt tension does not increase the maximum bolt tension caused by wind or earthquake. This initial tension will only clamp the base ring to the concrete. Both are in equal compression until the external load is applied. The external load reduces the compression in the concrete before additional load is applied to the bolts. After the external load overcomes all the compression in the concrete, the stress in the bolt will increase to the value it would have been, had there been no initial tension.

Grouting

Grout under base plates shall provide full uniform load transfer between the bottom of the base plate and the top of the foundation. Load transfer to the foundation must be through the grout, not through the shims or soleplate.

Prior to setting of the vessel, the top of the foundation should be bush hammered and cleaned. This ensures that the grout will adhere to the surface of the foundation. Bush hammering may be done strictly under the base plate or across the entire top of the foundation.

Once the vessel is leveled, shimmed and bolted it is ready to be grouted. Grouting shall consist of filling the void area between the top of foundation and the underside of base plate with cementitious grout. The grout shall be installed in accordance with the manufacturers recommendations and any applicable contract specifications.

Depending on the type of grout to be used, grout dams may be used.

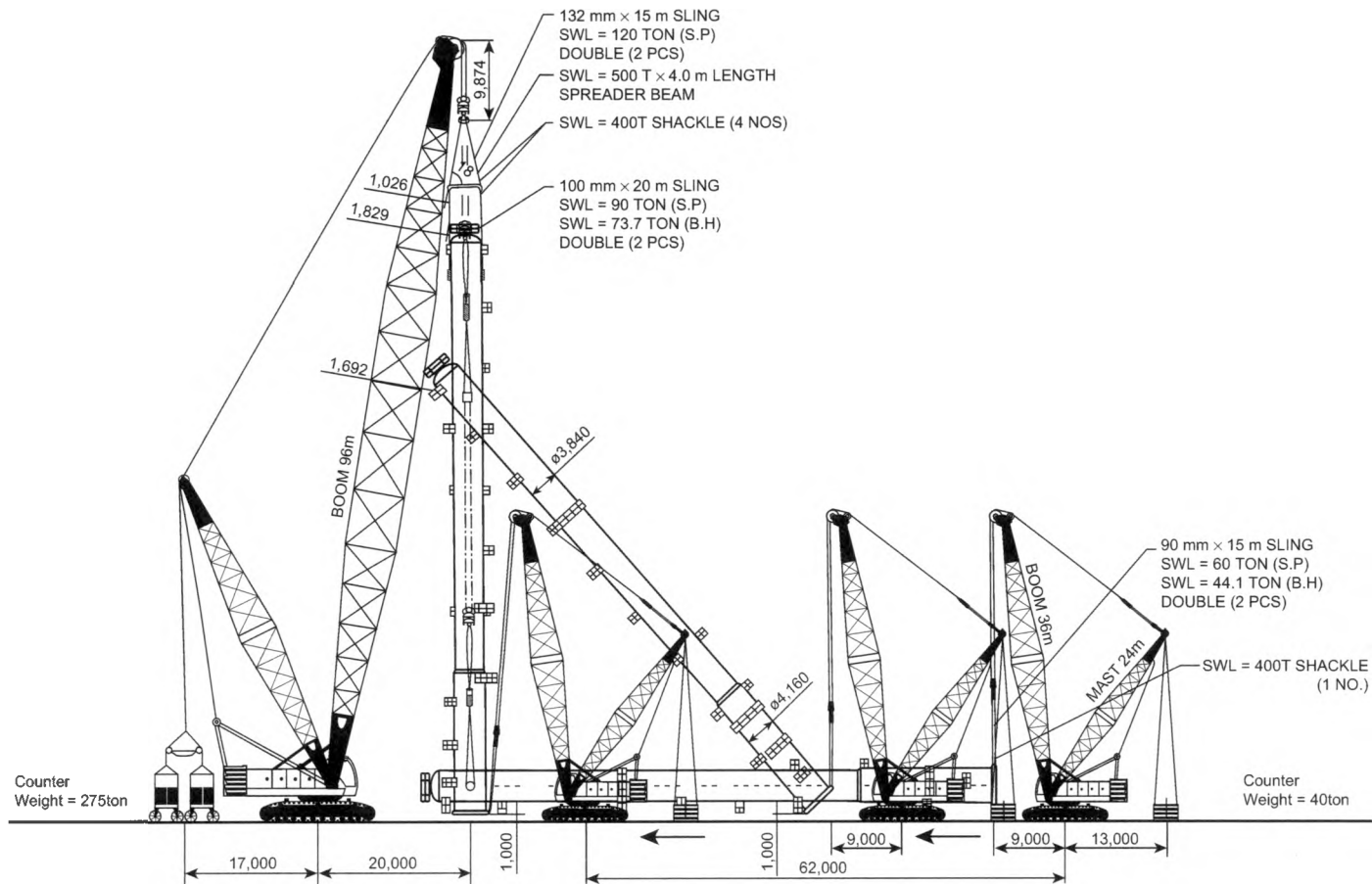


Figure 10-5. Typical example of erection study.

Steps in Design

Given the overall weight and geometry of the vessel and the location of the center of gravity based on the erected weight, apply the following steps to either complete the design or analyze the design.

Step 1: Select the type of lifting attachments as an initial starting point:

Lift end (also referred to as the "pick end"):

- a. *Head lug*: Usually the simplest and most economical, and produces the least stress.
- b. *Cone lug*: Similar to a head lug but located at a conical transition section of the vessel.
- c. *Side lug*: Complex and expensive.
- d. *Top flange lug*: The choice for high-pressure vessels where the top center flange and head are very rigid. This method is uneconomical for average applications.
- e. *Side flange lug*: Rarely used because it requires a very heavy nozzle and shell reinforcement.
- f. *Trunnions*: Simple and economical. Used on a wide variety of vessels.
- g. Other.

Tail end:

- a. Tail beam.
- b. Tail lug.
- c. Choker (cinch); see later commentary.

Tailing a column during erection with a wire rope choker on the skirt above the base ring is a fairly common procedure. Most experienced erectors are qualified to perform this procedure safely. There are several advantages to using a tailing choker:

- Saves material, design, detailing, and fabrication.
- Simplifies concerns with lug and shipping orientations.
- May reduce overall height during transportation.

There are situations and conditions that could make the use of a tailing choker impractical, costly, and possibly unsafe. Provide tailing lugs or a tailing beam if:

- The column is more than about 10 feet in diameter. The larger the diameter, the more difficult it is for the wire rope to cinch down and form a good choke on the column.
- The tail load is so great that it requires the use of slings greater than about 1½ inches in diameter. The larger the diameter of the rope, the less flexible it is and the more likely that it could slip up unexpectedly during erection.

Step 2: Determine the forces T and P for all angles of erection.

Step 3: Design/check the lifting attachments for the tailing force, T, and pick force, P.

Step 4: Design/check the base ring assembly for stresses due to tailing force, T.

Step 5: Determine the base ring stiffening configuration, if required, and design struts.

Step 6: Check shell stresses due to bending during lift. This would include midspan as well as any overhang.

Step 7: Analyze local loads in vessel shell and skirt due to loads from attachments.

Allowable Stresses

Per AISC:

Tension

$$\begin{aligned}
 F_t &= 0.6F_y \text{ on gross area} \\
 &= 0.5F_u \text{ on effective net area} \\
 &= 0.45F_y \text{ for pin-connected members}
 \end{aligned}$$

Compression

(for short members only)

$$\begin{aligned}
 F_c &= \text{Use buckling value.} \\
 &= \text{for vessel shell: } 1.33 \times \text{ASME Factor "B"}
 \end{aligned}$$

Shear

$$\begin{aligned}
 F_s &= \text{Net area of pin hole: } 0.45F_y \\
 &= \text{other than pin-connected members: } 0.4F_y \\
 &= \text{fillet welds in shear:} \\
 &\quad \text{E60XX: } 9600 \text{ lb/in. or } 13,600 \text{ psi} \\
 &\quad \text{E70XX: } 11,200 \text{ lb/in. or } 15,800 \text{ psi}
 \end{aligned}$$

Bending

$$F_b = 0.6F_y \text{ to } 0.75F_y, \text{ depending on the shape of the member}$$

Bearing

$$F_p = 0.9F_y$$

Combined

Shear and tension:

$$\frac{\sigma_a}{F_a} + \frac{\tau}{F_s} \leq 1$$

Tension, compression and bending:

$$\frac{\sigma_a}{F_a} + \frac{\sigma_b}{F_b} \leq 1 \text{ or } \frac{\sigma_T}{F_T} + \frac{\sigma_b}{F_b} \leq 1$$

Note: Custom-designed lifting devices that support lifted loads are generally governed by ASME B30.20 "Below the hook lifting devices." Under this specification, design stresses are limited to $F_y/3$. The use of AISC allowables with a load factor of 1.8 or greater will generally meet this requirement.

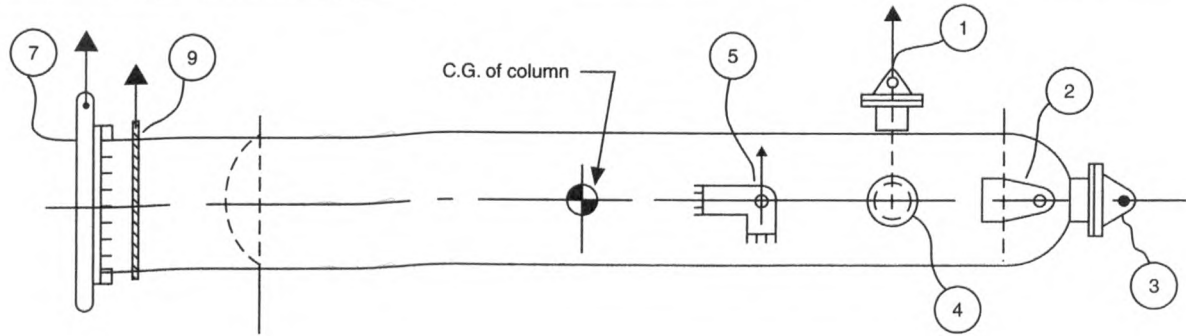
Notation

A = area, in.²
 A_a = area, available, in.²
 A_b = area, bolt, in.²
 A_n = net cross-sectional area of lug, in.²
 A_p = area, pin hole, in.²
 A_r = area, required, in.²
 A_s = area, strut, in.² or shear area of bolts
 C = lug dimension, see sketch
 D_o = diameter, vessel OD, in.
 D₁ = diameter, lift hole, in.
 D₂ = diameter, pin, in.
 D₃ = diameter, pad eye, in.
 D_{sk} = diameter, skirt, in.
 D_m = mean vessel diameter, in.
 E = modulus of elasticity, psi
 f_r = tail end radial force, lb
 f_L = tail end longitudinal force, lb
 f_s = shear load, lb or lb/in.
 F_a = allowable stress, combined loading, psi
 F_b = allowable stress, bending, psi
 F_c = allowable stress, compression, psi
 F_p = allowable stress, bearing pressure, psi
 F_s = allowable stress, shear, psi
 F_t = allowable stress, tension, psi
 F_y = minimum specified yield stress, psi
 I = moment of inertia, in.⁴
 J_w = polar moment of inertia of weld, in.⁴
 K = end connection coefficient
 K_L = overall load factor combining impact and safety factors, 1.5–2.0
 K_i = impact factor, 0.25–0.5
 K_r = internal moment coefficient in circular ring due to radial load
 K_s = safety factor
 K_T = internal tension/compression coefficient in circular ring due to radial load
 L_s = length of skirt/base stiffener/strut, in.

M = moment, in.-lb
 M_b = bending moment, in.-lb
 M_C = circumferential moment, in.-lb
 M_L = longitudinal moment, in.-lb
 M_T = torsional moment, in.-lb
 N_b = number of bolts used in tail beam or flange lug
 N = width of flange of tail beam with a web stiffener (N = 1.0 without web stiffener)
 n_L = number of head or side lugs
 P = pick end load, lb
 P_e = equivalent load, lb
 P_L = longitudinal load per lug, lb
 P_r = radial load, lb
 P_T = transverse load per lug, lb
 R_b = radius of base ring to neutral axis, in.
 r = radius of gyration of strut, in.
 R_c = radius of bolt circle of flange, in.
 S_u = minimum specified tensile stress of bolts, psi
 t_b = thickness of base plate, in.
 t_g = thickness of gusset, in.
 t_L = thickness of lug, in.
 t_p = thickness of pad eye, in.
 t_s = thickness of shell, in.
 T = tail end load, lb
 T_b = bolt pretension load, lbs
 T_t = tangential force, lb
 w₁ = fillet weld size, shell to re-pad
 w₂ = fillet weld size, re-pad to shell
 w₃ = fillet weld size, pad eye to lug
 w₄ = fillet weld size, base plate to skirt
 w₅ = uniform load on vessel, lb/in.
 W_E = design erection weight, lb
 W_L = erection weight, lb
 Z = section modulus, in.³
 α = angular position for moment coefficients in base ring, clockwise from 0°
 β = angle between parallel beams, degrees
 σ = stress, combined, psi
 σ_b = stress, bending, psi
 σ_p = stress, bearing, psi
 σ_c = stress, compression, psi
 σ_{cr} = critical buckling stress, psi
 σ_T = stress, tension, psi
 τ = shear stress, psi
 τ_T = torsional shear stress, psi
 θ = lift angle, degrees
 θ_B = minimum bearing contact angle, degrees
 θ_H = sling angle to lift line, horizontal, degrees
 θ_v = sling angle to lift line, vertical, degrees

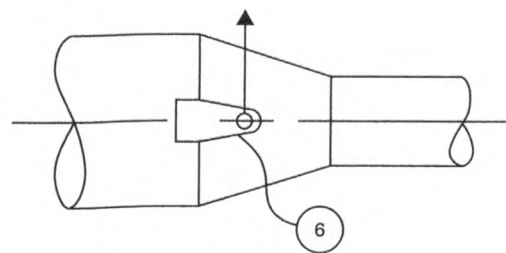
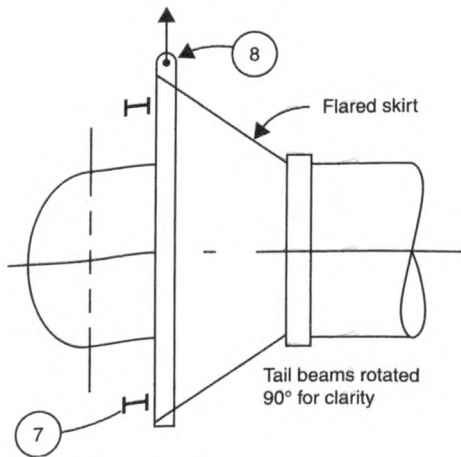
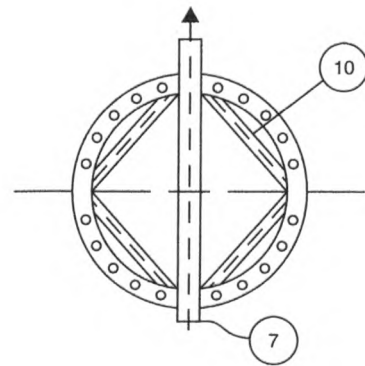
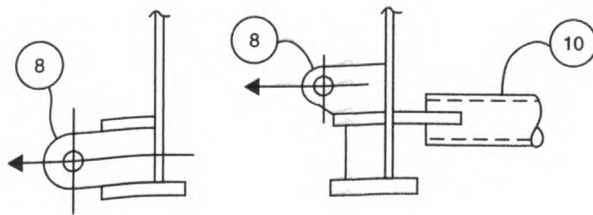
Procedure 10-3: Lifting Attachments and Terminology

Types of Lifting Attachments



Tail Lift End

Lift or Pick End



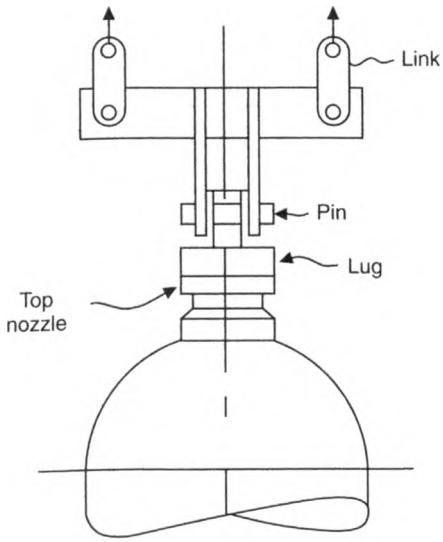
LIFT END OPTIONS

1. Shell flange lug
2. Top head lug
3. Top flange lug
4. Trunnion
5. Side lug
6. Cone lug

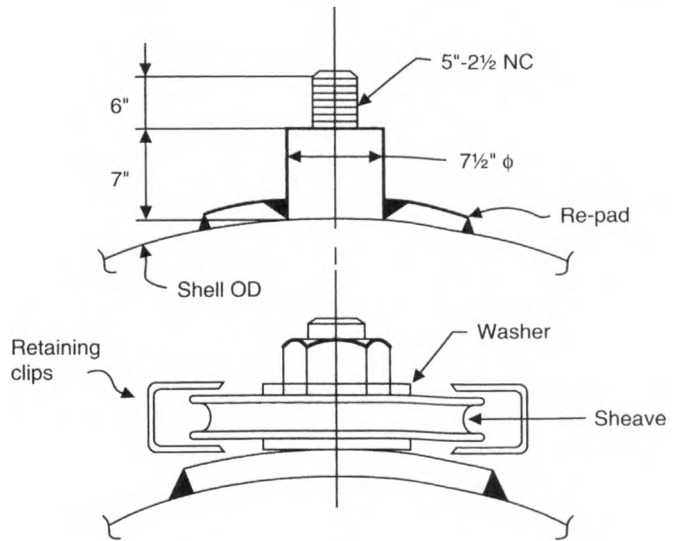
TAIL LIFT OPTIONS

7. Tail beam
8. Tail lug
9. Choker (sling)
10. Base ring stiffener

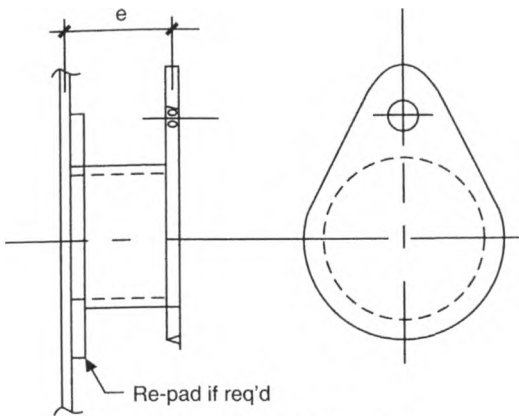
Miscellaneous Lifting Attachments



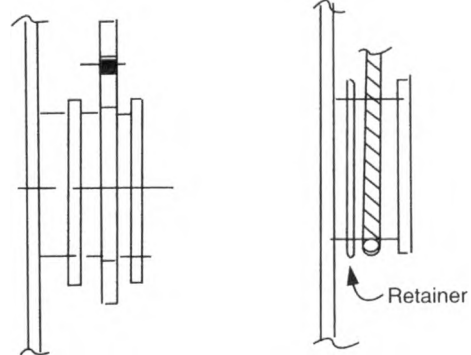
TOP FLANGE LUG WITH SPREADER



SHEAVE ASSEMBLY DIRECT MOUNT—200 TON

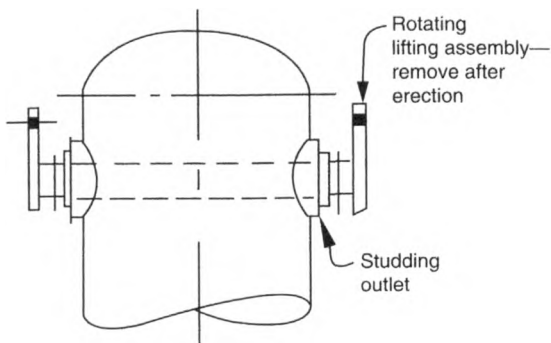


TRUNNION WITH FIXED LUG TYPE 1

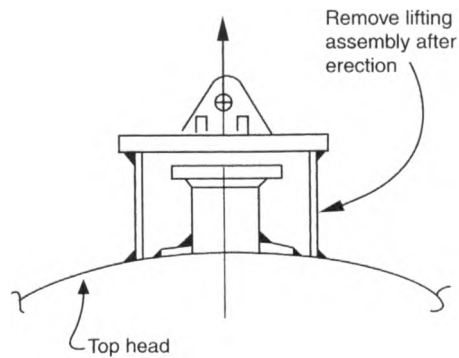


TRUNNION WITH ROTATING LUG TYPE 2

TRUNNION WITHOUT LUG TYPE 3



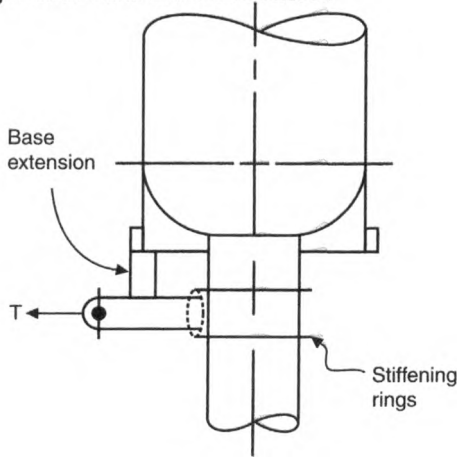
STIFFENER PIPE THROUGH STUDDING OUTLETS—BLIND AFTER ERECTION



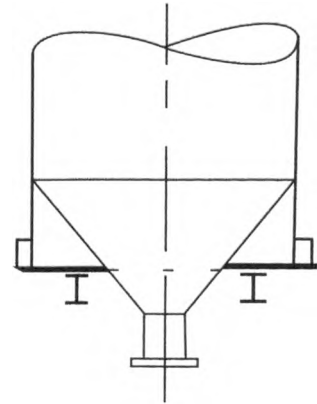
TEMPORARY TOP HEAD FLANGE LUG OVER TOP NOZZLE

Tailing Devices for Vessels with Chambers Projecting through Skirt

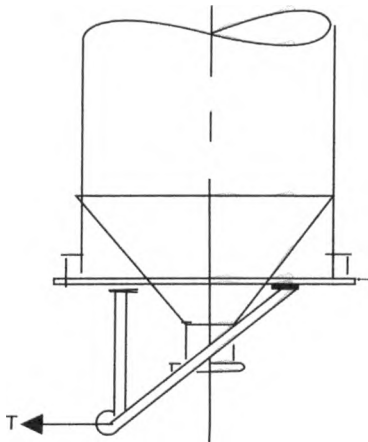
Utilize Projection with Base Extension



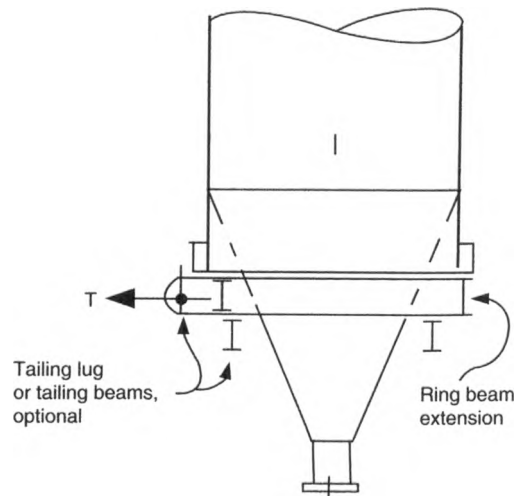
Parallel Tailing Beams Without Skirt Stiffeners



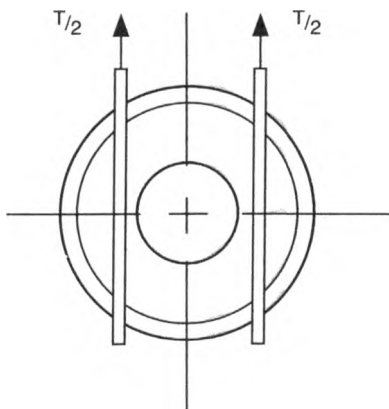
Frame-Type Tailing Device



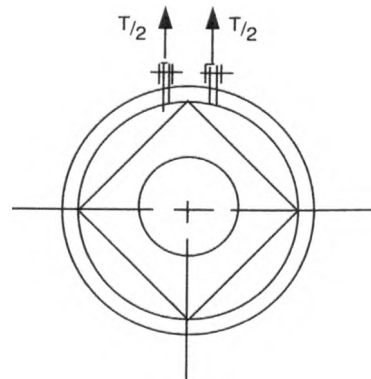
Bolted on Ring Beam Extension



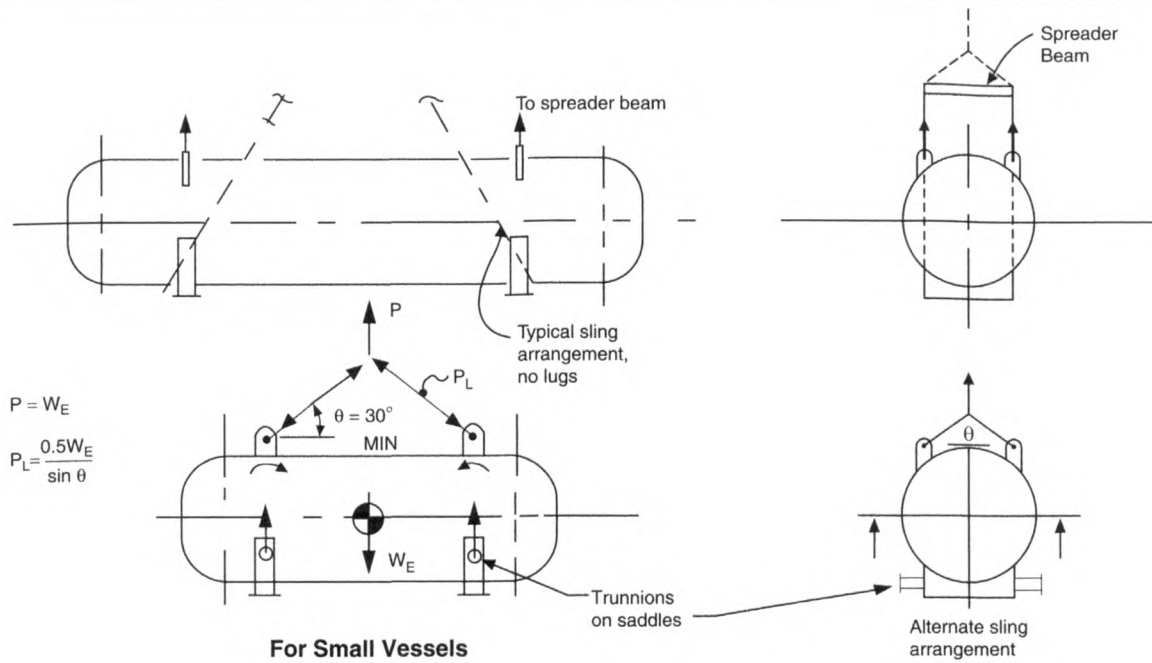
Base with Parallel Tailing Beams



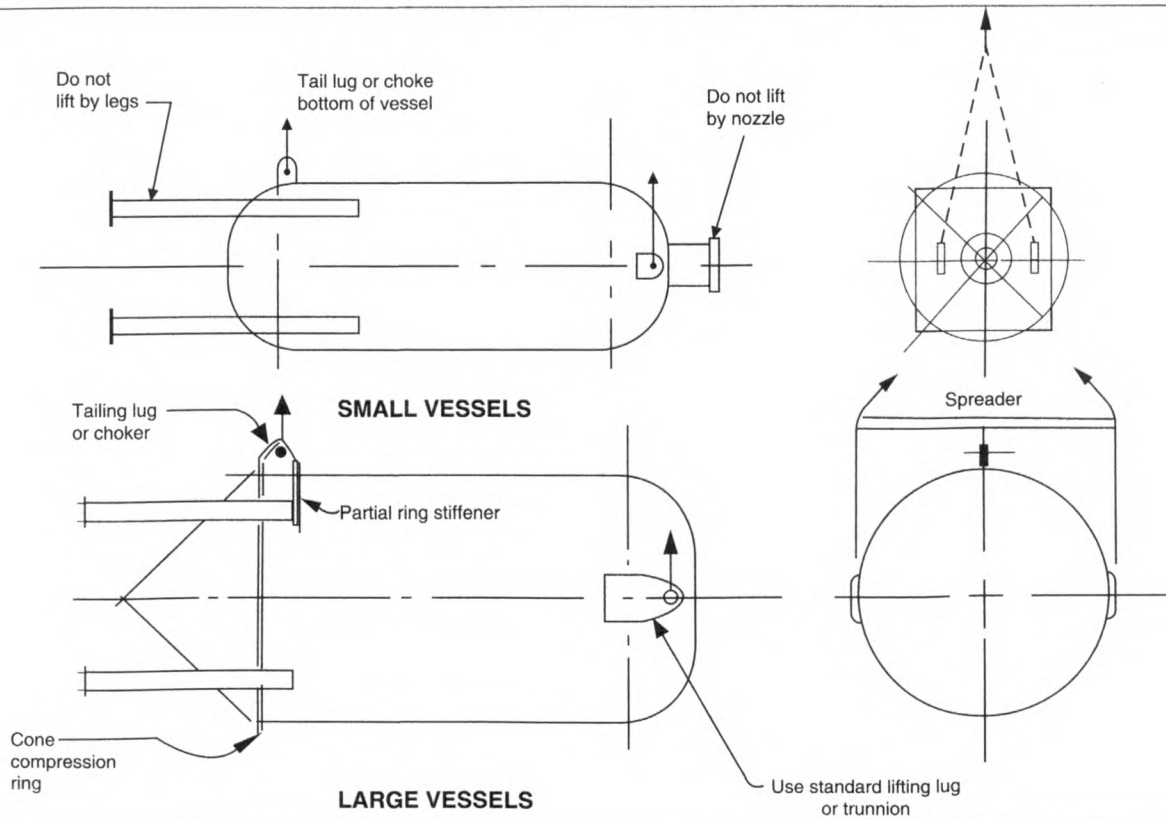
Base with Internal Base Stiffeners and Dual Tailing Lugs



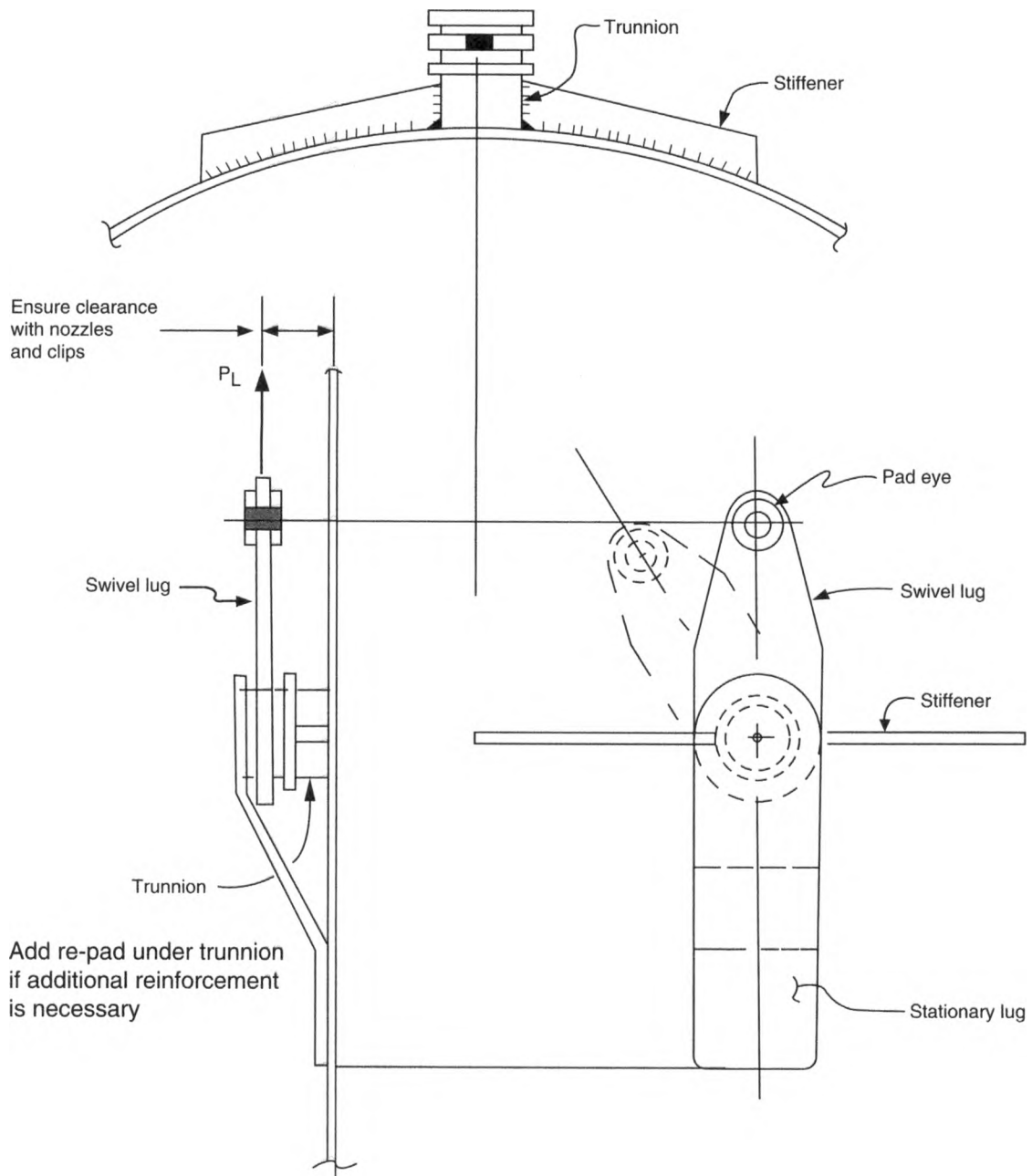
HORIZONTAL VESSELS



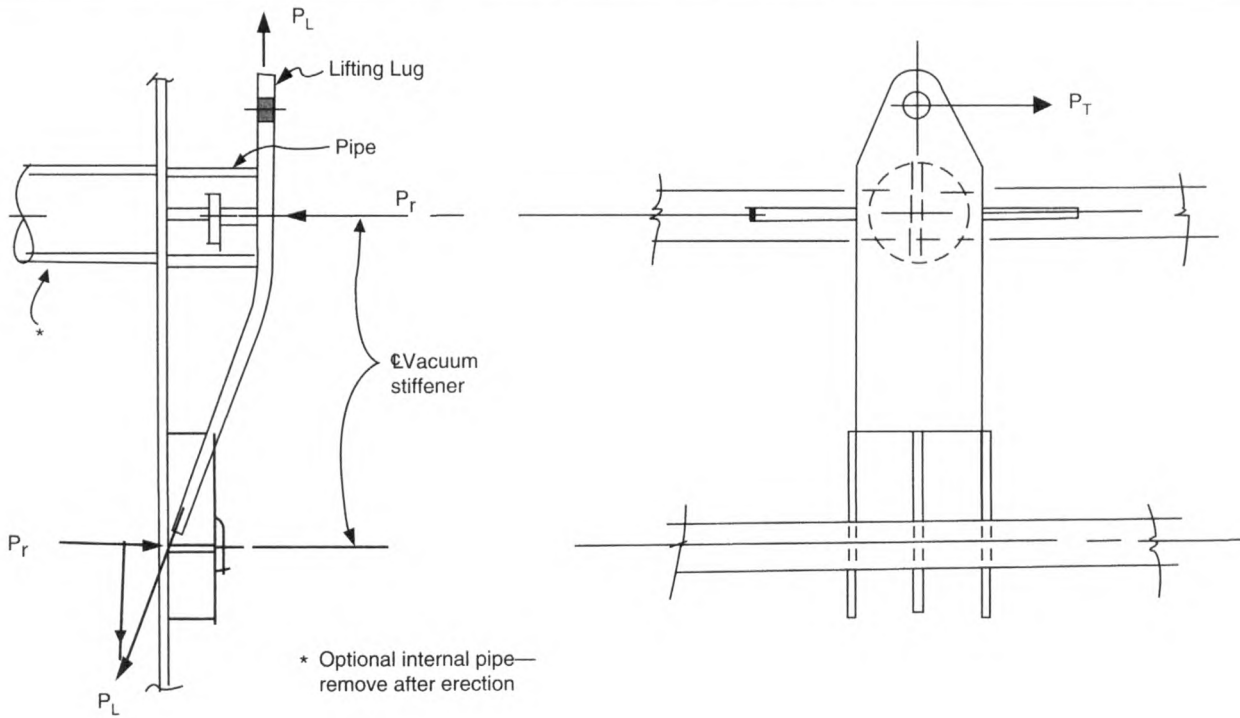
VESSELS, BINS, AND HOPPERS ON LEGS



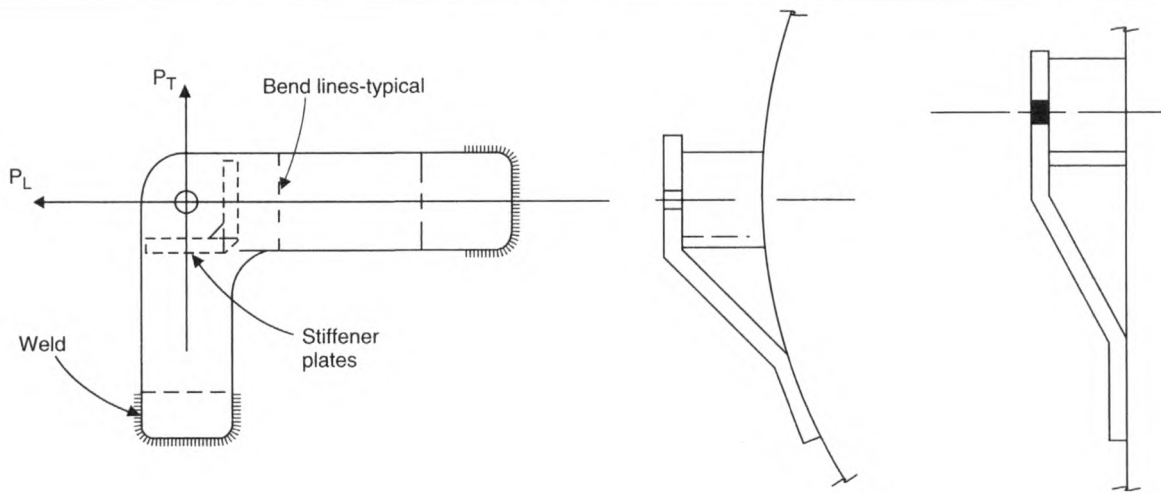
SIDE LUG WITH SWIVEL LUG



SIDE LUG COMBINATION WITH STIFFENERS

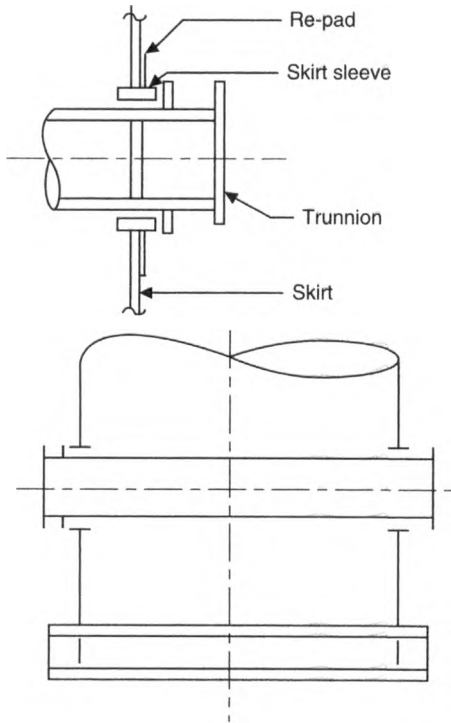


TYPICAL SIDE LUG

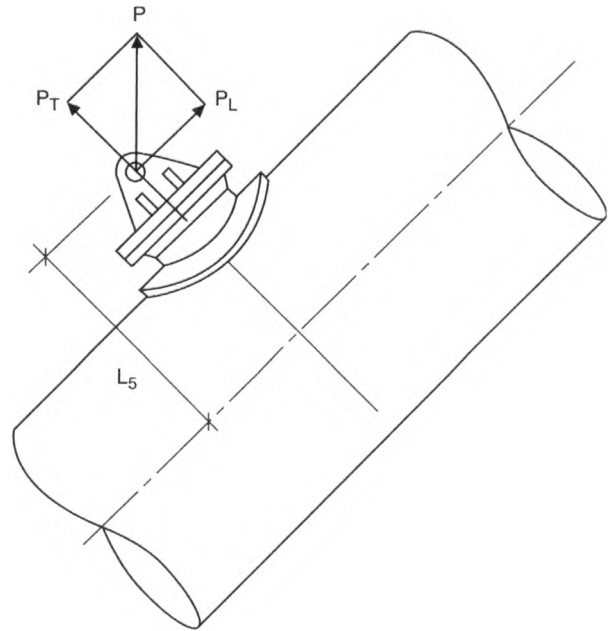


Tailing Trunnion

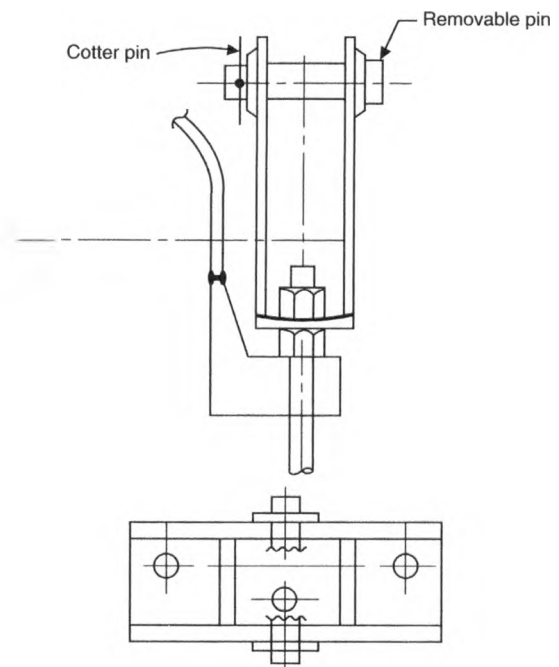
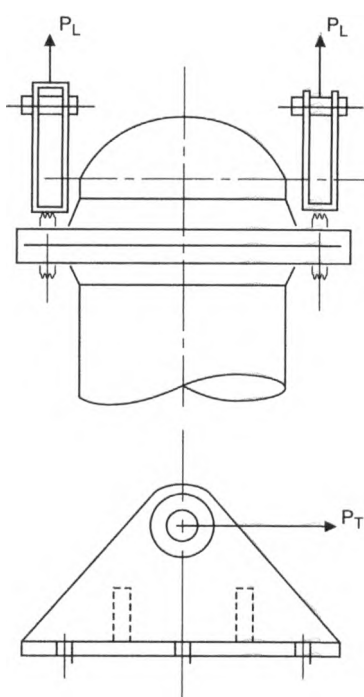
Utilizes reinforced openings in skirt with through pipe. Pipe is removed after erection and the openings used as skirt manways.



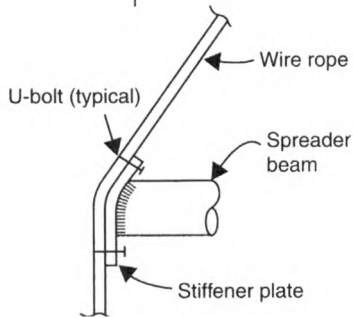
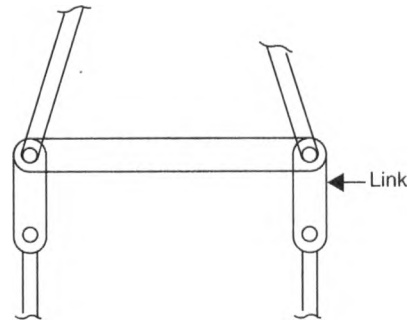
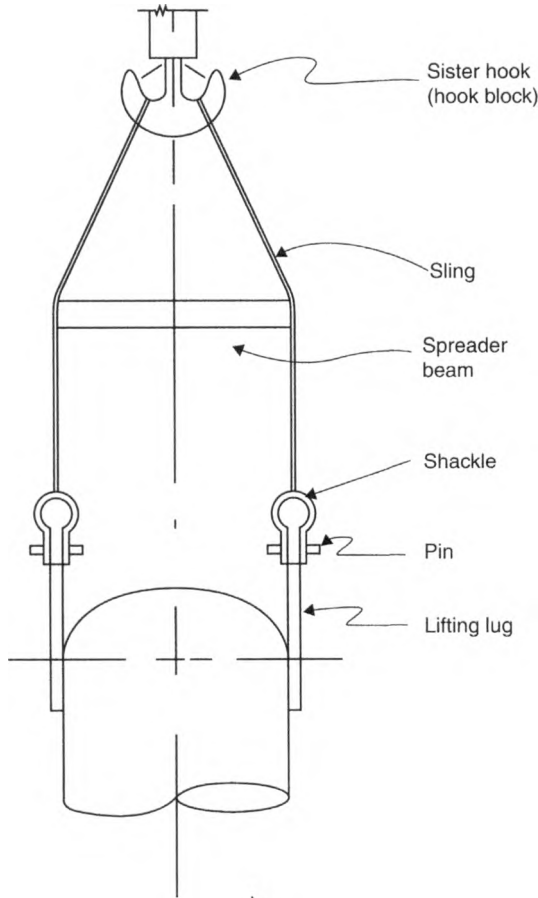
Shell Flange Lug



Lifting Device Utilizing Top Body Flanges



Rigging Terminology



- | | | |
|---------------|----------------------------|-----------------------|
| 1. Boom | 13. Jacks | 25. Guy streamers |
| 2. Mast | 14. Slings | 26. Bail |
| 3. Gin pole | 15. Pins | 27. Tensioning blocks |
| 4. Crane mats | 16. Spreader beams | 28. Hitch plate |
| 5. Dead men | 17. Equalizer beams | 29. Pin extractor |
| 6. Outriggers | 18. Links | 30. Choker |
| 7. Load block | 19. Shackles | 31. Tail crane |
| 8. Whip line | 20. Wire rope | 32. Tail sled |
| 9. Cranes | 21. Counterweight | |
| 10. Derricks | 22. Trailing counterweight | |
| 11. Hoist | 23. Struts | |
| 12. Hooks | 24. Lashings | |

Miscellaneous Lugs, $W_L < 60$ kips

Table 10-6
Lug dimensions

W_L kips	A	D_1	B	C	t_L	w_1	W_L kips	A	D_1	B	C	t_L	w_1
4	3	0.88	1.5	2	0.5	0.25	20	7	1.75	3	3	1	0.38
6	3.5	1	1.63	2	0.63	0.25	25	7	2.38	4	4	1	0.44
8	4	1.13	1.75	2	0.63	0.25	35	8	2.38	4	4	1.125	0.5
10	4.5	1.25	2	2	0.75	0.25	40	8	2.38	4	4	1.125	0.63
12	5	1.38	2.13	3	0.88	0.25	45	8	2.88	4	4	1.25	0.75
14	5.5	1.5	2.38	3	1	0.38	50	10	2.88	4	4	1.25	0.75
16	6.5	1.63	2.5	3	1	0.38	55	10	2.88	4	4	1.25	0.75
18	7	1.75	2.75	3	1	0.38	60	10	2.88	4	4	1.25	0.88

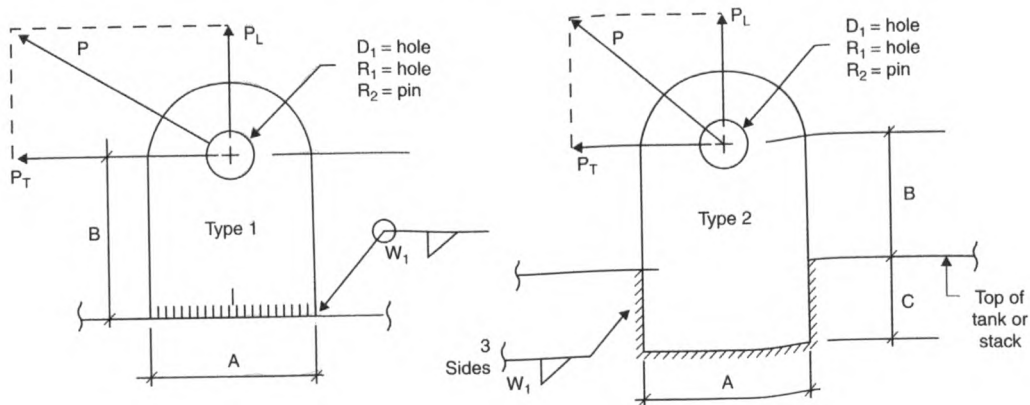


Figure 10-6. Dimensions and forces.

Calculations

Due to bending:

$$t_L = \frac{6P_T B}{A^2 F_b}$$

Due to shear:

$$t_L = \frac{P_T}{(A - D_1) F_s}$$

Due to tension:

$$t_L = \frac{P_L}{(A - D_1) F_t}$$

Notes

1. Table 10-6 is based on an allowable stress of 13.7 ksi.

2. Design each lug for a 2:1 safety factor.
3. Design each lug for a minimum 10% side force.

Hertzian Stress, Bearing

$$\sigma_p = 0.418 \sqrt{\frac{E \left(\frac{P}{t_L} \right) (R_1 - R_2)}{R_1 R_2}} < 2F_y$$

Shear Load in Weld

Type 1: greater of following:

$$\tau_w = \frac{6P_T B}{2A^2}$$

$$\tau_w = \frac{P_L}{2A}$$

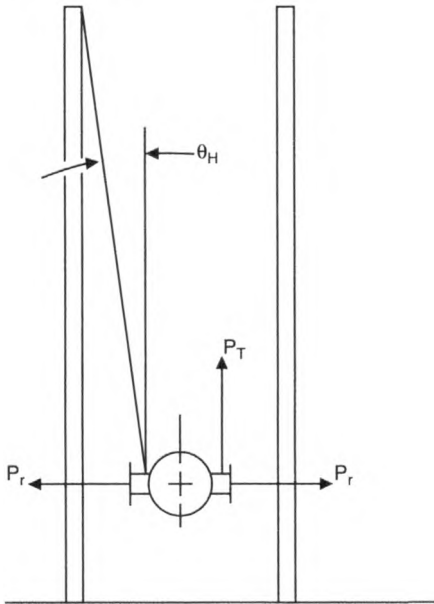
Type 2: Use design for top head lug.

Procedure 10-4: Lifting Loads and Forces

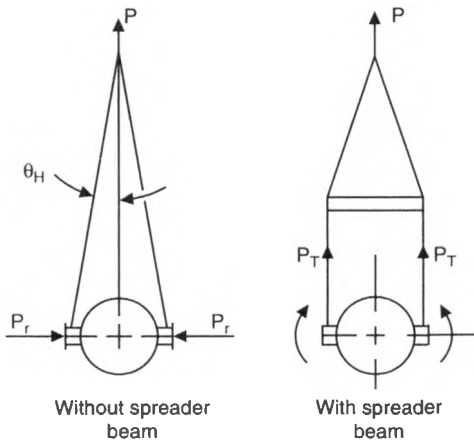
Effect of Lift Line Orientation to Lug

Initial Pick Point

Multipoint Lift

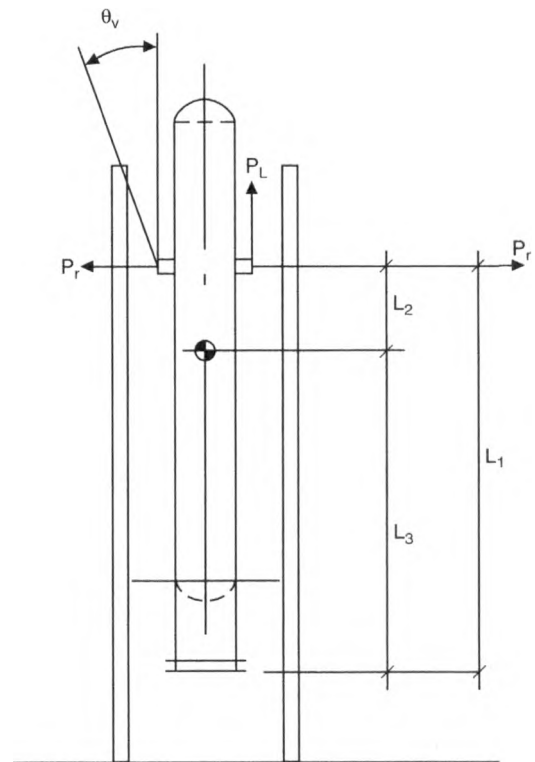


Single-Point Lift

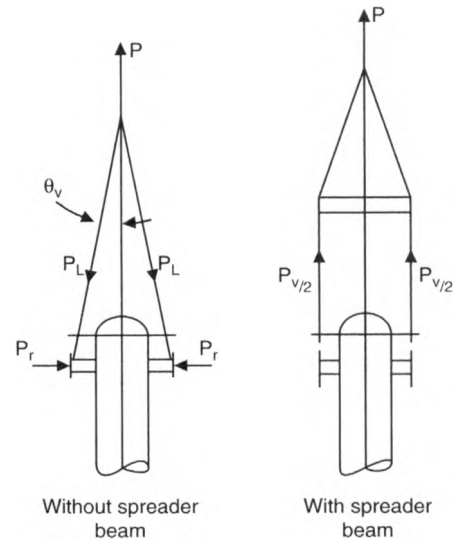


Erected Position

Multipoint Lift

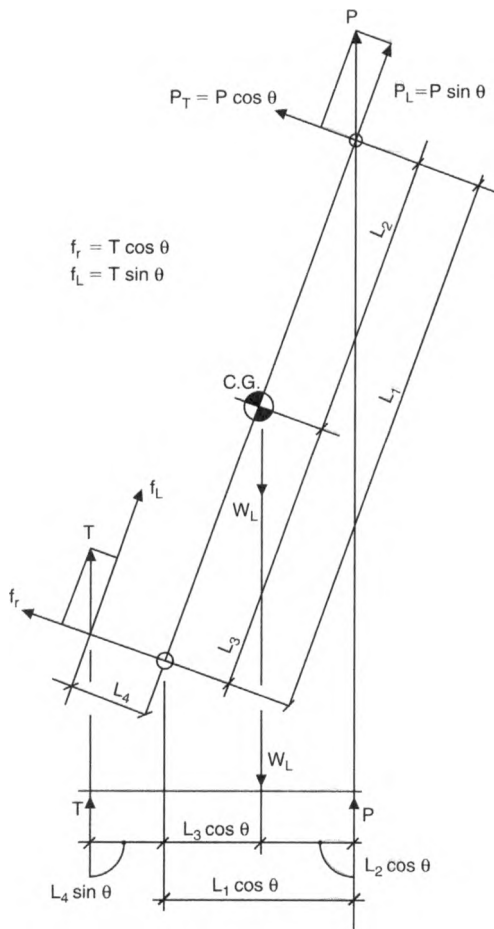


Single-Point Lift

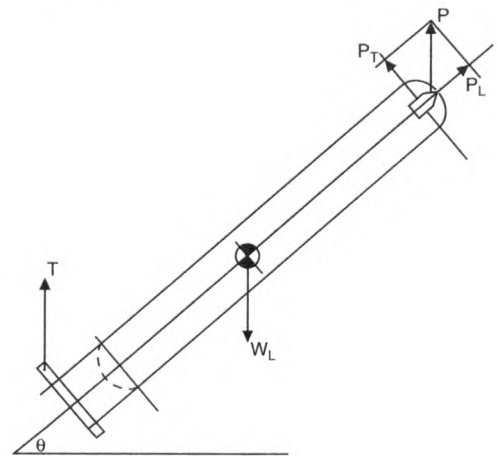


Force and Loading Diagrams

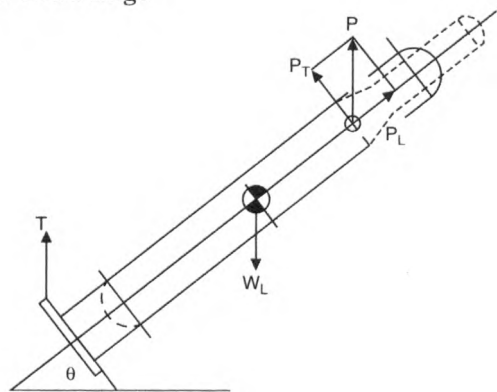
Free-Body Diagram



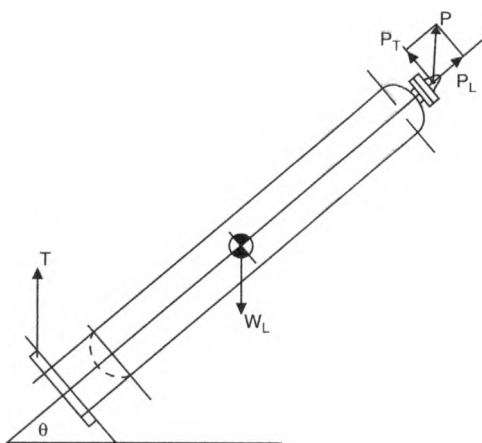
Top Head Lug



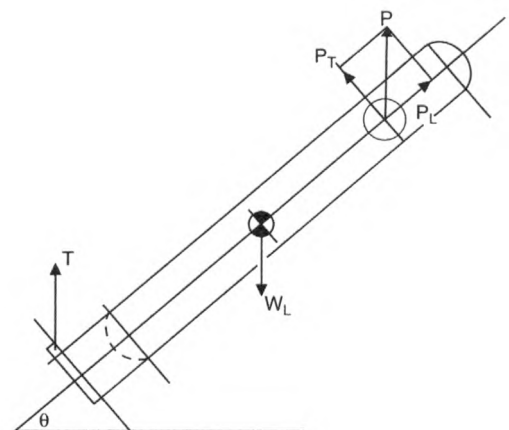
Side or Cone Lugs



Top Flange Lug



Trunnions



Loads

- Overall load factor, K_L

$$K_L = K_i + K_s$$

- Design lift weight, W_L

$$W_L = K_L W_E$$

- Tailing load, T

$$T = \frac{W_L \cos \theta L_2}{\cos \theta L_1 + \sin \theta L_4}$$

At $\theta = 0$, initial pick point, vessel horizontal:

$$T = \frac{W_L L_2}{L_1} \quad \text{and} \quad P = \frac{W_L L_3}{L_1} \quad \text{or} \quad P = W_L - T$$

At $\theta = 90^\circ$, vessel vertical:

$$T = 0 \quad \text{and} \quad P = W_L$$

- Calculate the loads for various lift angles, θ .

Loads T and P		
θ	T	P
0		
10		
20		
30		
40		
50		
60		
70		
80		
90		

Lift angles shown are suggested only to help find the worst case for loads T and P.

- Maximum transverse load per lug, P_T

$$P_T = \frac{P \cos \theta}{n_L}$$

- Maximum longitudinal load per lug, P_L

$$P_L = \frac{P \sin \theta}{n_L}$$

- Radial loads in shell due to sling angles, θ_v or θ_H

$$P_r = P_T \tan \theta_H \quad \text{Vessel in horizontal}$$

$$P_r = P_L \tan \theta_v \quad \text{Vessel in vertical}$$

- Tailing loads, f_L and f_r

$$f_L = T \cos \theta$$

$$f_r = T \sin \theta$$

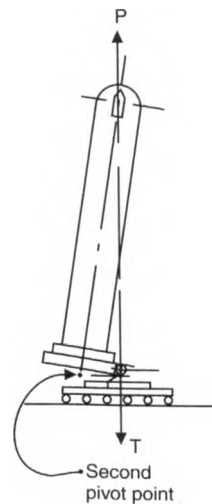
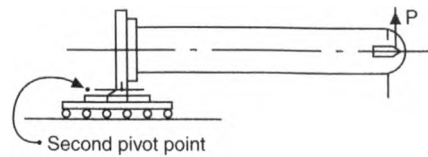
- Longitudinal bending stress in vessel shell, σ_b

$$\sigma_b = \frac{4M}{\pi D_m^2 t}$$

Maximum moment occurs at initial pick, when $\theta = 0$. See cases 1 through 4 for maximum moment, M.

Note

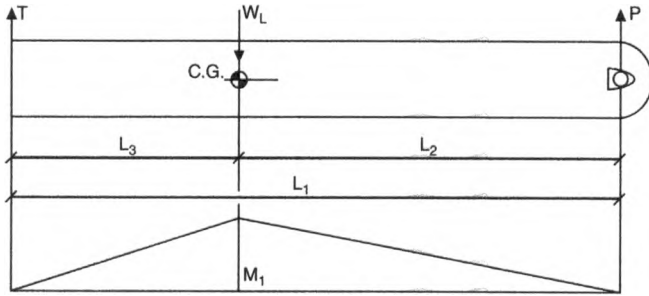
If the tailing point is below the CG as is the case when a tailing frame or sled is used, the tail support could see the entire weight of the vessel as erection approaches 90° .



Critical point:
When the pick point is immediately above the tail pivot point

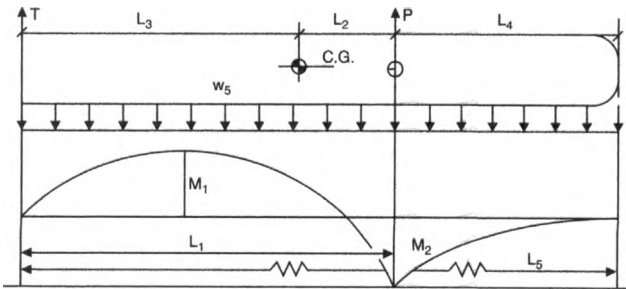
Dimensions and Moments for Various Vessel Configurations

Case 1: Top Head Lug, Top Head Trunnion, or Top Head Flange



$$M_1 = \frac{W_L L_3 L_2}{L_1}$$

Case 2: Side Lug or Side Trunnion

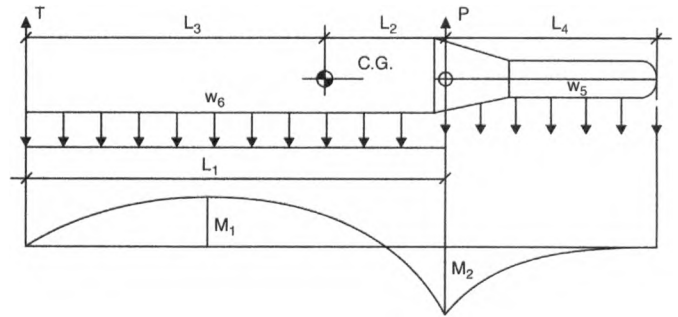


$$w_5 = \frac{W_L}{L_5}$$

$$M_1 = \frac{w_5}{8L_1^2} (L_1 + L_4)^2 (L_1 - L_4)^2$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

Case 3: Cone Lug or Trunnion

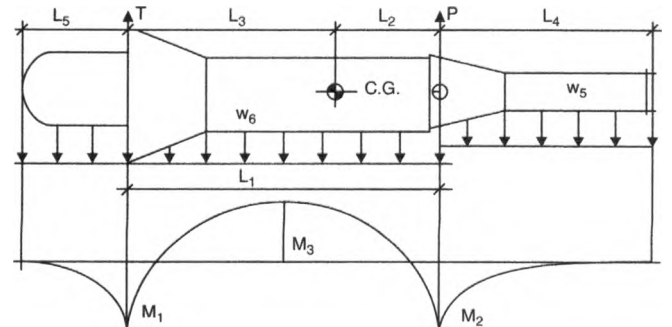


$$w_5 = \frac{W_{L1}}{L_4} \quad w_6 = \frac{W_{L2}}{L_1}$$

$$M_1 = \frac{w_6}{8L_1^2} (L_1 + L_4)^2 (L_1 - L_4)^2$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

Case 4: Cone Lug or Trunnion with Intermediate Skirt Tail



$$w_5 = \frac{W_{L1}}{L_4} \quad w_6 = \frac{W_{L2}}{L_1 + L_5}$$

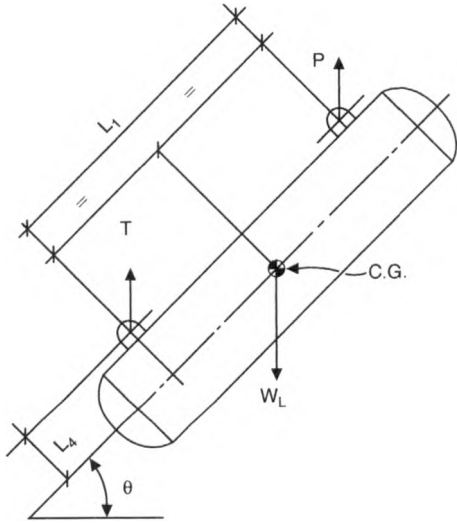
$$M_1 = \frac{w_6 L_5^2}{2}$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

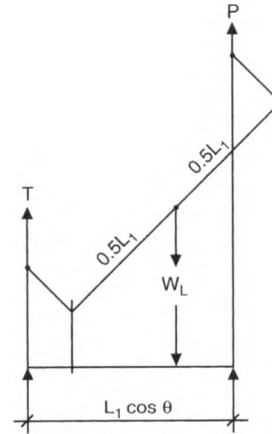
$$M_3 = \left(\frac{M_1 + M_2}{2} \right) - \frac{w_6 L_1^2}{8}$$

Find Lifting Loads at Any Lift Angle for a Symmetrical Horizontal Drum

Dimensions and Forces



Free-Body Diagram



Curve is based on the following equation:

$$\frac{P}{W_L} = \frac{L_4}{L_1} (\tan \theta) + 0.5$$

Example

Steam drum:

$W_L = 600$ kips

$L_1 = 80$ ft

$L_4 = 5$ ft

$\frac{L_1}{2L_4} = \frac{80}{10} = 8$

Results from curve

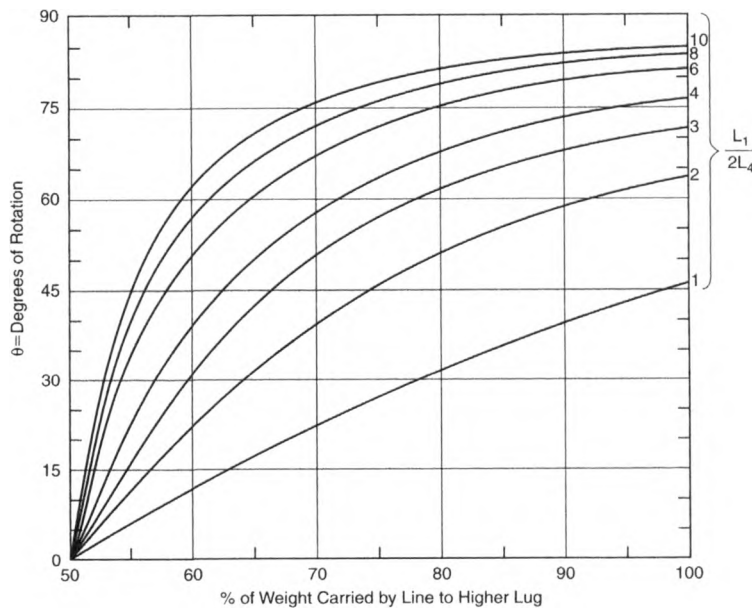
@ $\theta = 15^\circ = 51.6\%$

@ $\theta = 30^\circ = 53.6\%$

@ $\theta = 45^\circ = 56.3\%$

@ $\theta = 60^\circ = 60.8\%$

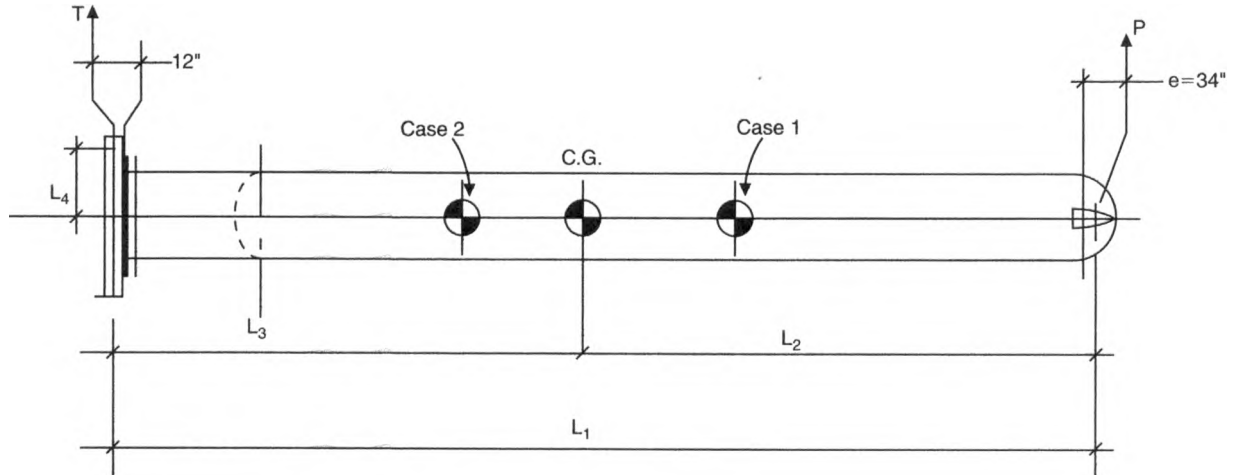
@ $\theta = 75^\circ = 73.3\%$



Reprinted by permission of The Babcock and Wilcox Company, a McDermott Company.

Sample Problem

Distillation column:
 18 ft in diameter x 280 ft OAL
 260 ft tangent-to-tangent
 $W_L = 200$ tons (400 kips)



Case 1: $L_3 > L_2$

$L_1 = 280 + 2.833 + 1 = 283.83$ ft
 $L_2 = 283.83 - 162 = 121.83$ ft
 $L_3 = 161 + 1 = 162$ ft
 $L_4 = 10$ ft

Case 2: $L_3 < L_2$

$L_1 = 283.83$ ft
 $L_2 = 162$ ft
 $L_3 = 121.83$ ft
 $L_4 = 10$ ft

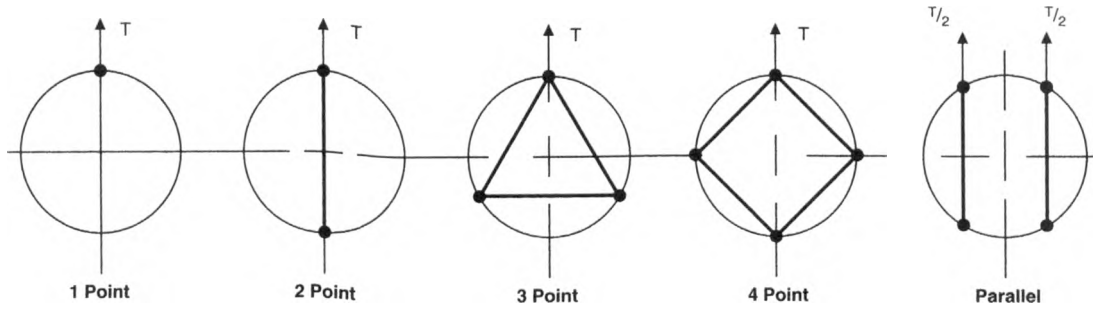
Loads T and P		
θ	T	P
0	171.7	228.3
10	170.6	229.4
20	169.6	230.4
30	168.3	231.7
40	166.8	233.2
50	164.8	235.2
60	161.9	238.1
70	156.6	243.4
80	143.2	256.8
90	0	400

Loads T and P		
θ	T	P
0	228.3	171.7
10	226.9	173.1
20	225.4	174.6
30	223.7	176.3
40	221.7	178.3
50	219.1	180.9
60	215.1	184.9
70	208.1	191.9
80	190.1	209.8
90	0	400

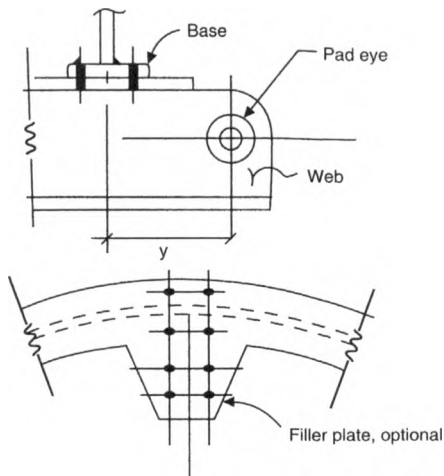
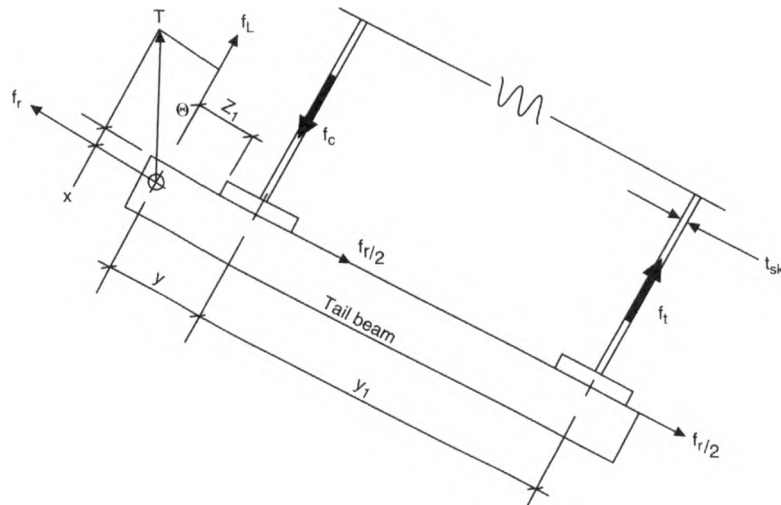
Procedure 10-5: Design of Tail Beams, Lugs, and Base Ring Details

Design of Base Plate, Skirt, and Tail Beam

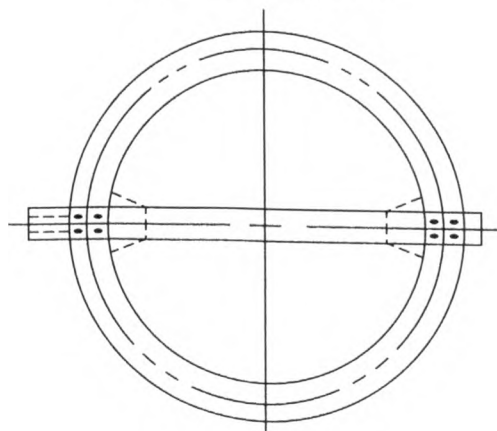
Base Ring-Stiffening Configuration



Loadings in Skirt and Base

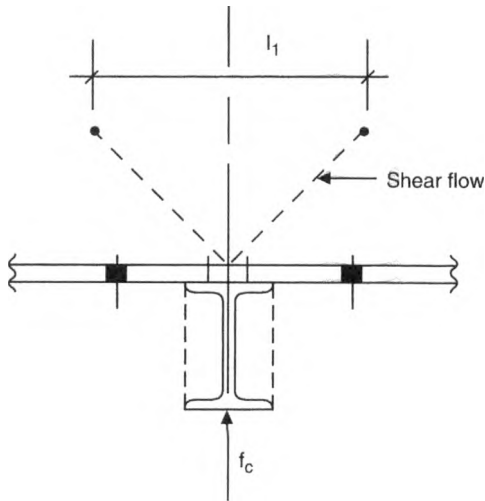


Tail Beam Connection Details



Skirt Crippling Criteria with Tailing Beam

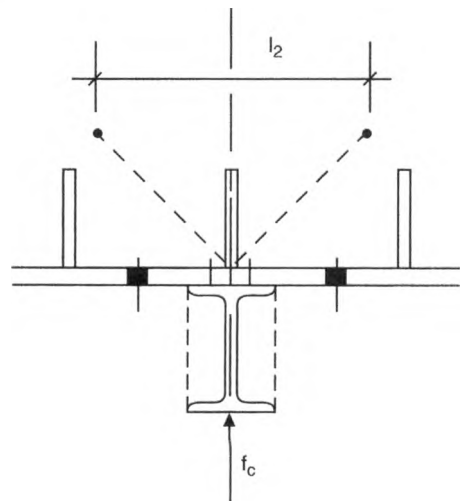
Base Type 1: Base Ring Only



$$l_1 = N + 2l_r$$

$$l_r = 16t_{sk}$$

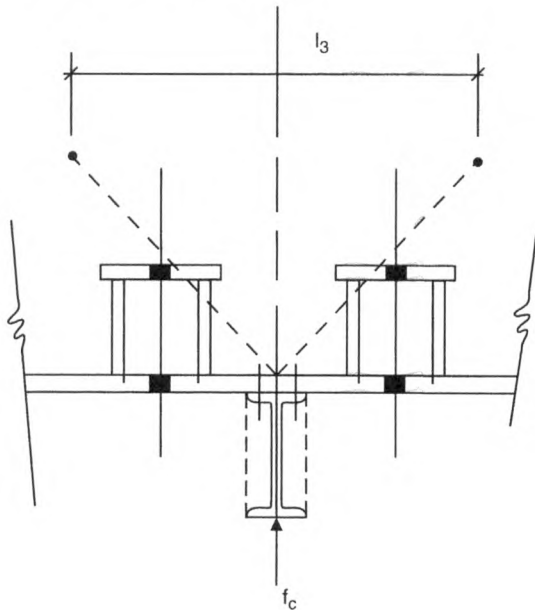
Base Type 2: Gussets Only



$$l_2 = N + 2l_r$$

$$l_r = 16t_{sk}$$

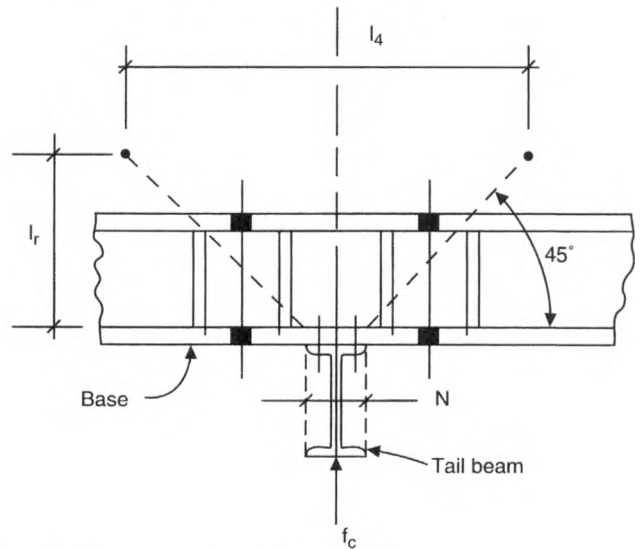
Base Type 3: w/Anchor Chairs



$$l_3 = N + 2l_r$$

$$l_r = 0.55\sqrt{D_{sk}t_{sk}}$$

Base Type 4: w/Continuous Top Ring

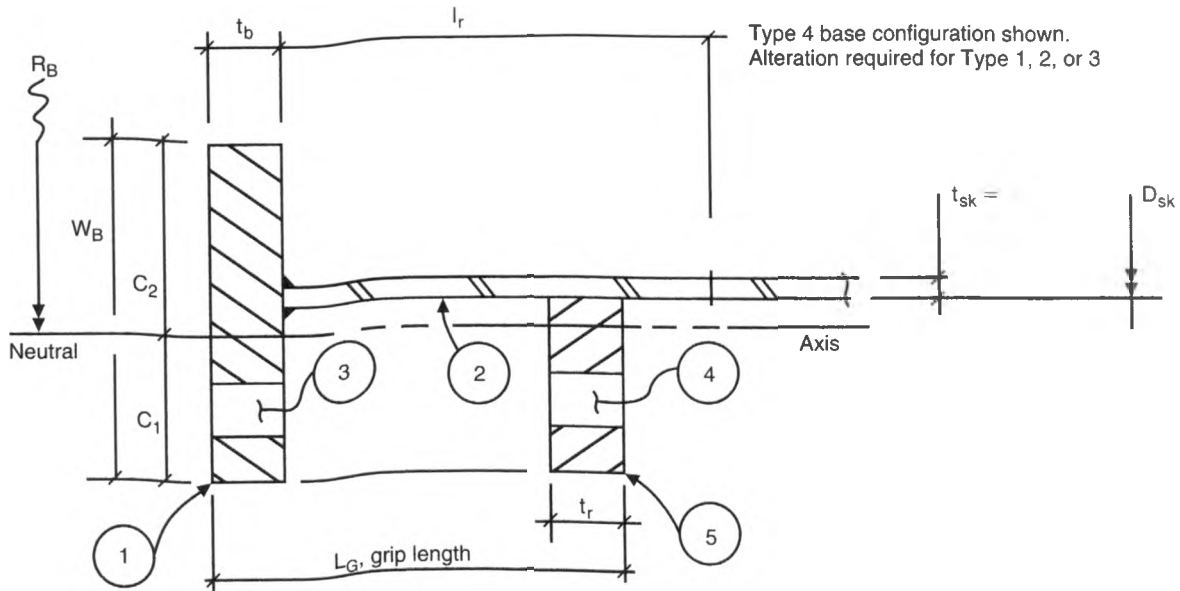


Note: $N = 1$ in. if a web stiffener is not used.

$$l_4 = N + 2l_r$$

$$l_r = (L_C - t_b) + 0.55\sqrt{D_{sk}t_{sk}}$$

Base Ring Design Check



Item	A	Y	Y ²	AY	AY ²	I _o
1						
2						
3(-)						
4(-)						
5						
Σ						

$$C_1 = \frac{\Sigma AY}{\Sigma A}$$

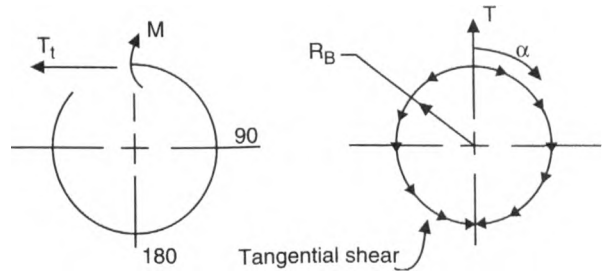
$$C_2 = W_B - C_1$$

$$I = \Sigma AY^2 + \Sigma I_o - C_1 \Sigma AY$$

$$R_B = \text{inside radius of base plate} + C_2$$

Internal Forces and Moments in the Skirt Base During Lifting

To determine the stresses in the base ring as a result of the tailing load, the designer must find the coefficients K_r and K_T based on angle α as shown and the type of stiffening in the skirt/base ring configuration.



$$M = K_r TR_B$$

$$T_t = K_T T$$

Skirt/Tail Beam Calculations

Tail Beam

- Tailing loads, f_L and f_r .

$$f_L = T \cos \theta$$

$$f_r = T \sin \theta$$

- Maximum bending moment, M_b .

$$M_b = x f_r + y f_L$$

- Maximum bending stress, σ_b .

$$\sigma_b = \frac{M_b}{Z}$$

Tail Beam Bolts

- Shear load, f_s .

$$f_s = \frac{0.5 f_r}{n}$$

- Shear stress, τ .

$$\tau = \frac{f_s}{A_b}$$

- Tension force, f_t .

Note: y_1 = mean skirt diameter or centerline of bolt group if a filler plate is used.

$$f_t = \frac{M_b}{y_1}$$

Skirt

- Tension stress in bolts, σ_T .

$$\sigma_T = \frac{f_T}{N_b A_b}$$

- Compressive force in skirt, f_c .

$$f_c = f_L + f_t$$

- Skirt crippling is dependent on the base configuration and lengths l_1 through l_4 .

$N = 1$ in. if web stiffeners are not used

N = width of top flange of tail beam if web stiffeners are used

- Compressive stress in skirt, σ_c .

$$\sigma_c = \frac{f_c}{t_{sk} l_n}$$

- Check shear stress, τ , in base to skirt weld.

$$\tau = \frac{f_r}{\pi D_{sk} \cdot 0.707 w_4}$$

Base Plate

- Bending moment in base plate, M_b .

$$M_b = K_r T R_B$$

- Find tangential force, T_t .

$$T_t = K_T T$$

- Total combined stress, σ .

$$\sigma_T = \frac{M_b C_1}{I} + \frac{T_t}{A} \quad (\text{tension})$$

$$\sigma_C = \frac{-M_b C_2}{I} - \frac{T_t}{A} \quad (\text{compression})$$

Size Base Ring Stiffeners

F_1 = force in strut or tailing beam, lb
 F_1 is (+) for tension and (-) for compression

- Tension stress, σ_T .

$$\sigma_T = \frac{F_n}{A_s}$$

- Critical buckling stress per AISC, σ_{cr} .

$$C_c = \sqrt{\frac{2\pi^2}{F_y}}$$

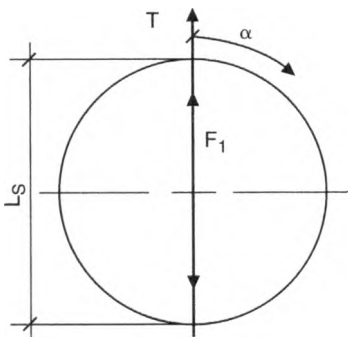
$$\sigma_{cr} = \frac{[(1 - (KL_s^2/r)/2C_c^2)]F_y}{(5/3) + ((3KL_s/r)/8C_c) - ((KL_s/r)^3/8C_c^3)}$$

- Actual compressive stress, σ_c .

$$\sigma_c = \frac{F_n}{A_s}$$

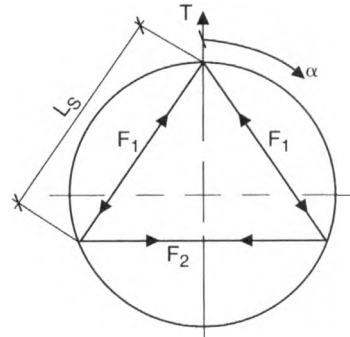
Note: Evaluate all struts as tension and compression members regardless of sign, because when the vessel is sitting on the ground, the loads are the reverse of the signs shown.

Two Point



$$F_1 = (+)0.5T$$

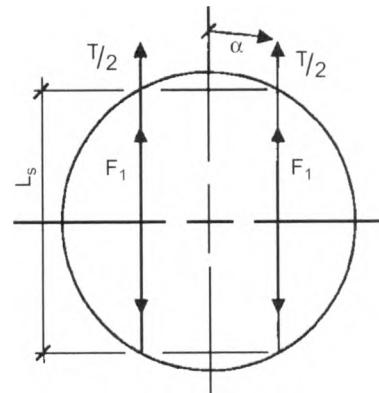
Three Point



$$F_1 = (+)0.453T$$

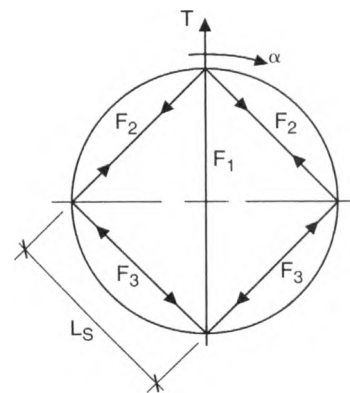
$$F_2 = (-)0.329T$$

Parallel Beams/Struts



$$F_1 = (+)0.25T$$

Four Point



$$F_1 = (+)0.5T$$

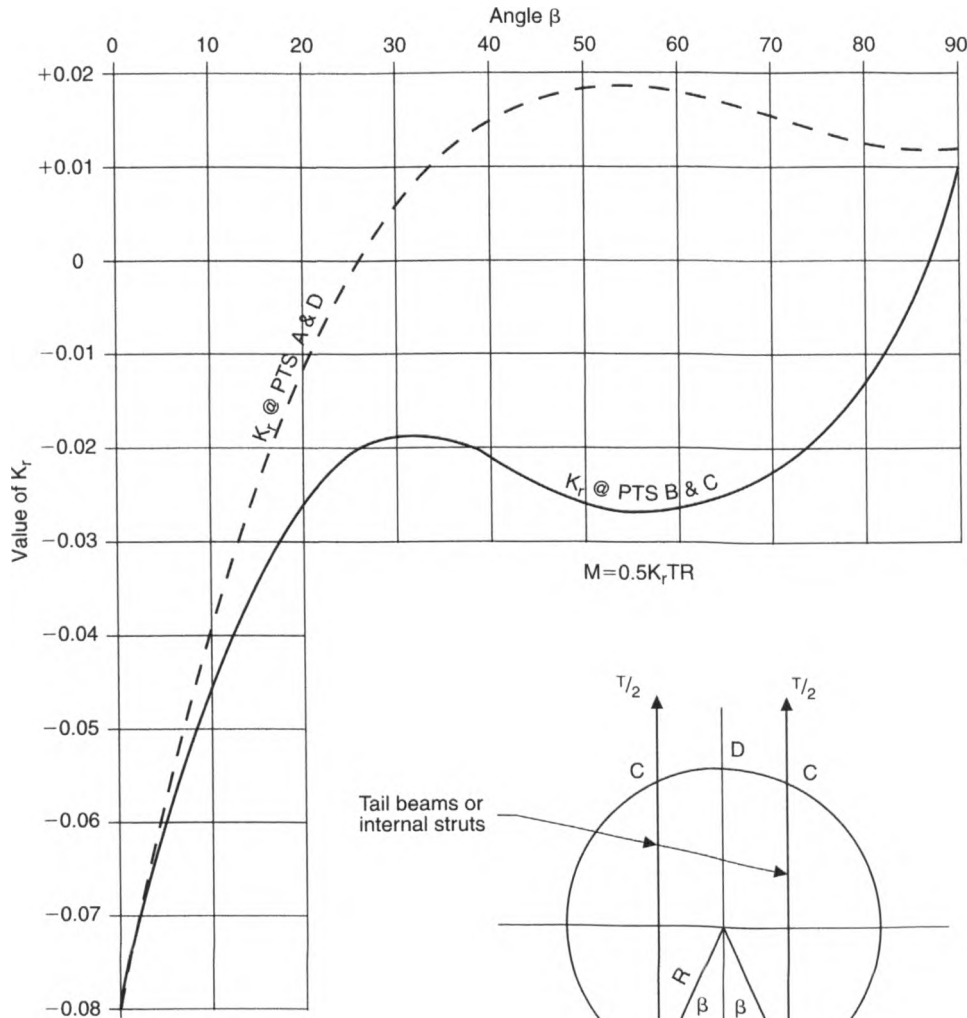
$$F_2 = (-)0.273T$$

$$F_3 = (+)0.273T$$

Table 10-7
Internal moment coefficients for base ring

Angle α	One Point		Two Point		Three Point		Four Point	
	K_r	K_T	K_r	K_T	K_r	K_T	K_r	K_T
0	0.2387	-0.2387	0.0795	-0.2387	-0.0229	0.1651	0.0093	-0.1156
5	0.1961	-0.2802	0.0587	-0.2584	-0.0148	0.1708	0.0048	-0.1188
10	0.1555	-0.3171	0.0398	-0.2736	-0.0067	0.1764	0.0012	-0.1188
15	0.1174	-0.3492	0.0229	-0.2845	-0.0055	0.1747	-0.0015	-0.1155
20	0.0819	-0.3763	0.0043	-0.2908	-0.0042	0.1729	-0.0033	-0.1089
25	0.0493	-0.3983	-0.0042	-0.2926	0.0028	0.1640	-0.0043	-0.0993
30	0.0197	-0.4151	-0.0145	-0.2900	0.0098	0.1551	-0.0045	-0.0867
35	-0.0067	-0.4266	-0.0225	-0.2831	0.0103	0.1397	-0.0041	-0.0713
40	-0.0299	-0.4328	-0.0284	-0.2721	0.0107	0.1242	-0.0031	-0.0534
45	-0.0497	-0.4340	-0.0321	-0.2571	0.0093	0.1032	-0.0017	-0.0333
50	-0.0663	-0.4301	-0.0335	-0.2385	0.0078	0.0821	-0.0001	-0.0112
55	-0.0796	-0.4214	-0.0340	-0.2165	0.0052	0.0567	0.0017	0.0126
60	-0.0897	-0.4080	-0.0324	-0.1915	0.0025	0.0313	0.0033	0.0376
65	-0.0967	-0.3904	-0.0293	-0.1638	0.0031	0.0031	0.0046	0.0636
70	-0.1008	-0.3688	-0.0250	-0.1338	0.0037	-0.0252	0.0055	0.0901
75	-0.1020	-0.3435	-0.0197	-0.1020	-0.0028	-0.0548	0.0056	0.1167
80	-0.1006	-0.3150	-0.0136	-0.0688	-0.0092	-0.0843	0.0049	0.1431
85	-0.0968	-0.2837	-0.0069	-0.0346	-0.0107	-0.1134	0.0031	0.1688
90	-0.0908	-0.2500	0	0	-0.0121	-0.1425	0	0.1935
95	-0.0830	-0.2144	0.0069	0.0416	-0.0114	-0.1694	-0.0031	-0.1688
100	-0.0735	-0.1774	0.0135	0.0688	-0.0107	-0.1963	-0.0049	-0.1431
105	-0.0627	-0.1394	0.0198	0.1020	-0.0074	-0.2194	-0.0057	-0.1167
110	-0.0508	-0.1011	0.0250	0.1338	-0.0033	-0.2425	-0.0055	-0.0901
115	-0.0381	-0.0627	0.0293	0.1638	0.0041	-0.2603	-0.0046	-0.0636
120	-0.0250	-0.0250	0.0324	0.1915	0.0114	-0.2781	-0.0033	-0.0376
125	-0.0016	0.0118	0.0340	0.2165	0.0107	-0.1060	-0.0017	-0.0126
130	0.0116	0.0471	0.0335	0.2385	0.0100	0.0661	0.0001	0.0112
135	0.0145	0.0804	0.0321	0.2571	0.0083	0.0448	0.0017	0.0333
140	0.0268	0.1115	0.0284	0.2721	0.0066	0.0234	0.0031	0.0534
145	0.0382	0.1398	0.0225	0.2831	0.0045	0.0104	0.0041	0.0713
150	0.0486	0.1551	0.0145	0.2900	0.0024	-0.0026	0.0045	0.0867
155	0.0577	0.1870	0.0042	0.2926	-0.0005	-0.0213	0.0043	0.0993
160	0.0654	0.2053	-0.0083	0.2908	-0.0015	-0.0399	0.0033	0.1089
165	0.0715	0.2198	-0.0225	0.2845	-0.0028	-0.0484	0.0015	0.1155
170	0.0760	0.2301	-0.0398	0.2736	-0.0041	-0.0569	-0.0012	0.1188
175	0.0787	0.2366	-0.0587	0.2584	-0.0046	-0.0597	-0.0048	0.1188
180	0.0796	0.2387	-0.0795	0.2387	-0.0051	-0.0626	-0.0093	0.1156

Values of Moment Coefficient, K_r , for Base Ring With Two Parallel Tail Beams or Internal Struts



Notes:

1. Based on R. J. Roark, *Formulas for Stress and Strain*, 3rd Edition, Case 25.
2. The curve shows moment coefficients at points C and D. The moment coefficients at point A and B are equal and opposite.
3. Positive moments put the inside of the vessel in circumferential tension.

4. The signs of coefficients are for hanging loads. For point support loads underneath the vessel, the signs of the coefficients should be reversed.

Design of Vessel for Choker (Cinch) Lift at Base

- Uniform load, p .

$$p = \frac{T}{R}$$

- Moments in ring at points A and C.

$$M_A = -0.1271TR$$

$$M_C = -0.0723TR$$

- Tension/compression forces in ring at points A and C.

$$T_A = -0.6421T$$

$$T_C = -1.2232T$$

- Combined stress at point A, inside of ring.

$$\sigma_A = \frac{T_A}{A} + \frac{M_A}{Z_{in}}$$

- Combined stress at point A, outside of ring.

$$\sigma_A = \frac{T_A}{A} - \frac{M_A}{Z_{out}}$$

- Combined stress at point C, inside of ring.

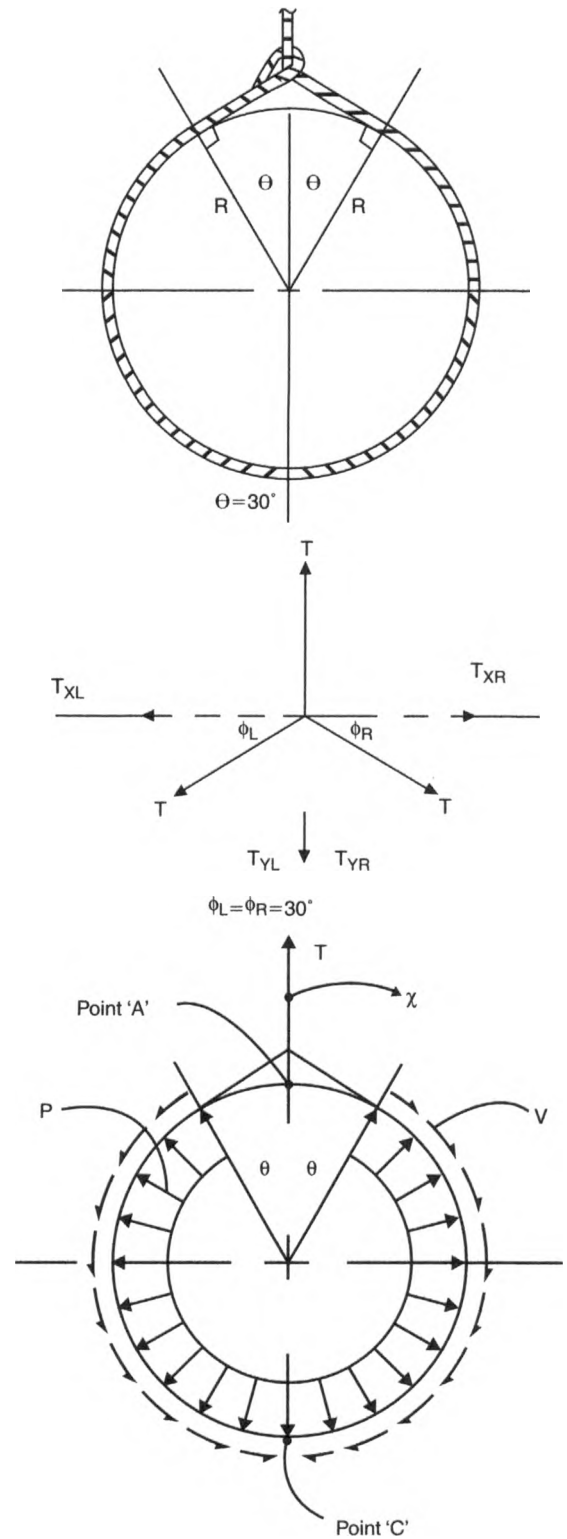
$$\sigma_C = \frac{T_C}{A} + \frac{M_C}{Z_{in}}$$

- Combined stress at point C, outside of ring.

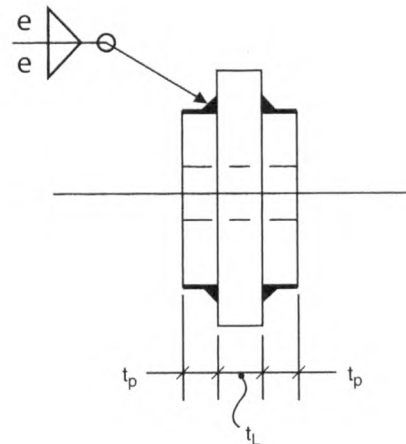
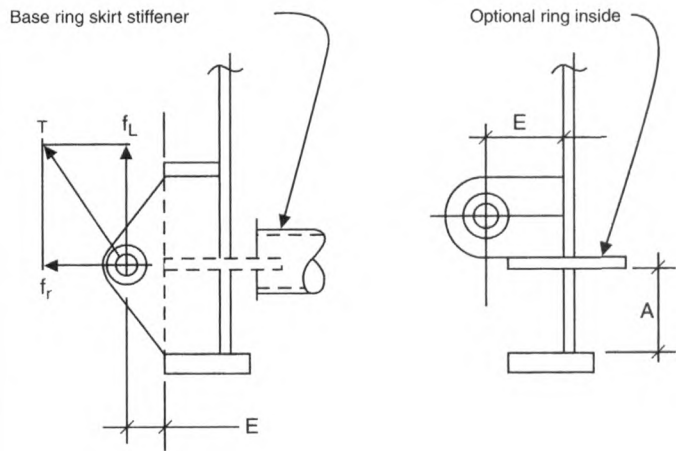
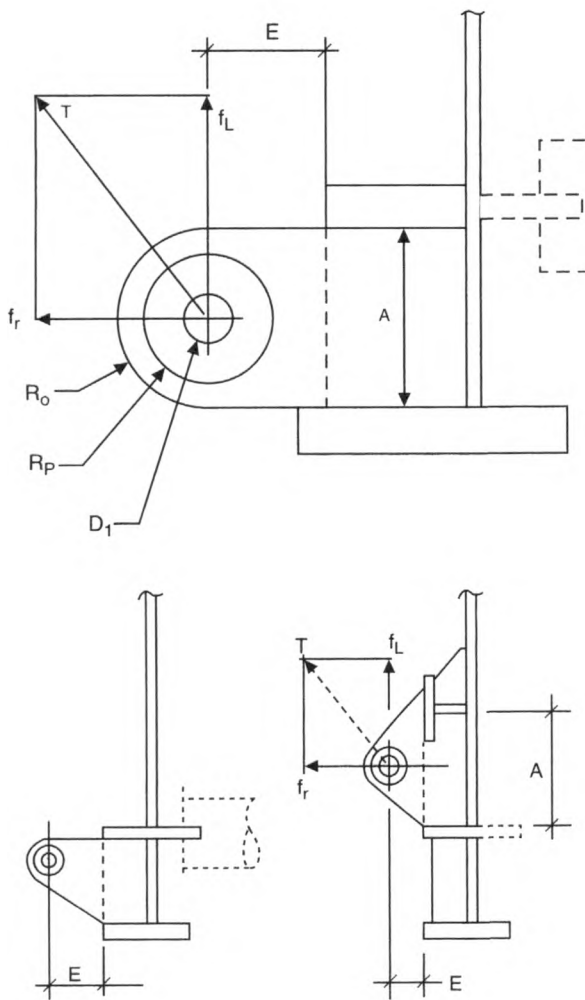
$$\sigma_C = \frac{T_C}{A} - \frac{M_C}{Z_{out}}$$

Note: Assume that the choker is attached immediately at the base ring even though this may be impossible to achieve. Then use the properties of the base ring for A and Z.

From R. J. Roark, *Formulas for Stress and Strain*, 5th Edition, McGraw-Hill Book Co., Table 17, Cases 12 and 18 combined.



Design of Tailing Lugs



**Table 10-8
Dimensions for tailing lugs**

Tail Load (kips)	t_L	t_p	E	R_o	R_p	D_1	e
<10				None required			
10 to 20	0.75	NR	3	4	NR	2.375	NR
21 to 40	0.75	0.375	3	4	3.5	2.375	0.3125
41 to 70	1	0.5	3	4	3.5	2.375	0.3125
71 to 100	1	0.5	4	5	4.5	3.4375	0.3125
101 to 130	1.5	0.5	4	5	4.5	3.4375	0.3125
131 to 170	1.625	0.75	4	5	4.5	3.4375	0.375
171 to 210	1.625	0.75	5	6	5.5	4.5	0.375
211 to 250	2	0.75	5	6	5.5	4.5	0.4375
251 to 300	2.25	1	5	6	5.5	4.5	0.5
>300				Special design required			

Formulas

The tailing lug is designed like all other lugs. The forces are determined from the tailing load, T , calculated per this procedure. The ideal position for the tailing lug is to be as close as possible to the base plate for stiffness and transmitting these loads through the base to the skirt. The option of using a tailing lug versus a tailing beam is the designer's choice. Either can accommodate internal skirt rings, stiffeners, and struts.

Design as follows:

- Area required at pin hole, A_r

$$A_r = \frac{T}{F_s}$$

- Area available at pin hole, A_a

$$A_a = (At_L) - (D_1 t_L)$$

- Bending moment in lug, M_b

$$M_b = f_L E$$

- Section modulus of lug, Z

$$Z = \frac{t_L A^2}{6}$$

- Bending stress in lug, σ_b

$$\sigma_b = \frac{M_b}{Z}$$

- Area required at pin hole for bearing, A_r

$$A_r = \frac{T}{F_p}$$

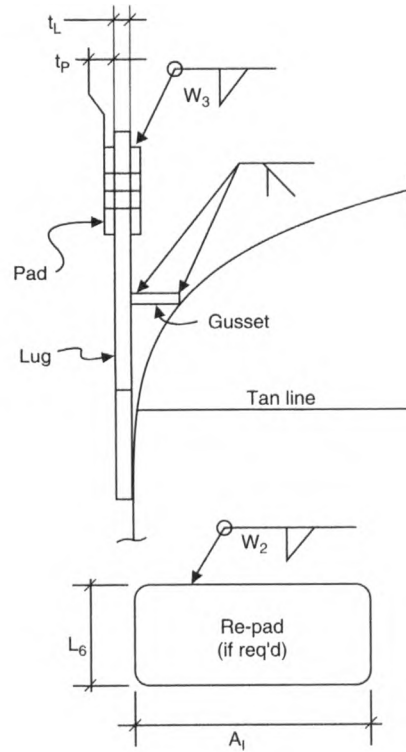
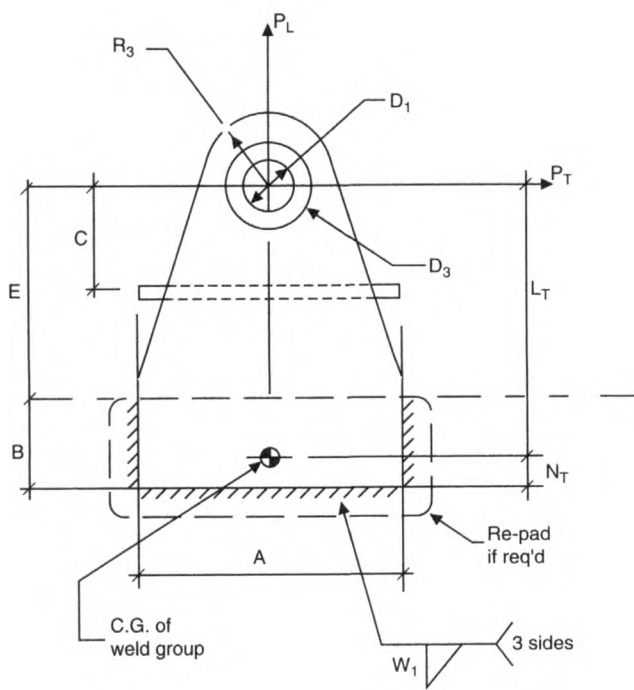
- Area available at pin hole for bearing, A_a

$$A_a = D_2 t_L$$

Note: Substitute $t_L + 2t_p$ for t_L in the preceding equations if pad eyes are used.

Procedure 10-6: Design of Top Head and Cone Lifting Lugs

Design of Top Head/Cone Lug



Dimensions

$$N_T = \frac{B^2}{A + 2B}$$

$$L_T = E + B - N_T$$

$$\theta_1 = \arctan \frac{2L_1}{A}$$

$$L_2 = \frac{L_1}{\sin \theta_1}$$

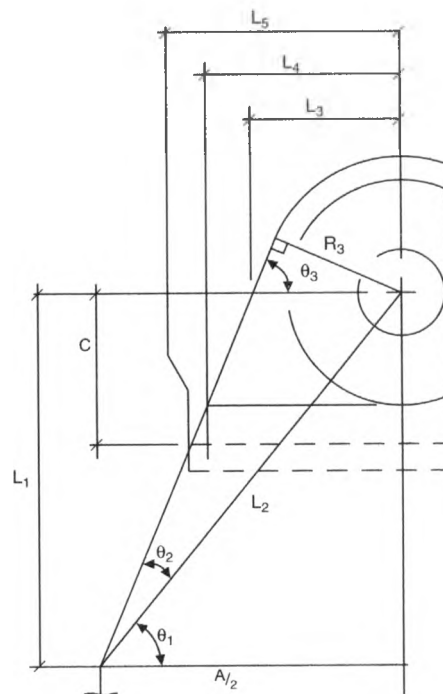
$$\theta_2 = \arcsin \frac{R_3}{L_2}$$

$$\theta_3 = \theta_1 + \theta_2$$

$$L_3 = \frac{R_3}{\sin \theta_3}$$

$$L_4 = 0.5A - \frac{L_1 - 0.5D_3}{\tan \theta_3}$$

$$L_5 = 0.5A - \frac{L_1 - C}{\tan \theta_3}$$



Lug

- Maximum bending moment in lug, M_L .

$$M_L = PE$$

- Section modulus, lug, Z .

$$Z = \frac{A^2 t_L}{6}$$

- Bending stress, lug, σ_b .

$$\sigma_b = \frac{M_L}{Z}$$

- Thickness of lug required, t_L .

$$t_L = \frac{6M_L}{A^2 F_b}$$

- Tension at edge of pad, σ_T .

$$\sigma_T = \frac{P_L}{2L_4 t_L}$$

- Net section at pin hole, A_p .

$$A_p = 2L_3 t_L + 2t_p (D_3 - D_1)$$

- Shear stress at pin hole, τ .

$$\tau = \frac{P_L}{A_p}$$

- Net section at top of lug, A_n .

$$A_n = t_L \left(R_3 - \frac{D_1}{2} \right) + 2t_p \left(\frac{D_3 - D_1}{2} \right)$$

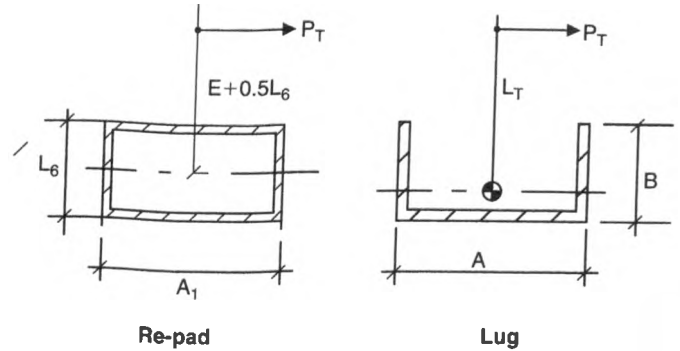
- Shear stress at top of lug, τ .

$$\tau = \frac{P_T}{A_n}$$

- Pin bearing stress, σ_p .

$$\sigma_p = \frac{P_T}{D_3 (t_L + 2t_p)}$$

Check Welds



- Polar moment of inertia, J_w .

$$\text{Re-pad: } J_w = \frac{(A_1 + L_6)^3}{6}$$

$$\text{Lug: } J_w = \frac{(A + 2B)^3}{12} - \frac{B^2(A + B)^2}{(A + 2B)}$$

- Moment, M_1 .

$$M_1 = L_T P_T$$

Lug Weld

- Find loads on welds.
- Transverse shear due to P_T , f_1 .

$$f_1 = \frac{P_T}{A + 2B}$$

- Transverse shear due to M_1 , f_2 .

$$f_2 = \frac{M_1 (B - N_T)}{J_w}$$

- Longitudinal shear due to M_1 , f_3 .

$$f_3 = \frac{M_1 B}{J_w}$$

- Combined shear load, f_r .

$$f_r = \sqrt{(f_1 + f_2)^2 + f_3^2}$$

- Size of weld required, w_1 .

$$w_1 = \frac{f_r}{0.707F_s}$$

Note: If w_1 exceeds the shell plate thickness, then a re-pad must be used.

Re-pad Weld

- Moment, M_2 .

$$M_2 = P_T(E + 0.5L_6)$$

- Transverse shear due to P_T , f_1 .

$$f_1 = \frac{P_T}{2A_1 + 2L_6}$$

- Transverse shear due to M_2 , f_2 .

$$f_2 = \frac{0.5M_2L_6}{J_w}$$

- Longitudinal shear due to M_2 , f_3 .

$$f_3 = \frac{M_2L_6}{J_w}$$

- Combined shear load, f_r .

$$f_r = \sqrt{(f_1 + f_2)^2 + f_3^2}$$

- Size of weld required, w_1 .

$$w_2 = \frac{f_r}{0.707F_s}$$

Pad Eye Weld

- Unit shear load on pad, f_4 .

$$f_4 = \frac{P_T t_p \pi D_2}{2t_p + t_L}$$

- Size of weld required, w_3 .

$$w_3 = \frac{f_4}{0.707F_s}$$

Top Head Lug for Large Loads

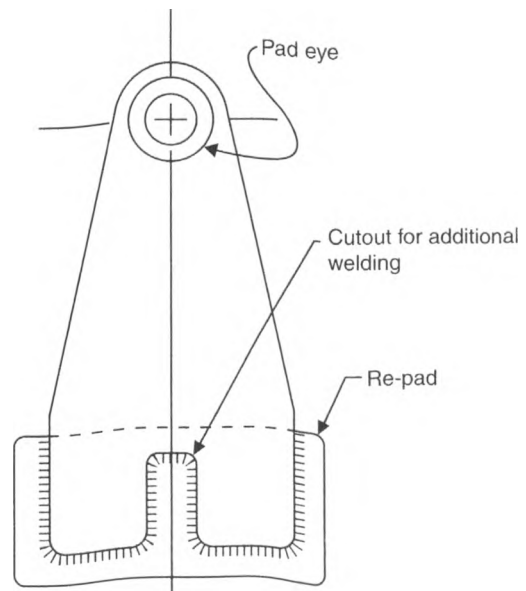


Table 10-9
Dimensions for top head or cone lugs

Type	Note	Total Erection Weight (tons)	Shackle Size (tons)	Lug Thickness t_L	A	B	C	E	R_3	W_1	Gusset Thickness t_g	Pads			Lift Hole Dia D_1	Lug Matl. Min. Yield (psi)
												D_3	t_p	W_2		
36-in. to 48-in. Inside Diameter																
1-A		0-30	35	1	12	12	7	13	3	$\frac{3}{8}$	$\frac{1}{2}$				$2\frac{1}{2}$	30,000
1-B		31-65	50	$1\frac{1}{2}$	14	12	8	14	4	$\frac{3}{4}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	3	30,000
1-C	1	66-100	50	$1\frac{3}{4}$	16	14	9	15	$4\frac{1}{2}$	1	$\frac{3}{4}$	8	$\frac{1}{2}$	$\frac{1}{4}$	3	30,000
54-in. to 72-in. Inside Diameter																
2-A		0-30	35	1	12	12	7	15	3	$\frac{3}{8}$	$\frac{1}{2}$				$2\frac{1}{2}$	30,000
2-B		31-65	50	$1\frac{1}{2}$	16	14	8	17	4	$\frac{5}{8}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	3	30,000
2-C	1	66-100	50	$1\frac{3}{4}$	18	14	9	18	$4\frac{1}{2}$	$\frac{7}{8}$	$\frac{1}{2}$	8	$\frac{1}{2}$	$\frac{1}{4}$	3	30,000
2-D	2	101-150	75	2	20	16	11	20	5	$1\frac{1}{4}$	$\frac{3}{4}$	9	$\frac{3}{4}$	$\frac{3}{8}$	$3\frac{1}{2}$	38,000
78-in. to 108-in. Inside Diameter																
3-A		0-30	35	1	14	10	6	18	3	$\frac{1}{2}$	$\frac{1}{2}$				$2\frac{3}{8}$	30,000
3-B		31-65	50	1	20	12	7	19	4	$\frac{5}{8}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
3-C	1	66-100	50	$1\frac{1}{4}$	22	14	9	21	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	8	$\frac{3}{4}$	$\frac{1}{2}$	$2\frac{7}{8}$	30,000
3-D	2	101-150	75	$1\frac{3}{4}$	22	16	10	23	5	$1\frac{1}{4}$	1	9	1	$\frac{3}{8}$	$3\frac{3}{8}$	38,000
3-E	3	151-200	130	2	25	18	12	25	$6\frac{1}{2}$	$1\frac{3}{8}$	1	12	1	$\frac{1}{2}$	$4\frac{3}{8}$	38,000
114-in. to 144-in. Inside Diameter																
4-A		0-30	35	1	14	10	5	20	3	$\frac{1}{2}$	$\frac{1}{2}$				$2\frac{3}{8}$	30,000
4-B		31-65	50	1	22	14	7	22	4	$\frac{1}{2}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
4-C	1	66-100	50	$1\frac{1}{4}$	26	14	9	25	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	8	$\frac{3}{4}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
4-D	2	101-150	75	$1\frac{3}{4}$	26	16	12	27	5	$1\frac{1}{4}$	1	9	1	$\frac{3}{8}$	$3\frac{3}{8}$	38,000
4-E	3	151-200	130	2	28	18	12	27	$6\frac{1}{2}$	$1\frac{3}{8}$	1	12	1	$\frac{1}{2}$	$4\frac{3}{8}$	38,000
150-in. to 180-in. Inside Diameter																
5-A		0-30	35	1	14	10	5	21	3	$\frac{1}{2}$	$\frac{1}{2}$				$2\frac{3}{8}$	30,000
5-B		31-65	50	1	22	14	6	23	4	$\frac{5}{8}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
5-C	1	66-100	50	$1\frac{1}{4}$	26	14	10	28	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	8	$\frac{3}{4}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
5-D	2	101-150	75	$1\frac{3}{4}$	26	16	12	30	5	$1\frac{1}{4}$	1	9	1	$\frac{3}{8}$	$3\frac{3}{8}$	38,000
5-E	3	151-200	130	2	28	18	12	30	$6\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	12	1	$\frac{1}{2}$	$4\frac{3}{8}$	38,000
186-in. to 216-in. Inside Diameter																
6-A		0-30	35	1	16	10	4	24	3	$\frac{1}{2}$	$\frac{1}{2}$				$2\frac{3}{8}$	30,000
6-B		31-65	50	1	24	14	6	26	4	$\frac{1}{2}$	$\frac{1}{2}$	7	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
6-C	1	66-100	50	$1\frac{1}{4}$	28	14	9	31	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	8	$\frac{3}{4}$	$\frac{1}{4}$	$2\frac{7}{8}$	30,000
6-D	2	101-150	75	$1\frac{3}{4}$	28	16	12	34	5	$1\frac{1}{4}$	1	9	1	$\frac{3}{8}$	$3\frac{3}{8}$	38,000
6-E	3	151-200	130	2	30	18	12	34	$6\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	12	1	$\frac{1}{2}$	$4\frac{3}{8}$	38,000

Notes:
 1. For 75-ton shackle, increase lift hole to 3.375
 2. For 130-ton shackle, increase lift hole to 4.375
 3. For 150-ton shackle, increase lift hole to 5.125

Procedure 10-7: Design of Flange Lugs

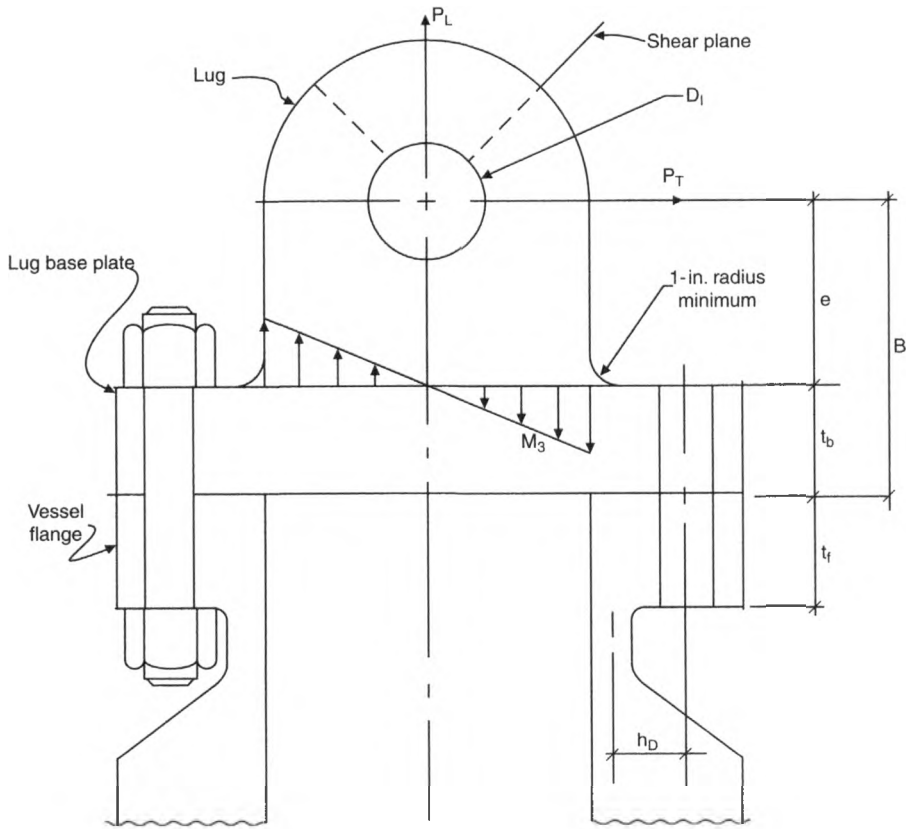
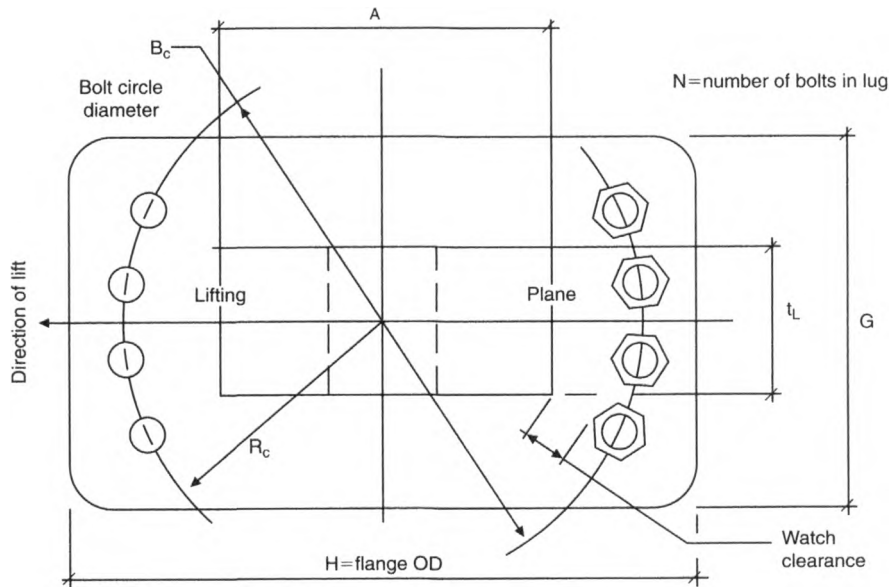


Table 10-10
Flange lug dimensions

Load Capacity (tons)	D ₁	t _L	t _b	A	B	G	H	E
50	3.38	2	2	8	11	12	30	9
100	5	3	3	14	12	24	36	9
200	6	4	4	18	14	30	40	10
400	8	5	5	20	16	36	46	11
600	9	6	6	24	22	40	58	16
800	10	9	7	28	24	42	60	17

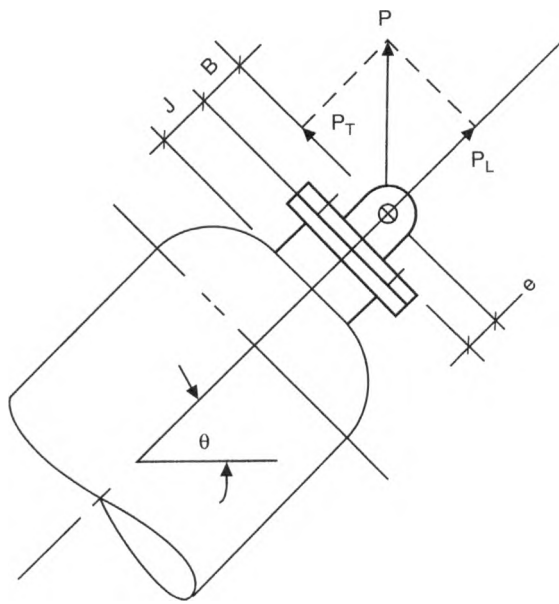
Table 10-11
Bolt properties

Bolt Size	A _b	A _s	T _b
0.5-13	0.196	0.112	12
0.625-11	0.307	0.199	19
0.75-10	0.442	0.309	28
0.875-9	0.601	0.446	39
1-8	0.785	0.605	51
1.125-8	0.994	0.79	56
1.25-8	1.227	1	71
1.375-8	1.485	1.233	85
1.5-8	1.767	1.492	103
1.75-8	2.405	2.082	182
2-8	3.142	2.771	243
2.25-8	3.976	3.557	311
2.5-8	4.909	4.442	389
2.75-8	5.94	5.43	418
3-8	7.069	6.506	501
3.25-8	8.3	7.686	592
3.5-8	9.62	8.96	690
3.75-8	11.04	10.34	796
4-8	12.57	11.81	910

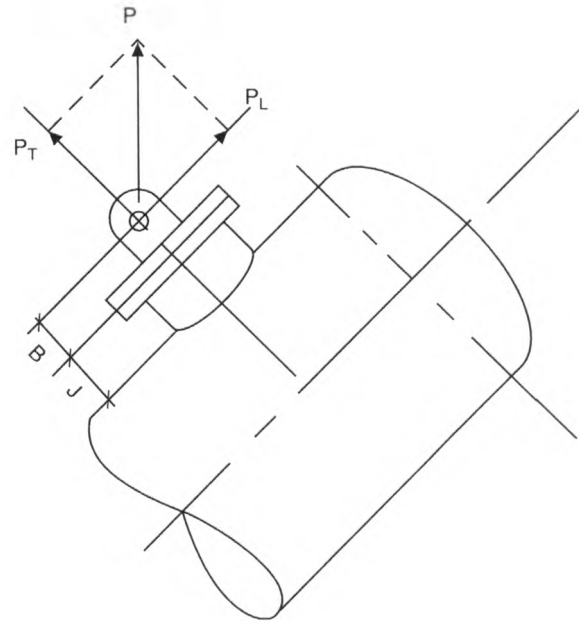
Table 10-12
Values of S_u

Bolt Dia, d _b	Material	S _u (ksi)
< 1	A-325	120
1.125-1.5	A-325	105
1.625-2.5	A-193-B7	125
2.625-4	A-193-B7	110

Top Flange Lug



Side Flange Lug



$$P_L = P \sin \theta$$

$$P_T = P \cos \theta$$

$$P_E = \frac{P_L}{A} + \frac{3P_T e}{A^2}$$

$$M_1 = P_T B$$

$$M_2 = P_T (B + J)$$

$$M_3 = P_T e$$

$$M_u = X_n \cos \alpha_n N_b$$

$$M_a = \frac{M_u M_1}{\sum M_u}$$

$$X_n = R_b \cos \alpha_n$$

$$y_n = R_b \sin \alpha_n$$

$$F_n = \frac{M_a}{X_n N_b}$$

$$f_s = \frac{P_T}{N}$$

$$\sigma_T = \frac{F_n}{A_s}$$

$$F_s = 15 \text{ ksi} \left(1 - \frac{\sigma_T A_b}{T_b} \right)$$

$$A_s = 0.7854 (d - 0.1218)^2$$

$$\tau = \frac{f_s}{A_s} < F_s$$

$$T_b = 0.75_u A_s$$

$$0.6F_y < F_T < 40 \text{ ksi}$$

Design Process

1. Determine loads
2. Check of lug:
 - a. Shear at pin hole.
 - b. Bending of lug.
 - c. Bearing at pin hole.
3. Check of base plate.
4. Check of nozzle flange.
5. Check of flange bolting.
6. Check of local load at nozzle to head or shell junction.

Step 1: Determine loads.

- Determine loads P_T and P_L for various lift angles, θ .
- Determine uniform loads w_1 and w_2 for various angles, θ .
- Using w_1 and w_2 , solve for worst case of combined load, P_E .
- Determine worst-case bending moment in lug, M_3 .

Step 2: Check of lug.

- a. Shear at pin hole:
 - Area required, A_r .

$$A_r = \frac{P_E}{F_s}$$

- Area available at pin hole, A_a .

$$A_a = (At_L) - (D_1t_L)$$

- b. Bending of lug due to M_3 :

- Section modulus, Z .

$$Z = \frac{t_L A^2}{6}$$

- Bending stress, lug, σ_b .

$$\sigma_b = \frac{M_3}{Z}$$

- c. Bearing at pin hole:

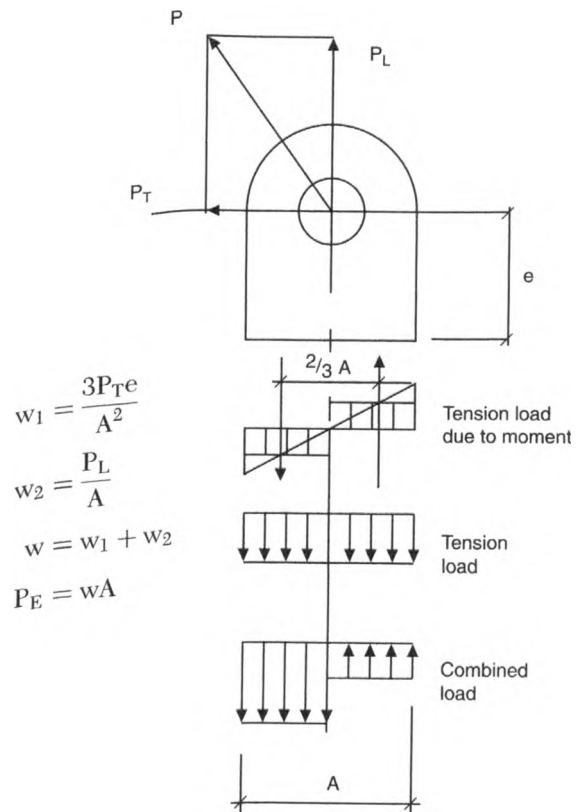
- Bearing required at pin hole A_r .

$$A_r = \frac{P_E}{F_p}$$

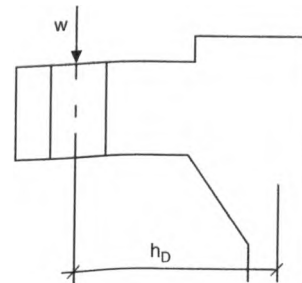
- Bearing available, A_a .

$$A_a = D_2 t_L$$

Maximum Tension in Lug



Check of Nozzle Flange



- Unit load, w .

$$w = \frac{P_E}{\pi B_c}$$

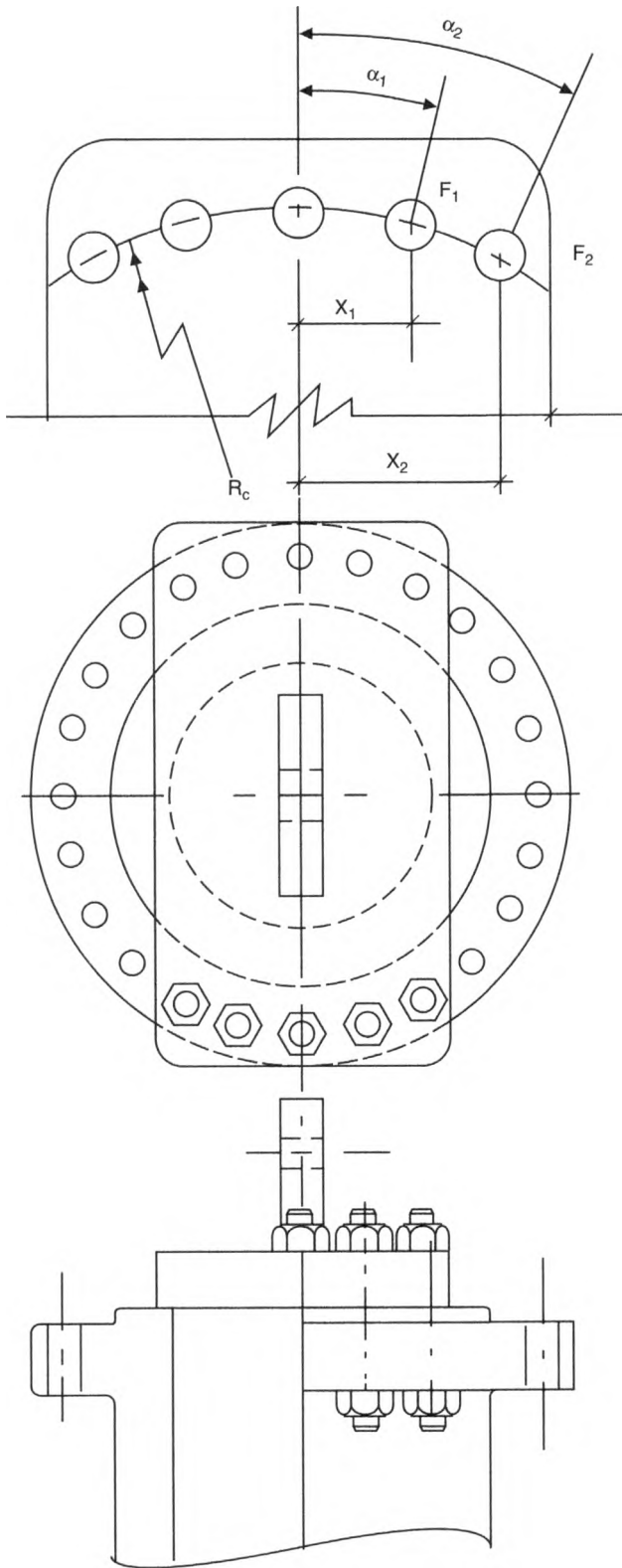
- Bending moment, M .

$$M = wh_D$$

- Bending stress, σ_b .

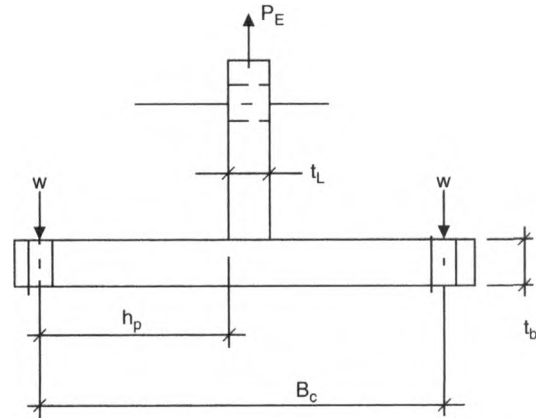
$$\sigma_b = \frac{6M}{t_f^2}$$

Bolt Loads for Rectangular Lugs



Design of Full Circular Base Plate for Lug

- If a full circular plate is used in lieu of a rectangular plate, the following evaluation may be used.



- Unit load on bolt circle, w .

$$w = \frac{P_E}{\pi B_C}$$

- Edge distance from point of load, h_p .

$$h_p = \frac{B_C - t_L}{2}$$

- Bending moment, M .

$$M = wh_p$$

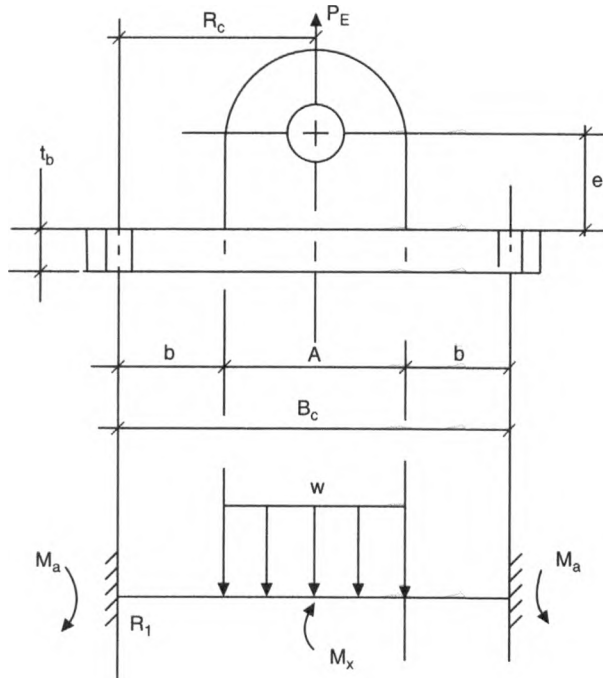
- Bending stress, σ_b .

$$\sigma_b = \frac{6M}{t_b^2}$$

- Check bolting same as rectangular flange.

Design of Lug Base Plate

(From R. J. Roark, *Formulas for Stress and Strain*, McGraw-Hill Book Co, 4th Edition, Table III, Case 34.)



- Uniform load, w .

$$w = \frac{P_E}{A}$$

- End reaction, R_1 .

$$R_1 = \frac{wA}{2}$$

- Edge moment, M_a .

$$M_a = \frac{wA}{24B_c} \left[\frac{24R_c^3}{B_c} - \frac{6(b+A)A^2}{B_c} + \frac{3A^3}{B_c} + 4A^2 - 24R_c^2 \right]$$

- Moment at midspan, M_x .

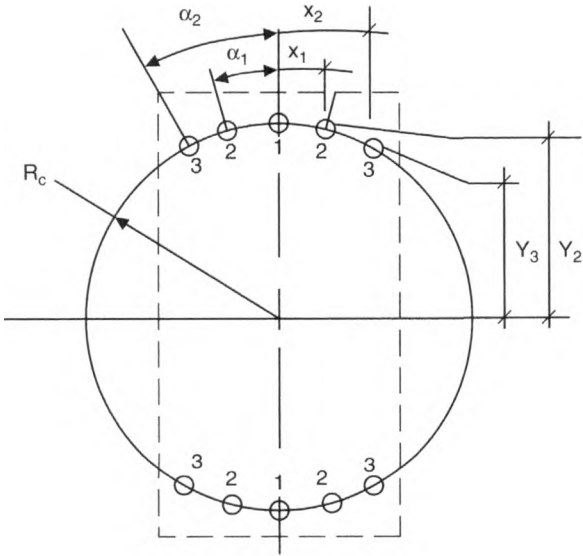
$$M_x = M_a + R_1R_c - \frac{wA}{2} \left[\frac{(R_c - b)^2}{A} \right]$$

- Thickness required, t_b .

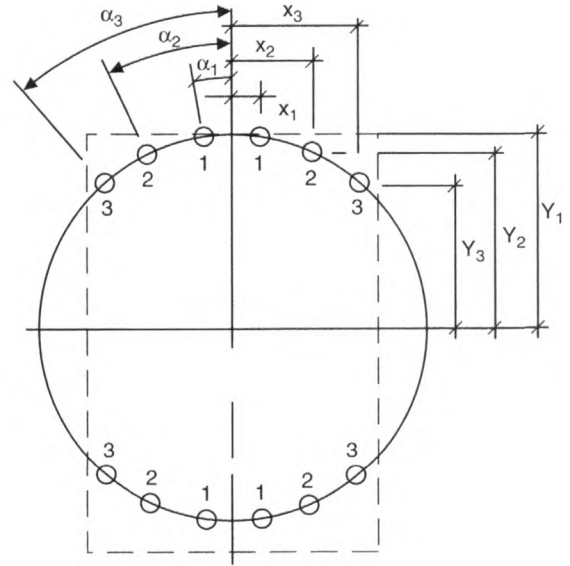
$$t_b = \sqrt{\frac{6M_x}{GF_b}}$$

Check of Bolts

Case 1: Bolts on Centerline

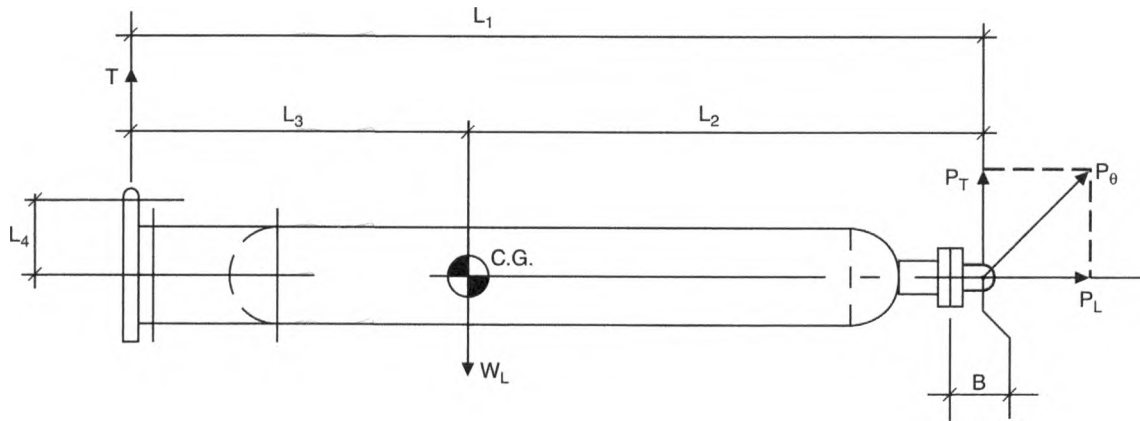


Case 2: Bolts Straddle Centerline



Bolt	1	2	3	4	5	Bolt	1	2	3	4	5
α_n						α_n					
x_n						x_n					
y_n						y_n					
N_b						N_b					
M_u						M_u					
M_a						M_a					
F_n						F_n					
σ_T						σ_T					
F_s						F_s					

Sample Problem: Top Flange Lug



Given

- $L_1 = 90$ ft
- $L_2 = 50$ ft
- $L_3 = 40$ ft
- $L_4 = 9.5$ ft
- F_y bolting = 75 ksi
- F_y lug = 36 ksi
- F_y flange = 36 ksi
- $F_s = 0.4(36) = 14.4$ ksi
- $F_T = 0.6(36) = 21.6$ ksi
- $F_b = 0.66(36) = 23.76$ ksi
- $W_L = 1200$ kips
- $B_c = 54$ in.
- $R_c = 27$ in.
- $B = 22$ in.
- $t_b = 6$ in.
- $t_L = 6$ in.
- $t_f = 11$ in.
- $D_1 = 9$ in.

- $D_2 = 8$ in.
- Bolt size = 3-1/4-8 UNC
- $A_b = 8.3$ in.²
- $A_s = 7.686$ in.²
- $T_b = 592$ kips
- $S_u = 110$ ksi
- $e = 16$ in.
- $G = 40$ in.
- $A = 24$ in.
- $h_D = 9.5$ in.
- $b = 0.5(B_c - A)$

Results

- P_T max = 537 kips @ $\theta = 10^\circ$
- P_L max = 1200 kips @ $\theta = 90^\circ$
- P_E max = 1277 kips @ $\theta = 40^\circ$
- σ_T bolt, max = 20.11 ksi ≤ 40 ksi
- τ bolt, max = 6.98 ksi ≤ 10.77 ksi

Step 1: Determine loads.

θ	Angle of Lift, Degrees									
	0	10	20	30	40	50	60	70	80	90
T_θ	666	654	642	629	613	592	564	517	417	0
P_θ	534	546	558	571	587	608	636	683	783	1200
P_T	534	537	525	494	450	391	318	234	136	0
P_L	0	95	191	286	377	465	551	642	771	1200
w_1	0	3.96	7.96	11.92	15.71	19.38	22.96	26.75	32.13	50
w_2	44.5	44.75	43.75	41.16	37.5	32.58	26.5	19.5	11.33	0
w	44.5	48.71	51.71	53.08	53.21	51.96	49.46	46.25	43.46	50
P_E	1068	1169	1241	1274	1277	1247	1187	1110	1043	1200
M_1	11,748	11,814	11,550	10,868	9900	8602	6996	5148	2992	0
f_s , bolts (10)	53.4	53.7	52.5	49.4	45	39.1	31.8	23.4	13.6	0
f_s , bolts (12)	44.5	44.75	43.75	41.16	37.5	32.6	26.5	19.5	11.33	0
T , bolts (10)	6.94	6.98	6.83	6.42	5.85	5.08	4.13	3.04	1.77	0
T , bolts (12)	5.79	5.82	5.69	5.35	4.88	4.24	3.44	2.53	1.47	0

Step 2: Check bolts for tension load.

Case 1: N = (10) Bolts					Case 2: N = (12) Bolts				
α_n	0	15	30		7.5	22.5	37.5		
$\cos \alpha_n$	1	0.965	0.866		0.991	0.923	0.793		
X_n	0	7	13.5		3.52	10.33	16.44		
N_b	2	4	4		4	4	4		
M_u	0	27.05	46.76	$\Sigma = 73.81$	13.95	38.13	52.15	$\Sigma = 104.22$	
M_a	0	4329	7484	$\Sigma = 11,814$	1581	4322	5911	$\Sigma = 11,814$	
F_n	0	154.6	138.6		112.3	104.6	89.9		
σ_T		20.11	18.03		14.61	13.61	11.7		
F_s		10.77	11.21		11.93	12.13	12.53		

1.0 Check Lug

a. Shear at pin hole:

- Area required, A_r .

$$A_r = \frac{P_E}{F_S} = \frac{1277}{14.4} = 88.68 \text{ in.}^2$$

- Area available at pin hole, A_a .

$$A_a = (At_L) - (D_1 t_L) = (24 \cdot 6) - (9 \cdot 6) = 90 \text{ in.}^2$$

b. Bending of lug due to M_3 :

- Maximum moment, M_3 .

$$M_3 = P_{Te} = 537(16) = 8592 \text{ in.} - \text{kips}$$

- Section modulus, Z .

$$Z = \frac{t_L A^2}{6} = \frac{(6 \cdot 24^2)}{6} = 576 \text{ in.}^3$$

- Bending stress, lug, σ_b .

$$\sigma_b = \frac{M_3}{Z} = \frac{8592}{576} = 14.91 \text{ ksi}$$

- Thickness required, t_L .

$$t_L = \frac{6M}{F_b A^2} = \frac{6 \cdot 8592}{23.76(24^2)} = 3.76 \text{ in.}$$

c. Bearing at pin hole:

- Bearing required at pin hole, A_r .

$$A_r = \frac{P_E}{F_P} = \frac{1277}{32.4} = 39.41 \text{ in.}^2$$

- Bearing available, A_a .

$$A_a = D_2 t_L = 8 \cdot 6 = 48 \text{ in.}^2$$

2.0 Check Lug Base Plate

- Uniform load, w .

$$w = \frac{P_E}{A} = \frac{1277}{24} = 53.2 \frac{\text{kips}}{\text{in.}}$$

- End reaction, R_1 .

$$R_1 = \frac{P_E}{2} = \frac{1277}{2} = 638.5 \text{ kips}$$

- Edge moment, M_a .

$$M_a = \frac{wA}{24B_c} \left[\frac{24R_c^3}{B_c} - \frac{6(b+A)A^2}{B_c} + \frac{3A^3}{B_c} + 4A^2 - 24R_c^2 \right]$$

$$M_a = 0.985(8748 - 2496 + 768 + 2304 - 17,496) = -8049 \text{ in.} - \text{kips}$$

- Moment at mid, M_x .

$$M_x = M_a + R_1 R_c - \frac{wA}{2} \left[\frac{(R_c - b)^2}{A} \right]$$

$$M_x = -8049 + 17,240 - 3831 = 5360 \text{ in.} - \text{kips}$$

- Section modulus, Z .

$$Z = \frac{(t_b^2 G)}{6} = \frac{(6^2 \cdot 40)}{6} = 240 \text{ in.}^3$$

- Bending stress, σ_b .

$$\sigma_b = \frac{M_x}{Z} = \frac{5360}{240} = 22.33 \text{ ksi}$$

- Allowable bending stress, F_b .

$$F_b = 0.66F_y = 0.66(36) = 23.76 \text{ ksi}$$

3.0 Check of Vessel Flange

- Unit load, w .

$$w = \frac{P_E}{\pi B_c} = \frac{1277}{\pi 54} = 7.52 \frac{\text{kips}}{\text{in.}}$$

- Bending moment, M_b .

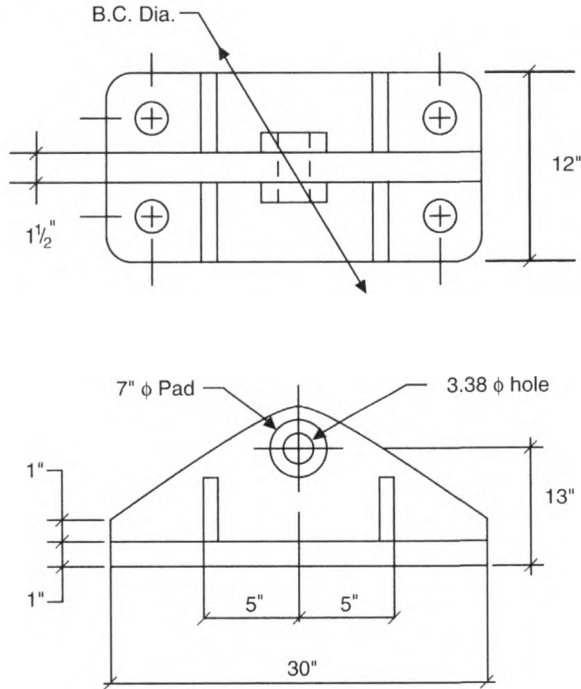
$$M_b = wh_D = 7.52(9.2) = 69.25 \text{ in.} - \text{kips}$$

- Bending stress, σ_b .

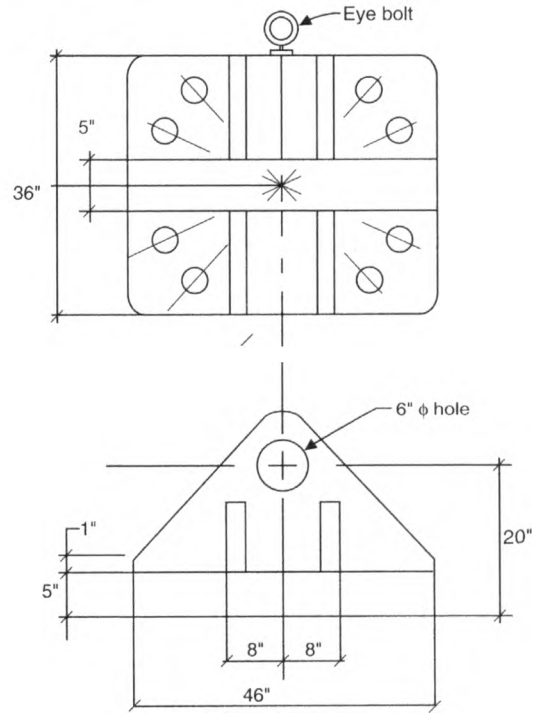
$$\sigma_b = \frac{6M_b}{t_f^2} = \frac{[6(69.25)]}{11.25^2} = 3.28 \text{ ksi}$$

Top Flange Lugs—Alternate Construction

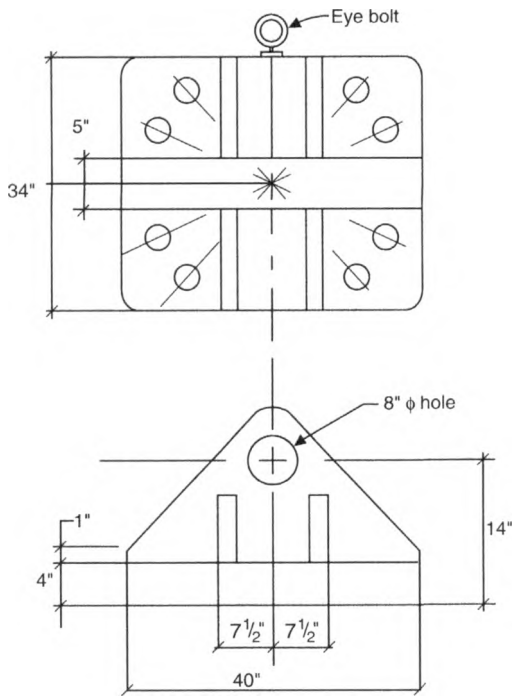
50-Ton Capacity



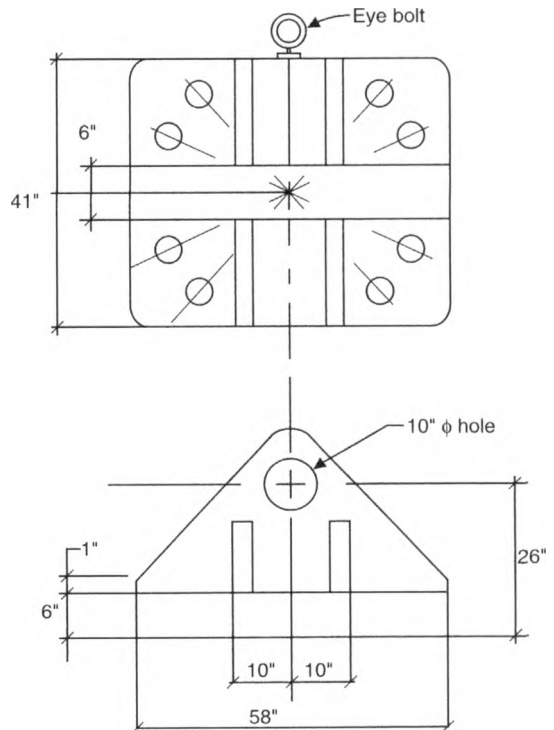
400-Ton Capacity



200-Ton Capacity

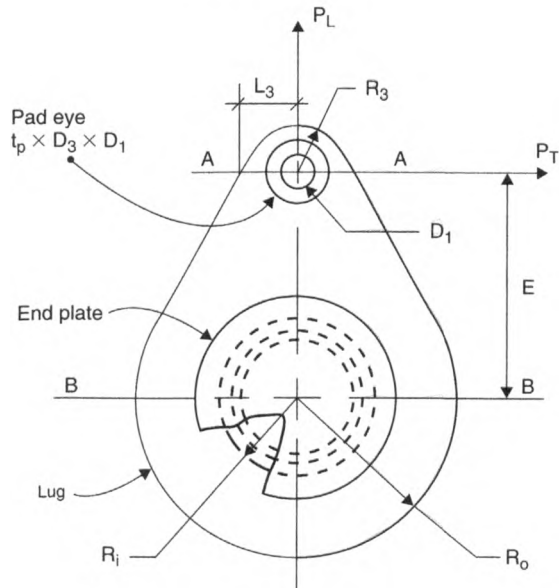


600-Ton Capacity

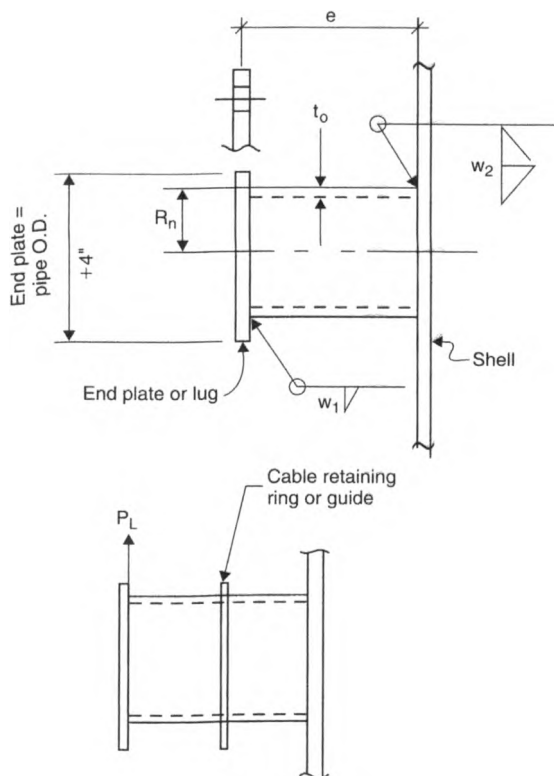


Procedure 10-8: Design of Trunnions

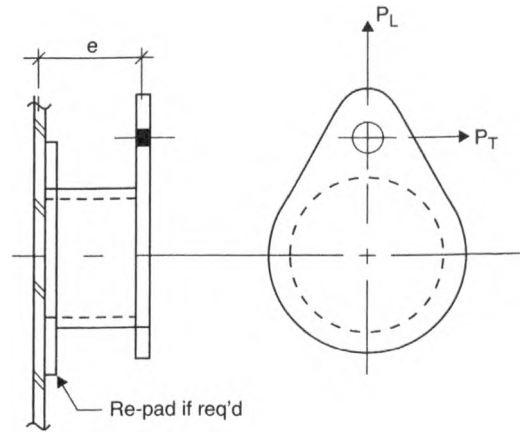
Lug Dimensions



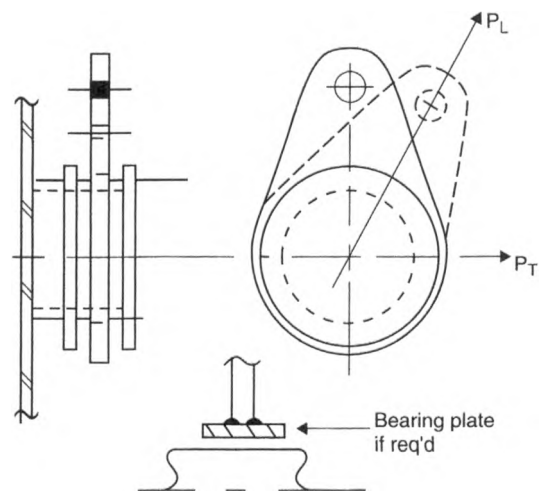
Dimensions for Trunnion



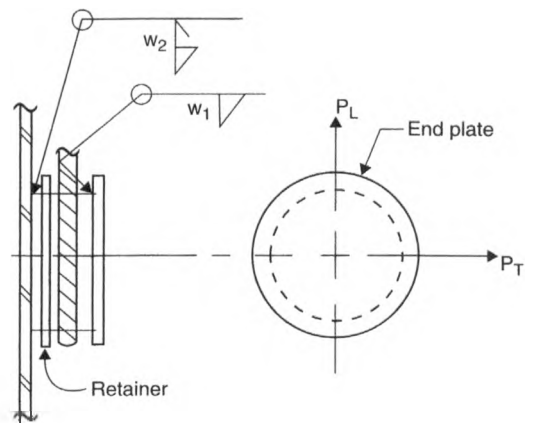
Type 1: Trunnion and Fixed Lug



Type 2: Trunnion and Rotating Lug



Type 3: Trunnion Only



Type 1: Trunnion and Fixed Lug

There are four checks to be performed:

1. Check lug.
2. Check trunnion.
3. Check welds.
4. Check vessel shell.

Check Lug

Transverse (vessel horizontal)

$$M = P_T E \quad \text{and} \quad Z = \frac{4R_o^2 t_L}{6}$$

Therefore,

$$t_L = \frac{1.5P_T E}{R_o^2 F_b}$$

Longitudinal (vessel vertical)

- Cross-sectional area at pin hole, A_p .

$$A_p = 2t_p(D_3 - D_1)$$

- Cross-sectional area at top of lug, A_n .

$$A_n = t_L \left(R_T - \frac{D_1}{2} \right) + 2t_p \left(\frac{D_3 - D_1}{2} \right)$$

- Shear stress, τ .

$$\tau = \frac{P_L}{A_p} \quad \text{or} \quad \tau = \frac{P_L}{A_n}$$

- Pin bearing stress, σ_p .

$$\sigma_p = \frac{P_L}{D_2(t_L + 2t_p)}$$

Check Trunnion

- Longitudinal moment, M_L (vessel vertical).

$$M_L = P_L e$$

- Torsional moment, M_T (vessel horizontal).

$$M_T = P_T E$$

- Bending stress, σ_b .

$$\sigma_b = \frac{M_L}{Z}$$

- Torsional shear stress, τ_T .

$$\tau_T = \frac{M_T}{2\pi R_n t_o}$$

Check Welds

- Section modulus of weld, S_w .

$$S_w = \pi R_n^2$$

- Polar moment of inertia, J_w .

$$J_w = 2\pi R_n^3$$

- Shear stress in weld due to bending moment, f_s .

$$f_s = \frac{M_L}{S_w}$$

- Torsional shear stress in weld, τ_T .

$$\tau_T = \frac{M_T R_n}{J_w}$$

- Size of welds required, w_1 and w_2 .

$w_1 >$ thickness of end plate

$w_2 =$ width of combined groove and fillet welds

$$w_2 = \frac{f_s}{F_s} > \frac{3}{8} \text{ in.}$$

Type 2: Trunnion and Rotating Lug

- Net section at Section A-A, A_p .

$$A_p = 2t_p(D_3 - D_1)$$

- Shear stress at pin hole, τ .

$$\tau = \frac{P_L}{A_p}$$

- Net section at Section B-B, A_n .

$$A_n = 2t_L(R_o - R_i)$$

- Shear stress at trunnion, τ .

$$\tau = \frac{P_L}{A_n}$$

- Minimum bearing contact angle for lug at trunnion, θ_B .

$$\theta_B = \frac{(15.9P_L)}{R_n t_L F_p}$$

- Pin hole bearing stress, σ_p .

$$\sigma_p = \frac{P_L}{D_3(t_L + 2t_p)}$$

Check Welds

- Longitudinal moment, M_L (vessel vertical).

$$M_L = P_L e$$

- Section modulus of weld, S_w .

$$S_w = \pi R_n^2$$

- Shear stress in weld due to bending moment, f_s .

$$f_s = \frac{M_L}{S_w}$$

- Size of welds required, w_1 and w_2 .

w_1 > thickness of end plate

w_2 = width of combined groove and fillet welds

$$w_2 = \frac{f_s}{F_s} > \frac{3}{8} \text{ in.}$$

Type 3: Trunnion Only

Vessel Vertical

- Longitudinal moment, M_L .

$$M_L = P_L e$$

- Bending stress in trunnion, σ_b .

$$\sigma_b = \frac{M_L}{Z}$$

Vessel Horizontal

- Circumferential moment, M_c .

$$M_c = P_L r e$$

- Bending stress in trunnion, σ_b .

$$\sigma_b = \frac{M_c}{Z}$$

Check Welds

- Longitudinal moment, M_L (vessel vertical).

$$M_L = P_L e$$

- Section modulus of weld, S_w .

$$S_w = \pi R_n^2$$

- Shear stress in weld due to bending moment, f_s .

$$f_s = \frac{M_L}{S_w}$$

- Size of welds required, w_1 and w_2 .

w_1 > thickness of end plate

w_2 = width of combined groove and fillet welds

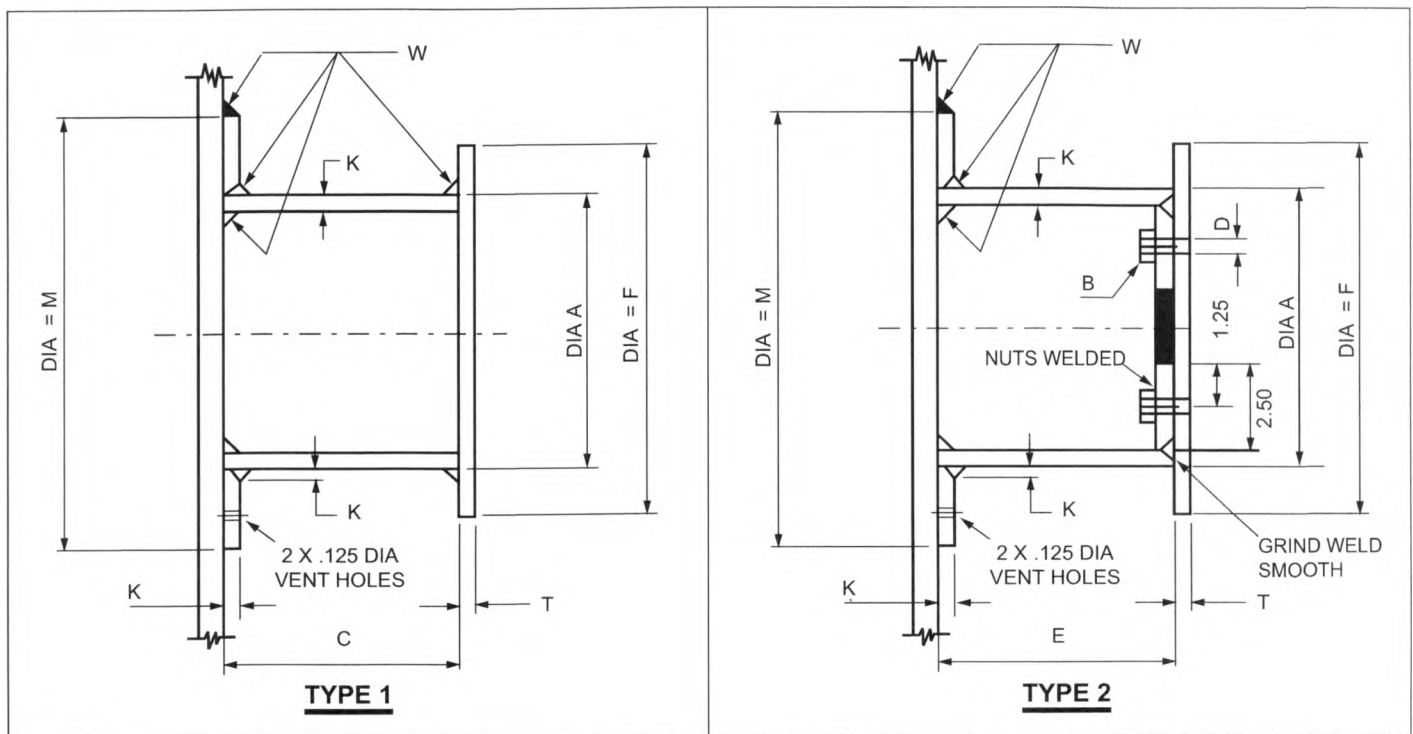
$$w_2 = \frac{f_s}{F_s} > \frac{3}{8} \text{ in.}$$

Table 10-13
Dimensions of trunnions

	Allowable Load, Tons	Pipe Size "A"	C	D	E	F	M	N	T	W	Weight, Lbs
1	0-5	4" Std	4	-	-	8.5	8	-	0.5	0.25	30
2	5-10	6" Std	5	-	-	10.5	12	-	0.5	0.25	55
3	10-15	8" Std	6	0.625	8	12.5	15	4	0.5	0.375	90
4	15-25	10" Std	6	0.625	8	15	18	4	0.75	0.375	150
5	25-35	12" Sch 80	6	0.75	8	17	21	4	0.75	0.375	240
6	35-45	12" Sch 80	6	0.75	8	17	21	4	0.75	0.5	240
7	45-60	14" Sch 80	6	0.75	8	18	24	8	1	0.5	375
8	60-75	14" Sch 80	7	0.875	8	18	24	8	1	0.625	400
9	75-100	16" Sch 80	7	0.875	8	25	27	8	1	0.625	625
10	100-125	16" Sch 80	7	0.875	8	25	27	8	1.125	0.875	660
11	125-150	18" Sch 80	7	0.875	8	27	30	12	1.125	0.875	850
12	150-200	18" Sch 80	8	0.875	8	27	30	12	1.125	1	875
13	200-250	20" Sch 80	8	0.875	10	30	36	16	1.25	1	1000
14	300	24" Sch 80	8	1	10	34	36	16	1.25	1.125	1440
15	400	24" Sch 80	8	1	10	34	40	20	1.375	1.125	1675
16	500	30" x 1.25"	10	1.25	12	42	48	20	1.375	1.375	2400
17	600	36" x 1.25"	10	1.25	12	48	60	24	1.5	1.375	3600

Notes:

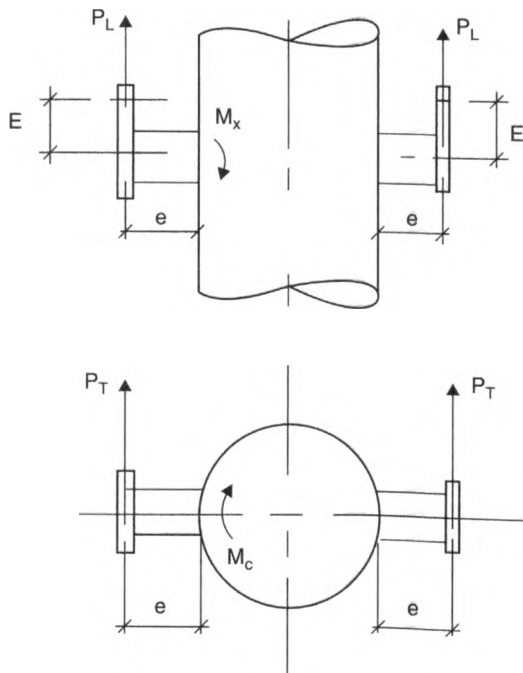
1. Do not use re-pads for cyclic service
2. $B = D - .125$
3. $K = \text{Pipe Wall Thk}$
4. Dimensions are given for reference only. All loadings and stress shall be checked prior to use.



Procedure 10-9: Local Loads in Shell Due to Erection Forces

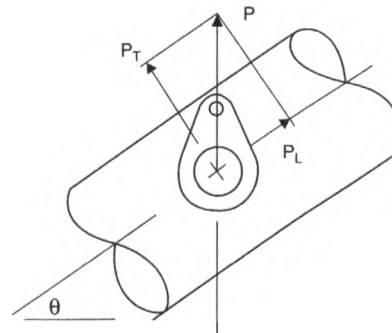
Trunnions

Fixed Lug Trunnion



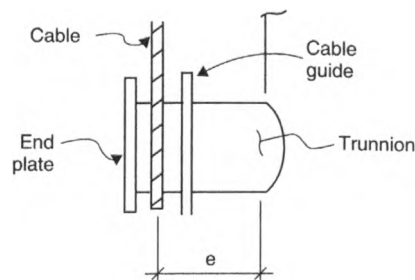
- *Maximum longitudinal moment, M_x .*
 $M_x = P_L e$
- *Maximum circumferential moment, M_c .*
 $M_c = P_T e$
- *Maximum torsional moment, M_T .*
 $M_T = P_T E$
- *Loads for any given lift angle, θ .*
 $P_L = 0.5P \sin \theta$
 $P_T = 0.5P \cos \theta$

Rotating Trunnion



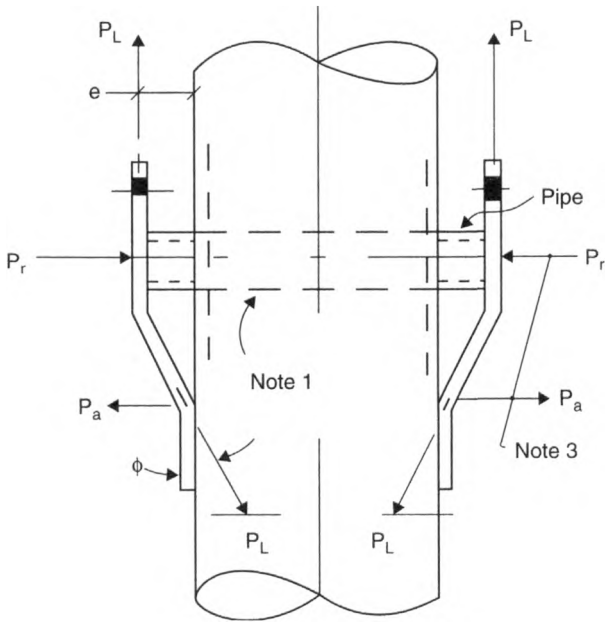
- *Maximum longitudinal moment, M_x .*
 $M_x = P_L e$
- *Maximum circumferential moment, M_c .*
 $M_c = P_T e$
- *Loads for any given lift angle, θ .*
 $P_L = 0.5P \sin \theta$
 $P_T = 0.5P \cos \theta$

Trunnion—No Lug



- *Maximum longitudinal moment, M_x .*
 $M_x = P_L e$
- *Maximum circumferential moment, M_c .*
 $M_c = P_T e$
- *Loads for any given lift angle, θ .*
 $P_L = 0.5P \sin \theta$
 $P_T = 0.5P \cos \theta$

Side Lugs



Notes:

1. Optional internal pipe. Remove after erection.
2. Radial load, P_r , is the axial load in the internal pipe stiffener if used in lieu of radial load in shell.
3. Circumferential ring stiffeners are optional at these elevations.

- Circumferential moment, M_c .

$$M_c = P_T e$$

- Longitudinal moment, M_x .

$$M_x = P_L e$$

- Load on weld group, f .

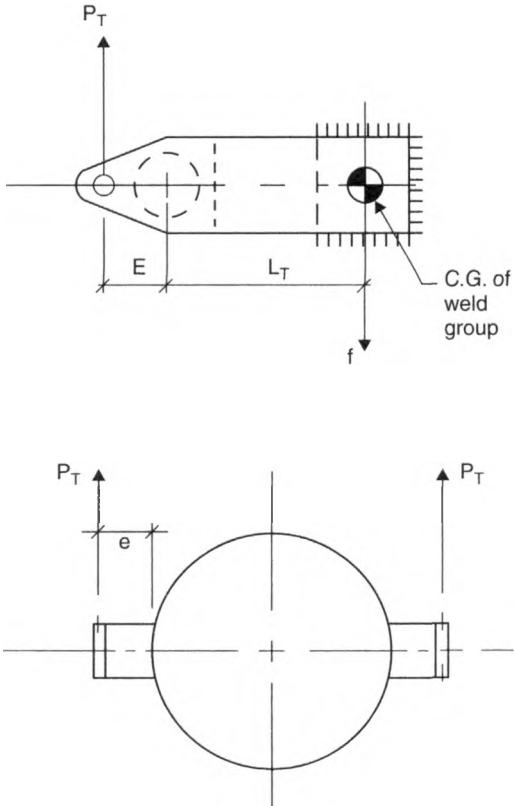
$$f = \frac{P_T E}{L_T}$$

- Radial loads, P_r and P_a .

$$P_r = P_L e$$

$$P_a = P_L \sin \phi$$

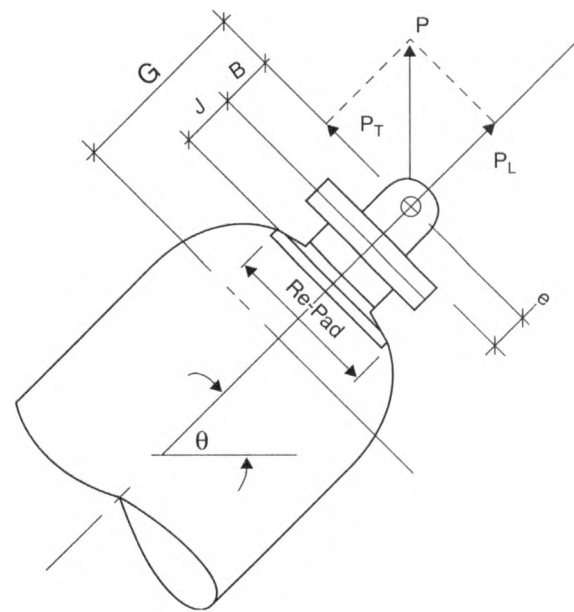
Top Flange Lug



- Loads, P_T and P_L .

$$P_T = P \cos \theta$$

$$P_L = P \sin \theta$$



- *Moment on flange, M .*

$$M = P_T B$$

- *Moment on head, M .*

$$M = P_T (B + J)$$

- *Moment on vessel, M .*

$$M = P_T G$$

- *Radial load on head and nozzle = P_L .*

- *Loads, P_T and P_L .*

$$P_L = P \cos \theta$$

$$P_T = P \sin \theta$$

- *Moment on flange, M .*

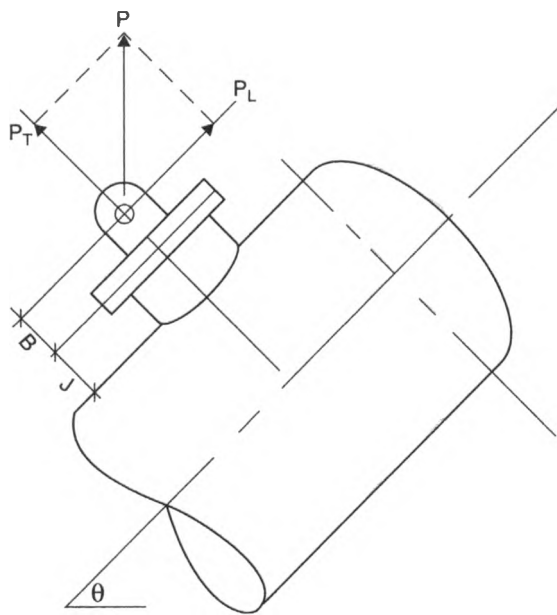
$$M = P_L B$$

- *Longitudinal moment on shell, M_x .*

$$M = P_T (B + J)$$

- *Radial load on shell and nozzle = P_T .*

Side Flange Lug



Procedure 10-10: Miscellaneous

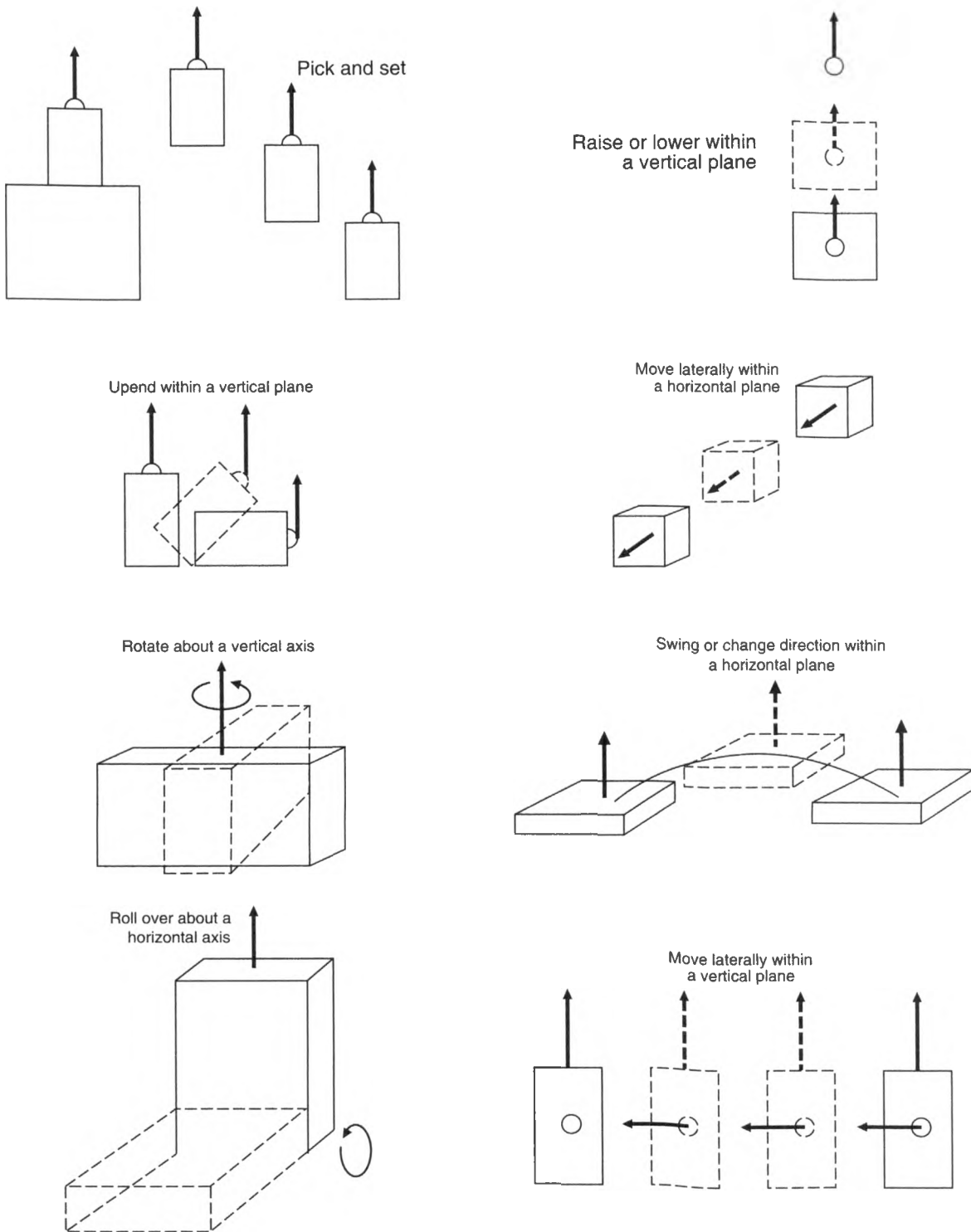


Figure 10-7. Fundamental handling operations. *Reprinted by permission of the Babcock and Wilcox Company, a McDermott Company.*

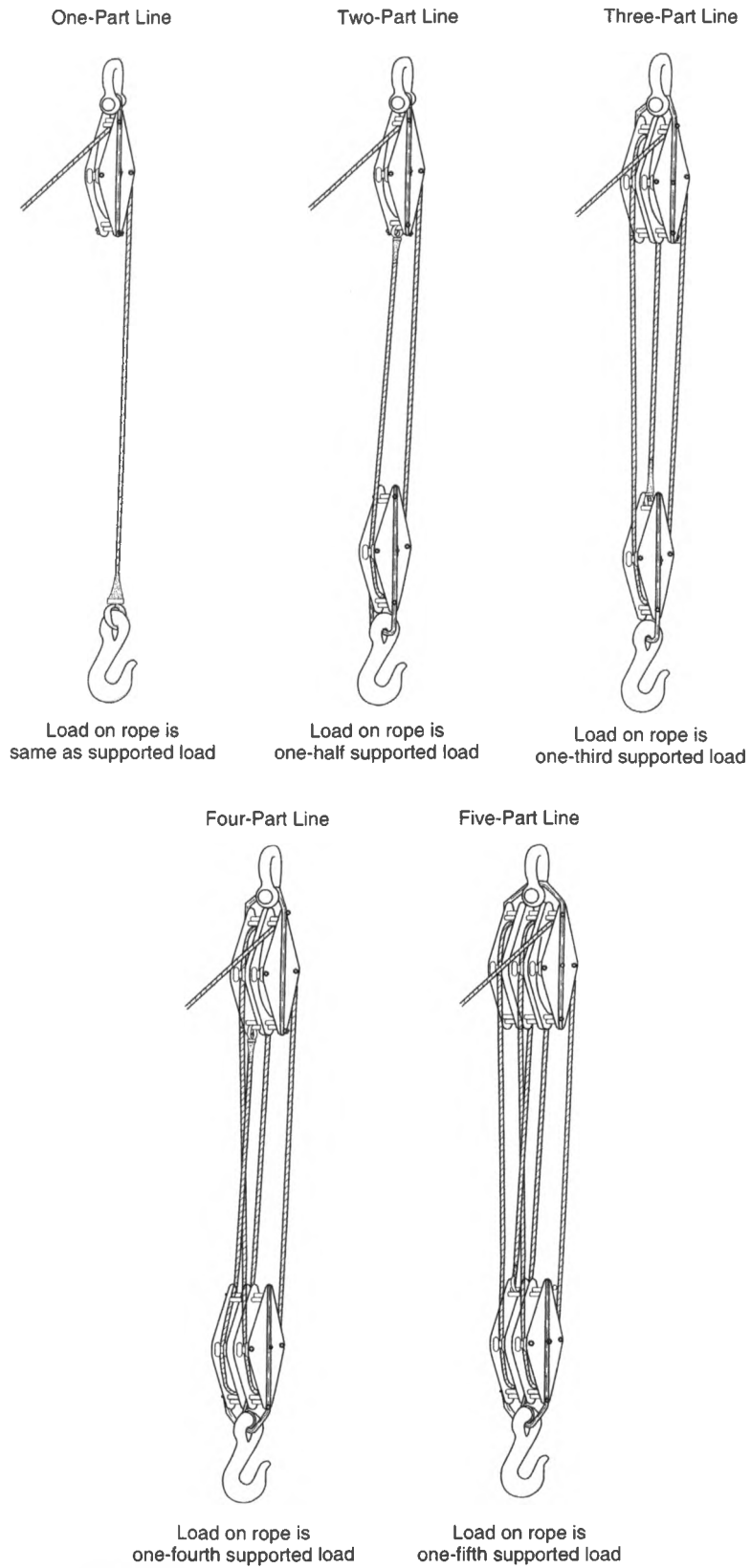
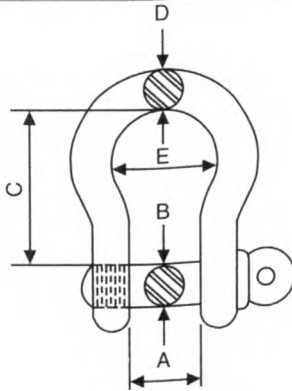
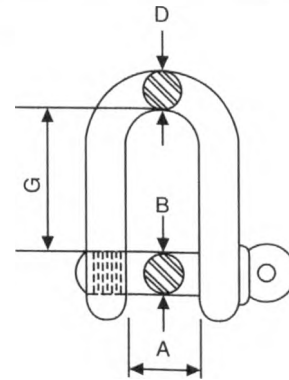


Figure 10-8. Loads on wire rope for various sheave configurations.

Table 10-14
Forged Steel Shackles



Anchor Shackle Screw Pin



Chain Shackle Screw Pin

Dimensions in Inches

Size D (in.)	Safe Load (lb)	D (min)	A	Tolerance A Dim.	B	B (min)	C	G	Tolerance C and G Dim.	E	F
1/4	475	7/32	15/32	± 1/16	5/16	9/32	1 1/8	3/8	± 1/16	3/4	1 1/16
3/8	1,050	1 1/32	2 1/32	± 1/16	7/16	25/64	1 7/16	1 1/4	± 1/8	1	3 1/32
7/16	1,450	25/64	2 3/32	± 1/16	1/2	29/64	1 11/16	1 7/16	± 1/8	1 1/8	1 1/16
1/2	1,900	29/64	1 3/16	± 1/16	5/8	9/16	1 7/8	1 5/8	± 1/8	1 3/8	1 5/16
5/8	2,950	9/16	1 1/16	± 1/16	3/4	43/64	2 13/32	2	± 1/8	1 5/8	1 9/16
3/4	4,250	43/64	1 1/4	± 1/16	7/8	25/32	2 27/32	2 3/8	± 1/4	2	1 7/8
7/8	5,750	25/32	1 7/16	± 1/16	1	57/64	3 5/16	2 19/16	± 1/4	2 1/4	2 1/8
1	7,550	57/64	1 1 1/16	± 1/16	1 1/8	1 1/32	3 3/4	3 3/16	± 1/4	2 1/2	2 3/8
1 1/8	8,900	1 1/32	1 27/16	± 1/8	1 1/4	1 7/64	4 1/4	3 9/16	± 1/4	2 3/4	2 5/8
1 1/4	11,000	1 7/64	2 1/32	± 1/8	1 3/8	1 15/64	4 1 1/16	3 15/16	± 1/4	3 1/8	3
1 3/8	13,300	1 15/64	2 1/4	± 1/8	1 1/2	1 1 1/32	5 1/4	4 7/16	± 1/4	3 1/2	3 5/16
1 1/2	15,600	1 1 1/32	2 3/8	± 1/8	1 5/8	1 29/64	5 3/4	4 7/8	± 1/4	3 3/4	3 5/8
1 3/4	21,500	1 35/64	2 1/2	± 1/8	2	1 25/32	7	5 3/4	± 1/4	4 1/4	4 1/8
2	28,100	1 25/32	3 1/4	± 1/8	2 1/4	2 1/64	7 3/4	6 3/4	± 1/4	5 1/4	5
2 1/4	36,000	2 1/64	3 3/4	± 1/8	2 1/2	2 15/64	9 1/4	7 1/8	± 3/4	5 1/2	5 1/4
2 1/2	45,100	2 15/64	4 1/8	± 1/8	2 3/4	2 15/32	10 1/2	8	± 3/4	6 1/4	6
3	64,700	2 1 1/16	5	± 1/8	3 1/4	2 29/32	13	11 1/2	± 3/4	6 3/4	6 1/2

Notes:

For shackles with safe loads greater than the maximum shown, use Crosby-Laughlin (The Crosby Group, Div. of American Hoist & Derrick Co, Tulsa, OK 74101), Skookum (Skookum Co., Inc., Portland, OR 97203), or equal with an ultimate strength at least 5 times the safe working load. Allowable loads are lower than OSHA requirements tabulated in Section 1926.251, Table H-19.

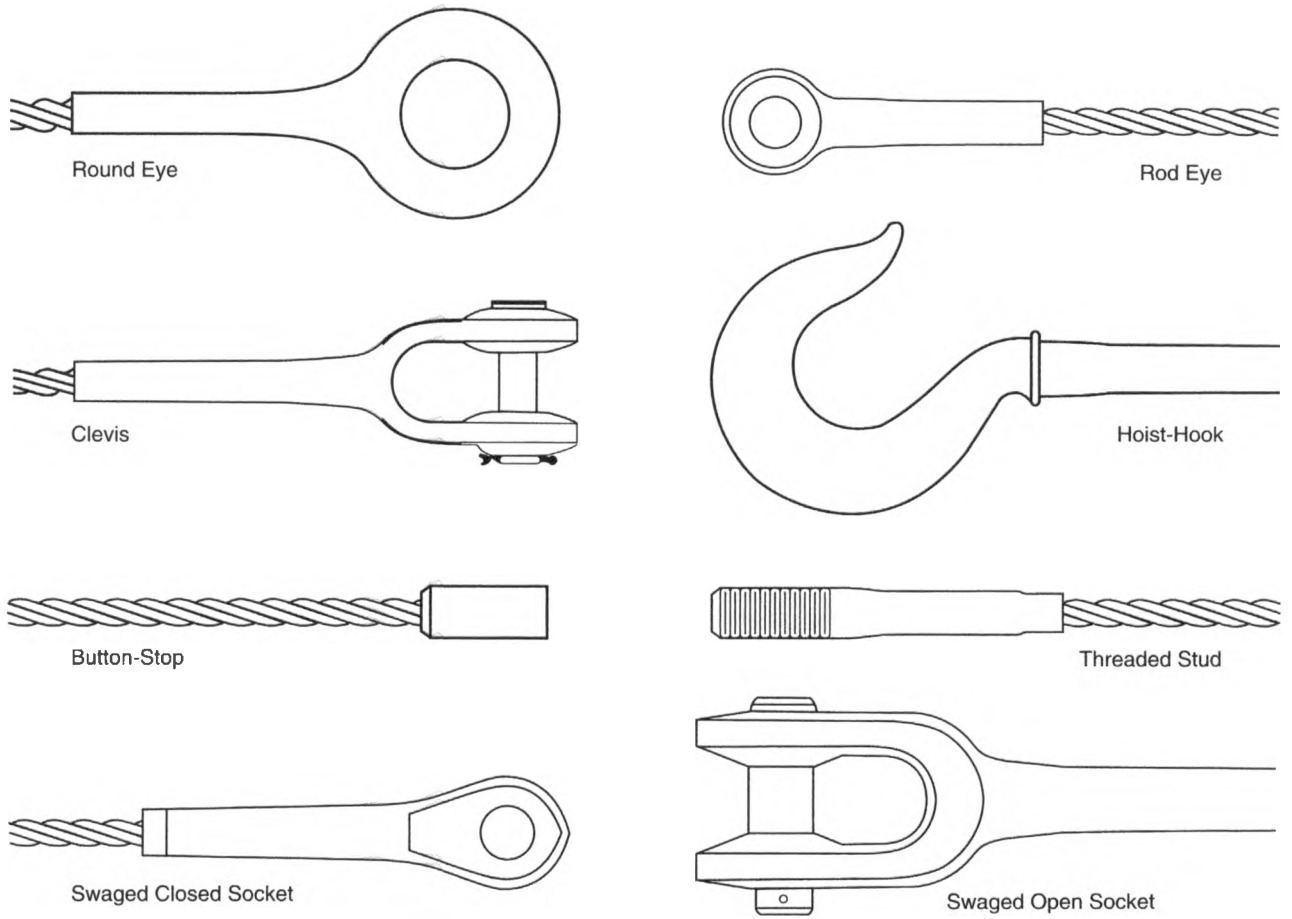


Figure 10-9. Wire rope end configurations.

Table 10-15
Material transportation and lifting

Material-Handling System	Description	Capacity t (t _m)
Site Transport:		
Flatbed trailers	Bed dimension 8 × 40 ft (2.4 × 12.2m)—deck height 60 in. (1524 mm) used to transport materials from storage to staging area.	20 (18)
Extendable trailers	Bed dimension up to 8 × 60 ft (2.4 × 18.3m)—deck height 60 in. (1524 mm) used to transport materials from storage to staging area.	15 (14)
Lowboy and dropdeck	Bed dimension up to 8 × 40 ft (2.4 × 12.2m)—deck height of 24 in. (610 mm) used to transport materials from storage to staging area.	60 (54)
Crawler transporter	Specially designed mechanism for handling heavy loads; Lampson crawler transporter, for an example of the Lampson design.	700 (635)
Straddle carrier	Mobile design to transport structural steel, piping, and other assorted items; straddle carrier, for an example of this design.	30 (27)
Rail	Track utilized to transport materials to installed location. Continuous track allows material in-stallation directly from delivery car.	as designed
Roller and track	Steel machinery rollers located relative to component center of gravity handle the load. Rollers traverse the web of a channel welded to top flange of structural member below.	2000 (1814)
Plate and slide	Sliding steel plates. Coefficient of friction—0.4 steel on steel, 0.09 greased steel on steel, 0.04 Teflon on steel. Sliding plate transport for movement of 1200 t (1089 t _m) vessel.	as designed
Air bearings or air pallets	Utilizes film of air between flexible diaphragm and flat horizontal surface. Air flow 3 to 200 ft ³ /min (0.001 to 0.09 m ³ /s). 1 lb (4.5N) lateral force per 1000 lb (454 kg) vertical load.	75 (68)
High line	Taut cable guideway anchored between two points and fitted with inverted sheave and hook.	5 (4.5)
Lifting:		
Chain hoist	Chain operated geared hoist for manual load handling capability. Standard lift heights 8 to 12 ft (2.4 to 3.7m).	25 (23)
Hydraulic rough terrain cranes	Telescopic boom mounted on rubber tired self-propelled carrier.	90 (82)
Hydraulic truck cranes	Telescopic boom mounted on rubber tired independent carrier.	450 (408)
Lattice boom truck cranes	Lattice boom mounted on rubber tired independent carrier.	800 (726)
Lattice boom crawler cranes	Lattice boom mounted on self-propelled crawlers.	2200 (1996)
Fixed position crawler cranes	Lattice boom mounted on self-propelled crawlers and equipped with specifically designed attachments and counterweights.	750 (680)
Tower gantry cranes	Tower mounted lattice boom gantry for operation above work site.	230 (209)
Guy derrick	Boom mounted to a mast supported by wire rope guys. Attached to existing building steel with load lines operated from independent hoist. Swing angle 360 deg (6.28 rad).	600 (544)
Chicago boom	Boom mounted to existing structure which acts as mast, and to which is attached boom topping lift and pivoting boom support bracket. Load lines operated from independent hoist. Swing angle from 180 to 270 deg (3.14 to 4.71 rad).	function of support structure
Stiff leg derrick	Boom attached to mast supported by two rigid diagonal legs and horizontal sills. Horizontal angle between each leg and sill combination ranges from 60 to 90 deg (1.05 to 1.57 rad); swing angle from 270 to 300 deg (4.71 to 5.24 rad).	700 (635)
Monorail	High capacity load blocks suspended from trolleys which traverse monorail beams suspended from boiler support steel. Provides capability to lift and move loads within boiler cavity.	400 (363)
Jacking systems	Custom designed hydraulic or mechanical system for high capacity special lifts.	as specified

Reprinted by permission of the Babcock and Wilcox Company, a McDermott Company.

Notes

1. This procedure is for the design of the vessel and the lifting attachments only. It is not intended to define rigging or crane requirements.
2. Lifting attachments may remain on the vessel after erection unless there is some process- or interference-related issue that would necessitate their removal.
3. Load and impact factors must be used for moving loads. It is recommended that a 25% impact factor and a minimum load factor of 1.5 be used. The combined load and impact factor should be 1.5–2.0.
4. Allowable stress compression should be $0.6F_y$ for structural attachments and ASME Factor “B” times 1.33 for the vessel shell.
5. Vessel shipping orientation should be established such that a line through the lifting lugs is parallel to grade if possible. This prevents the vessel from having to be “rolled” to the correct orientation for loading and offloading operations.
6. If a spreader beam is not used, the minimum sling angle shall be 30° from the horizontal position. At 30° , the tension in each sling is equal to the total design load. Thus a load factor of 2 is mandatory for these cases. This requires that each lug be designed for the full load.
7. Vessels should never be lifted by a nozzle or other small attachments unless specifically designed to do so.
8. All local loads in vessel shell or head resulting from loadings imposed during erection of the vessel shall be analyzed using a suitable local load procedure.
9. Tailing attachment shall be designed such that they may be unbolted without having to get under the load while it is suspended. As an alternative, the vessel must be set down at grade before a person can get under the base ring to unbolt the tailing beam. Be advised that the base and skirt may not be designed for point support if cribbing is used to build up the base for access.
10. A tailing lug, as opposed to a tailing beam, allows the load to be disconnected from the vessel without a person’s getting under a suspended load to unhook.
11. This procedure assumes that the pin diameter is no less than $\frac{1}{16}$ in. less than the hole diameter. If the pin diameter is greater than $\frac{1}{16}$ in. smaller than the hole diameter, then the bearing stresses in the lug at the contact point are increased dramatically due to the stress concentration effect.
12. Internal struts in the skirt or base plate are required only if the base/skirt configuration is overstressed.
13. If bearing or shear stresses are exceeded in the lug, add pad eyes.
14. Trunnions may be used as tailing devices as long as the resulting local loads in the skirt are analyzed.
15. Do not use less than Schedule 40 pipe for trunnions.
16. Specific notes for trunnions:
 - a. Type 1, fixed lug: Normal use but generally for small to medium vessels (less than 100 tons).
 - b. Type 2, rotating lug: Best use is when multiple vessels are to be lifted with the same lug. The lug may be removed by removing the end plate and sliding the lug off. Then the lug is reinstalled on the next vessel. For heavier loads, an internal sleeve should be attached to the lug to increase the bearing area on the trunnion.
 - c. Type 3, trunnion only: No size limitation or weight limitation. The cable and trunnions should be lubricated prior to lifting to prevent the cables from binding. The bend radius of the cables may govern the diameter of the trunnion. Check with erection contractor.