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BUCKLING STRESS CHECK FOR A VERTICAL VESSEL

When a vertical vessel is upended by using a lift crane and a tail crane, one of the stress checks that must be made is for buckling of the shell. Buckling usually occurs close to the CG on the top side of the vessel as the vessel is being lifted from the horizontal, ie, in the initial pick position (IPP). The shell at the bottom is overstressed as well but as it is in tension, only the buckling of the shell due to the compression at the top can be discernible. The buckling is sort of like pressing on the side of an empty Pepsi can. As you press harder, the ends rotate toward the middle and the shell buckles in at the center.

But, buckling can happen at other points on the vessel where the shell diameter (ϕ) is relative small and the wall thickness is thin, ie, with a vessel that has several ϕ's and where the shell thickness varies from the skirt to the top tangent line (T/L). In this case, each area that is suspect must be checked for buckling. The information below shows how to check a vessel for buckling. Calculating the bending moment at each suspect point is left up to the reader.

A thin wall vessel that has a small ϕ and is quite tall is more likely to be subjected to buckling than a vessel that is short and has a large ϕ.

The following five reference sheets provide the theory behind the buckling stress.

Reference sheet 1 gives the notation and allowable buckling stress development

Reference sheets 2 & 3 give more information of allowable buckling stress

Reference sheet 4 shows a graph of the above information

Reference sheet 5 shows the procedure for checking the buckling stress

1.0 INTRODUCTION

When lifting heater stacks and vessels, the possibility of buckling is an important consideration and should be check for all such lifts. Also, excessive bending in lined stacks must be limited to prevent damage to refractories.

2.0 NOTATION

In the following, this notation will be used:

f_c = critical buckling stress

F_B = allowable bending stress

E = modulus of elasticity

t = shell thickness

R = inside radius of cylinder

D = inside diameter of cylinder

F_y = yield stress of cylinder material

3.0 ALLOWABLE BUCKLING STRESS

In the past, various formulas have been used to arrive at an allowable buckling stress. Some of these are:

3.1 Formula per Noel Owen (developed from Alcoa Structural Handbook, 1950)

$$F \text{ critical} = f_c = 34{,}700 - 1150\ (R/t)^{1/2}$$

$$F\text{allowable} = F_B = \frac{F_c}{3} \leq 10\ \text{K/in}^2$$

3.2 Chicago Bridge and Iron Formula:

$$F_B = \left[(10^6)\ \frac{t}{R}\right]\left[2 - \frac{2}{3}\ \left(100\ \frac{t}{R}\right)\right]$$

3.3 Timoshenko Formula (Theory of Elastic Stability, 1936)

$$F_C = \frac{E}{\sqrt{3(1-u^2)}}\left(\frac{t}{R}\right) = 0.6E\left(\frac{t}{R}\right) \quad \text{for } u = 0.3$$

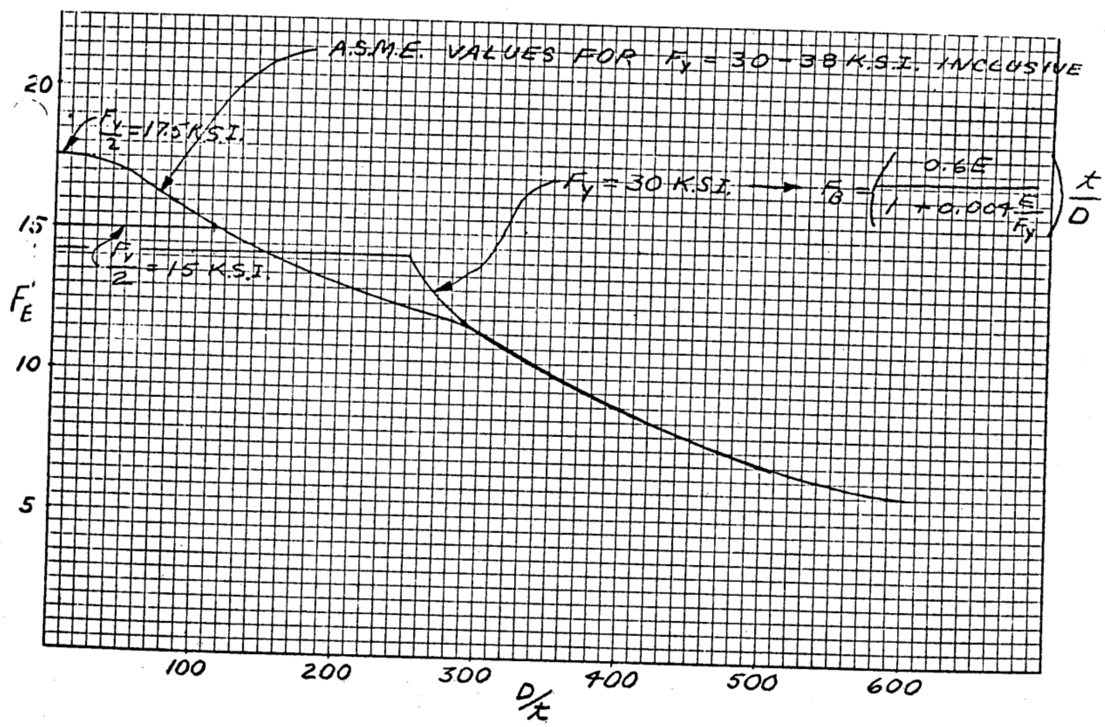
u = Poisson's ratio

$F_B = f_c/3 \leq 10\ \text{k/in}^2$

3.4 Wilson & Newmark Formula (University of Illinois Engineering Station Bulletin #255, 1933)

$$F_B = 1.5 \times 10^6 \left(\frac{t}{R}\right) \leq \frac{F_y}{3}$$

The Structural Department bases their allowable buckling stress for stacks on Donnell's equation for thing wall cylinders. The Vessel Department uses graphs from the A.S.M.E. Boiler and Pressure Vessel Code. A comparison showed that by both methods, the allowable stresses were the same for most cases.



In the past, allowable stress used by Construction Engineering were taken as 1/3 of the critical stress. A.S.M.E. Vessel Code and the Structural Department base their allowable on 1/2 f_c. Using 1/2 the critical is justified for Construction Engineering for the following reasons:

3.5 This is the accepted method used by Fluor Vessel and Structural Departments.

3.6 When designing Vessel Skirts, the Vessel Department increases the allowable buckling stress to 1.33 for earthquake and to 1.50 for hydrostatic conditions.

3.7 Timoshenko book on Elastic Stability indicates the critical buckling stress for bending of cylinder is actually 1.4 higher than calculated assuming a uniform compression load.

3.7.1 Equation for Buckling

Formulas based on equation for buckling of a cylinder with a uniform load.

$$f_c = E\left[\frac{0.6\,\frac{t}{R} - 10^{-7}\left(\frac{R}{t}\right)}{1 + 0.004\,E/F_y}\right] \qquad (1)$$

For values of $\frac{R}{t}$ normally used $10^{-7}\left(\frac{R}{t}\right)$ is negligible and

replacing $R = \frac{D}{2}$:

$$f_c = \frac{1.2\,E}{1 + 0.004\,\frac{E}{F_y}}\left(\frac{t}{D}\right)$$

Using allowable Safety Factor of 2 against Buckling:

$$F_B = \frac{f_c}{2}$$

$$F_B = \frac{.6E}{1 + .004\,\frac{E}{F_y}}\left(\frac{t}{D}\right) \leq \frac{F_y}{2} \qquad (2)$$

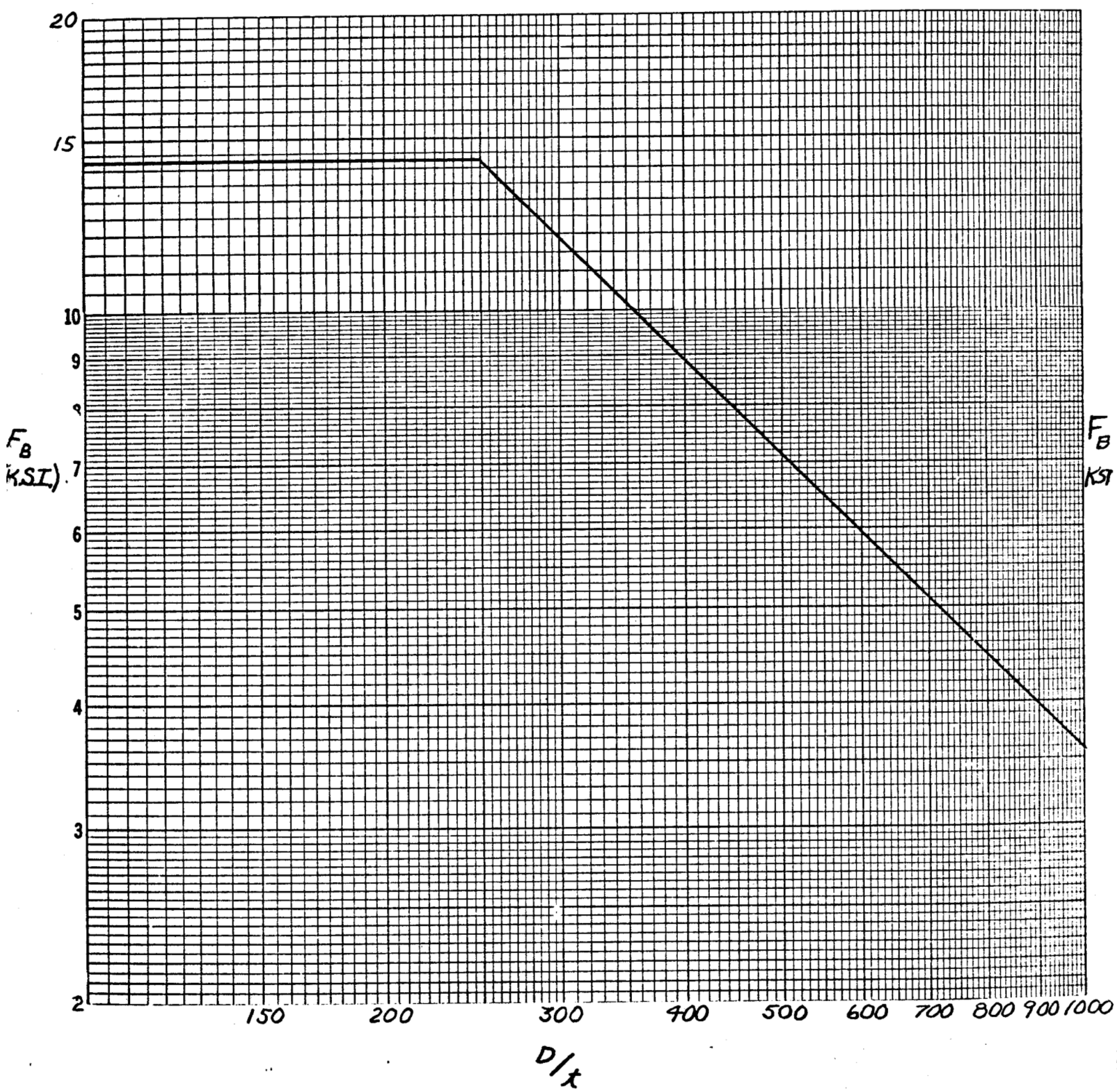
for F_y = 30 KSI and greater

$$F_B - 3558\,\frac{t}{D}\ \text{KSI} \leq 14.2$$

SEE GRAPH ON NEXT PAGE

FIGURE 2

F_y = 30 KSI or greater $\qquad F_B$ - 3558 $\frac{t}{D} \leq$ 14.23 KSI



4.0 PROCEDURE FOR CHECKING BUCKLING STRESS

4.1 Calculate the bending moment at critical sections.

4.2 Calculate the section modulas at point of critical moment.

$$\text{Section Modulas} - S = \pi R^2 t \text{ for } \frac{D}{t} \geq 60$$

$$S = \frac{\pi (D4 - d4)}{32d} \quad \text{for } \frac{D}{t} < 60$$

4.3 Calculate actual bending stress.

$$f_b = \frac{M}{S}$$

4.4 Determine F_B using figure 2 or Equation (2).

4.5 If $f_B \leq F_B$, lifting procedure is okay with respect to buckling.

4.6 If $f_B > F_B$, review lifting arrangement.

4.7 If the vessel or stack is lined, check for possible refractory damage due to bending. $\frac{L}{360} = \Delta$ = MAXIMUM DEFLECTION ALLOWED

For an example, see the calculation sheet below which shows a buckling check that I made a few weeks ago. The formulas and steps on the calculation sheet came from reference sheet 5 above.

The first step says to calculate the bending moment (M) at the critical section. In this case, the shell thickness is the same from the bottom T/L to the top T/L so the bending moment was taken at the CG where it is a maximum. If this had been a multiple ϕ vessel with a smaller ϕ just to the left of the CG (above the CG), then I would have also calculated the bending moment and checked for buckling at that point.

The next step says to calculate the section modulus (S) at the critical section. Note that to calculate the correct S, the D/t has to be determined. Then choose a formula for S depending on the D/t being lesser or greater than 60.

The next step is to calculate the actual bending stress (fb) where it = M/S.

Then determine the allowable bending stress (Fb) from either equation (2) or using the graph on reference sheet 4. Note that the graph is based on steel plate with a yield stress (Fy) of 30 ksi or greater.

If the actual bending stress fb is greater than the allowable bending stress Fb, then steps need to be taken to reduce the bending moment, ie, the vessel could be tailed higher up on the skirt or shell, or the lifting trunnions could be located further down on the shell below the Top T/L. As a last resort, the Rigging Engineer could go to the Vessels Engineer and have him increase the shell thickness at the point in question to reduce the bending stress. These are checks and steps that a Rigging Engineer needs to make in order to safely upend a vertical vessel without overstressing it. It is not too professional to say transport a vertical vessel from a fab shop in Japan to Saudi Arabia and then damage it during erection. These stresses and lifting details need to be worked out before the vessel is fabricated in the shop. If they haven't been addressed before the vessel arrives at site, then the Rigging Engineer must then take the appropriate steps to avoid over stressing the shell in buckling during upending, ie, by say not using the tail lug provided but tailing higher up on the skirt with a sling, etc.

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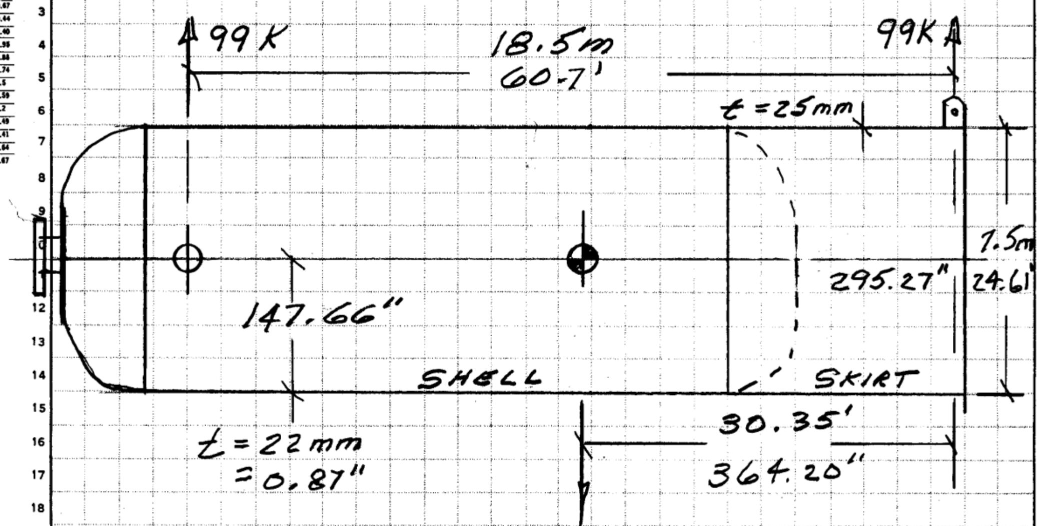
CALCULATIONS and SKETCHES

DATE 13 OCT 2012
CONT. NO.
BY K E GOODMAN CHK'D
SHEET NO. 1 OF 1

CHECK FOR BUCKLING STRESS:



$D/t = 295.27''/0.87'' = 339.39$

$\therefore S = \pi R^2 t = \pi * 147.66^2 in^2 * 0.87 in = 59,592.95 in^3$

$M = 364.20'' * 99k = 36,055.80$ K-in

$f_b = M/S = \quad = 1.65$ KSI

$F_y = 38$ KSI

$F_b = 10.5$ KSI SEE SHEET 4

$1.65 < 10.5 \Rightarrow$ OKAY

FORM E-050 REV. 4/69
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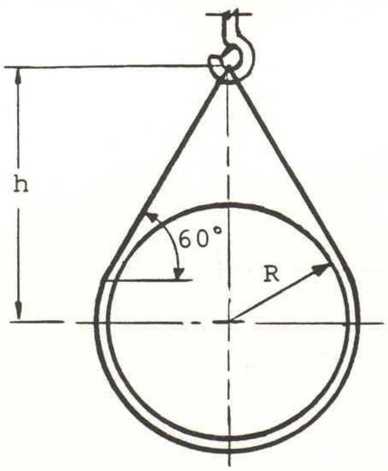
END OF BUCKLING CHECK

Just as a reminder, buckling or flattening of the bottom of the shell can also occur when a horizontal or vertical vessel is lowered down on temporary cribbing supports or saddles that were not design to support it. This can happen for instance when removing transportation saddles and temporarily resting the shell on cribbing or down ending a vertical vessel or stack and placing it in temporary saddles. The formula at the bottom of the following sheet shows how to calculate the radius of a temporary saddle, but it does not tell if there is enough saddle width or side support to prevent buckling. A Rigging Engineer or Vessels Engineer should be consulted if there is any doubt.

FORMULA FOR FINDING THE TOTAL SLING LENGTH "L" REQUIRED TO SLING A HORIZONTAL VESSEL IN A BASKET HITCH CONFIGURATION.



L = 7.7 R

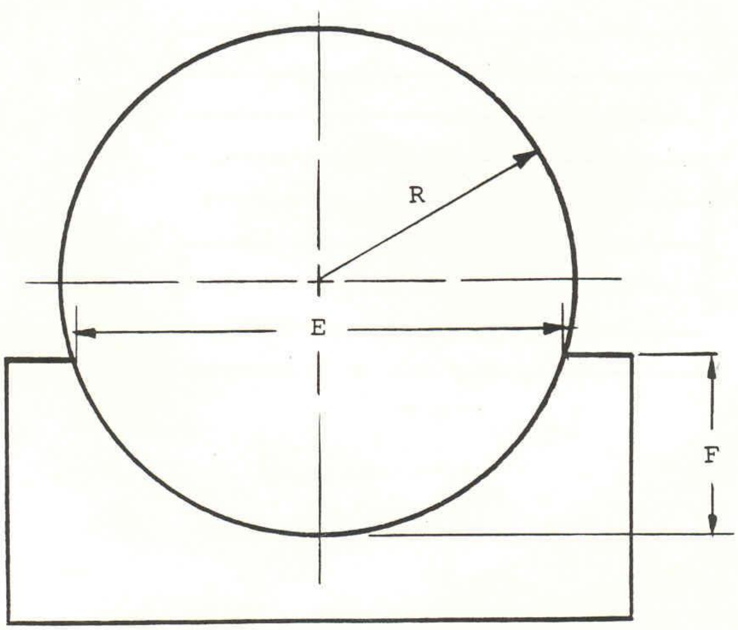
R = RADIUS

h = DISTANCE FROM THE CENTER OF THE VESSEL I.D. TO THE HOOK

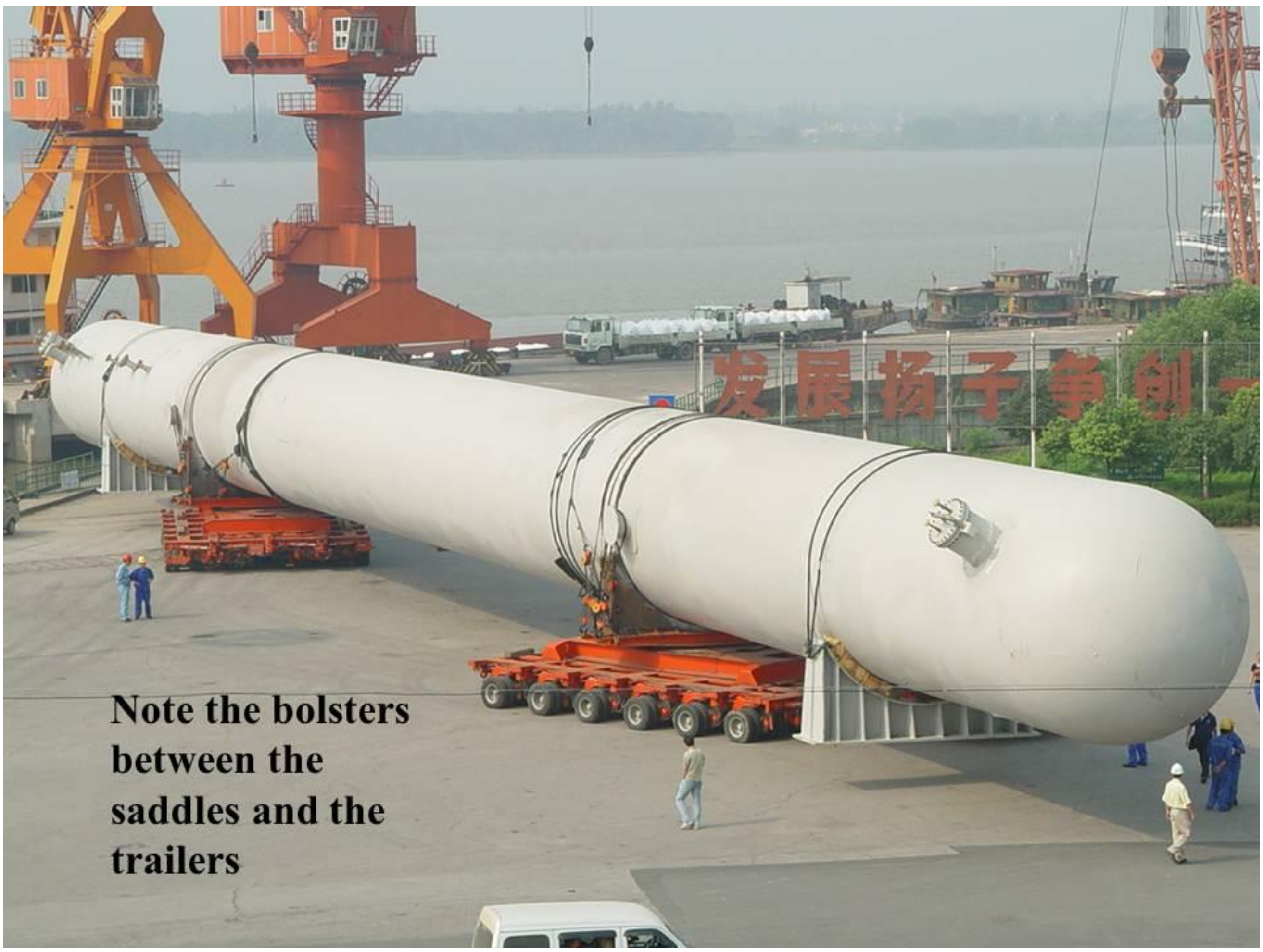
= 2 R

FORMULA FOR SELECTING A SADDLE TO FIT A VESSEL WITH RADIUS "R". MEASURE "E" AND "F" AND SOLVE FOR "R".

$$R = \frac{E^2 + 4F^2}{8F}$$



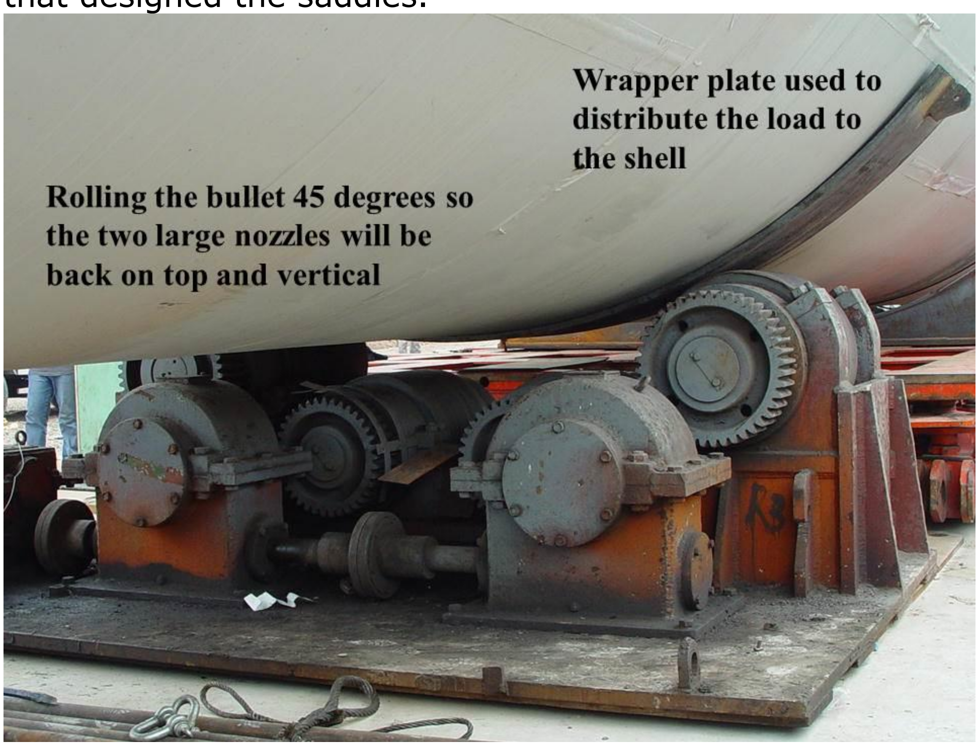
Slide show number 3 on my website shows offloading and transporting a 390 Te horizontal vessel called a bullet through a refinery in Nanjing, China. Note that the bullet is being transported on temporary transportation saddles designed by the Fluor Vessels Department because a very low profile was required to move the bullets under two 6 m pipe bridges. The depth of the saddles at the middle was 150 mm. The permanent saddles were designed by the fabricator and can be seen suspended by cables at each end of the bullet. We had the Fluor Vessels Department design the temporary transportation saddles because the fabricator did not have the expertise to design them, nor did he want the responsibility in case buckling occurred. We decided to assume the responsibility because moving the five bullets through the plant was much more economically feasible than developing a temporary haul road to go around the site, which would have included constructing a ro-ro ramp for offloading and shoring up under a high bridge with water under it.



Transportation Of a 390 Te Bullet On Low Profile Saddles

View Showing A Low Profile Transportation Saddle

After going under the two pipe bridges, the bullet had to be rotated about 45 degrees to locate the large nozzles back on top. The nozzles were set at 45 degrees off center for transportation clearances. Notice that large rolls were used to roll the bullet and that a 25 mm wrapper plate was placed between the rolls and the bullet shell to distribute the load to prevent buckling. This wrapper plate was required by the Fluor Vessel Engineer that designed the saddles.



View Showing The Rollers And Wrapper Plate

Dennis Moss, who is the head of the Fluor Vessels Department in Aliso Viejo, CA, was the designer of the low profile transportation saddles used above. For more information on buckling stresses and saddle design, you can purchase his book "**Pressure Vessel Design Manual**"at Amazon or other locations. Below is a link to some information of his on saddle design:

http://www.engineering-software.com/prDemos/SadDemo.pdf