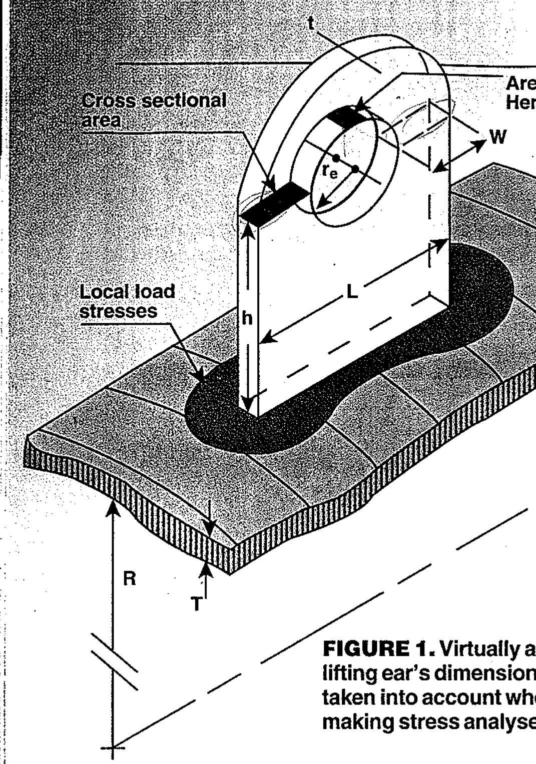

FIGURE 1. Virtually all of a lifting ear's dimensions must be taken into account when making stress analyses

YOU CAN'T TRUST YOUR (LIFT) EARS

$t = \left(\frac{.418}{\sigma}\right)^2 \frac{EF(r_e - r_s)}{r_e r_s}$

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Reactors, tanks, columns and other process vessels may support a variety of structural loads. Maintenance or inspection personnel may have to ascend to platforms attached to vessel walls; insulation may have to be suspended in place; the vessel itself occasionally must be hoisted.

If such loads are not too large, lifting ears (also known as plate clips or eyelets) fastened to the vessel wall can serve to transmit the lifting force. If the load seems large, however, it is prudent to calculate the strength of the ear, as well as the local load stresses in the shell to which the ear is welded. (We assume here that the weld between ear and shell is adequate, i.e., butt welded, and thus requires no further checking.)

Three kinds of stresses

The three critical stresses for a strength assessment of lifting ears are (Figure 1):

- The Hertzian contact stress or surface stress due to the load transfer from the hoist line's shackle pin into the eye of the lifting ear (for discussion of these stresses, see Reference 1 or 2)
- The stress arising in the smallest cross-sectional areas next to the eye
- The local load stresses in the vessel wall adjacent to the ear

The Hertzian contact stress is given by:

$$\sigma = \sqrt{\frac{E(F/t)}{2\pi(1-\nu^2)r_e\, r_s/(r_e - r_s)}} = 0.418\sqrt{\frac{E(F/t)}{r_e\, r_s/(r_e - r_s)}} \qquad (1)$$

where the simpler expression assumes that ν equals 0.30, its typical value. Note that F/t is the load of the cylinder-cylinder contact between eye and shackle pin. In the absence of data, this contact stress is best limited to twice the yield stress of the ear material.*

The stresses in the smallest cross sectional areas next to the eye can be determined by considering the curved portion of the ear around and above the eye as a beam fixed at these cross sectional areas. The normal stress (or membrane stress) is then calculated as

$$\sigma_{mem} = F/2wt \qquad (2)$$

and the bending stress as

$$\sigma_{bend} = 1.5[(r_e + 0.5w)/w^2t]F \qquad (3)$$

Both are primary stresses, so the limits are:

$$\sigma_{mem} \leq f \quad \leftarrow 2/3\,F_y \qquad (4)$$

and

$$\sigma_{mem} + \sigma_{bend} \leq 1.5f \quad \leftarrow F_y \qquad (5)$$

The stress intensity due to the local load stresses in the cylindrical vessel wall where the lifting ear is fastened is

$$\sigma = SCF[(RT)^{0.5}/T^2][line\ load] \qquad (6)$$

where *line load* is either P/L (thrust load) or $6M/L^2$ (moment load), and the value of *SCF* depends on whether the ear is longitudinal or circumferential (Figure 2) and whether the load is a thrust load or a moment load:

$$SCF = 1.55(R/T)^{0.38}(L/R + 0.125) \qquad (7)$$

for thrusts on longitudinal ears;

$$SCF = 0.23(R/T)^{0.62}(L/R) \qquad (8)$$

for moments on longitudinal ears;

$$SCF = 1.50(R/T)^{0.38}(L/R + 0.3) \qquad (9)$$

for thrusts on circumferential ears; and

$$SCF = 0.18(R/T)^{0.74}(L/R) \qquad (10)$$

for moments on circumferential ears.

These *SCF* equations are valid within the ranges $40 < (R/T) < 250$ together with $0.04 < (L/R) < 0.6$. They apply to cylindrical shells only.

If a thrust load and a moment load act simultaneously on the ear, sum the two stresses. Regardless of internal pressure in the vessel, the stress limit for the sum is twice the basic design stress of the vessel material:

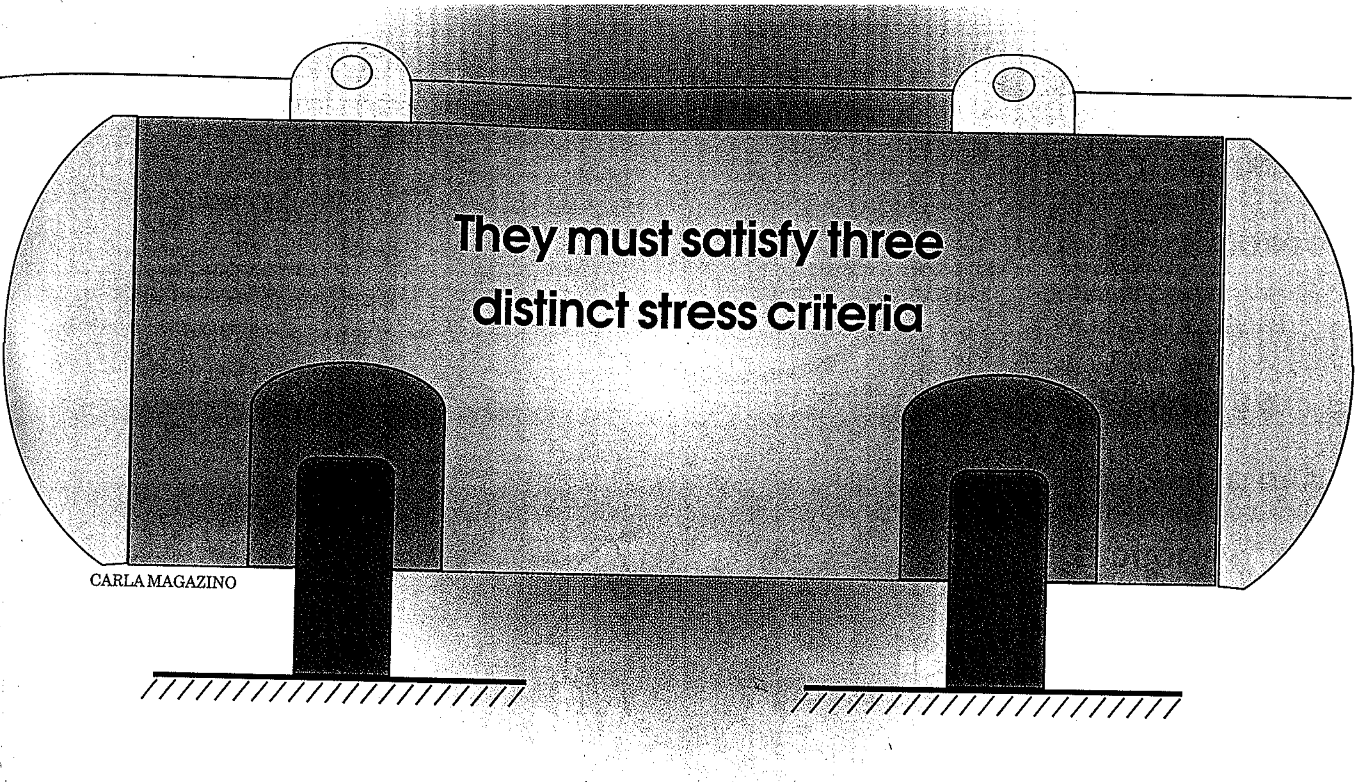
$$\sigma_{thrust} + \sigma_{moment} \leq 2f \quad \leftarrow 4/3\,F_y \qquad (11)$$

In practice, loads that result only in shear stress can be neglected safely.

And keep in mind...

Many plant owners require an additional safety margin of 2 for hoisting

*The only other indication of which the author is aware is a now-superseded German standard, DIN 1050 (June 1968), which stated 650 MPa as the contact stress limit for a particular grade of carbon steel St A. To all intents and purposes, St 37 is equivalent to the alloy designated in the U.S. as ASTM's A-283 Grade C.



calculations. In such cases, double the load on the lifting ear for the Hertzian contact stresses and for the stresses in the smallest cross sections of the clip.

Doubling is not required, however, for the local load stresses in the cylindrical wall. The aforementioned allowable stress of $2f$ assumes the presence of the maximum allowable internal pressure, and the absence of pressure during hoisting provides enough scope for the extra safety margin.

If the local load stresses in the cylindrical shell prove to be too high, consider using a reinforcing pad (Figure 3; also known as a doublings plate). To calculate the stress intensity at the ear, enter the combined thickness of shell and pad for T in the above equations.

For circular reinforcing pads, the stress intensity in the cylindrical shell next to the reinforcement can be assessed by considering the (circular) reinforcing plate as a nozzle. A convenient technique for assessing its stress intensity is the "Improved Shrink Ring Method" described in Appendix 2 of Reference 5. In this connection also, use a stress limit of $2f$ to check whether a given proposed reinforcing pad is large enough.

If rectangular reinforcing pads are to be used, the stress intensity in the shell next to the reinforcement can be assessed by considering the reinforcement plate as a loaded rectangular attachment. For making the actual stress intensity calculation, see Reference 6.

If lifting ears prove to be inappropriate for a given task, the engineer should instead consider the use of lifting trunnions. If they are used, the aforementioned shrink-ring method [5; Appendix 2] is convenient for assessing the local load stresses in the shell (with or without doublings plate) due to the trunnion.

NOMENCLATURE

- E Young's modulus
- F Force (load) on lifting ear
- f Basic design stress (two-thirds of the yield stress)
- h Height of center of lifting ear's eye
- L Length of ear
- M Moment load of lifting ear on cylindrical shell
- P Thrust force (radial with respect to cylindrical vessel)
- R Radius of cylindrical shell
- r_e Radius of lifting ear's eye (hole)
- r_s Radius of shackle pin
- T Thickness of cylindrical shell
- t Thickness of lifting ear
- w Width of area adjacent to eye (equal to $[0.5L - r_e]$)
- ν Poisson's ratio

Sample calculation

Problem: A horizontal vessel with an outside diameter of 1,400 mm, wall thickness of 6 mm, length of 2,500 mm and a total weight of 950 kg (equivalent to 9,500 N), including nozzles and saddles, is to be lifted by means of two 6-mm-thick longitudinal lifting ears, with dimensions as shown in Figure 4.

Other relevant data are as follows: yield stress for vessel shell and ear is 250 N/mm^2; Young's modulus is 192,000 N/mm^2; diameter of shackle pins is 38 mm; the cables attached to the ears make an angle of 30 deg with the vertical; the plant owner requires a safety margin of 2 for hoisting operations; the basic design stress is taken as two-thirds of the yield stress (i.e., (2/3)(250), or 166.7 N/mm^2). Are these lifting ears strong enough?

Solution: With two lifting ears and a 30-deg angle between cables and the vertical, the eye in each lifting ear is subjected to a force of [(9,500/2)/cos 30 deg], i.e., 5,485 N. The Hertzian contact stress is accordingly to be calculated

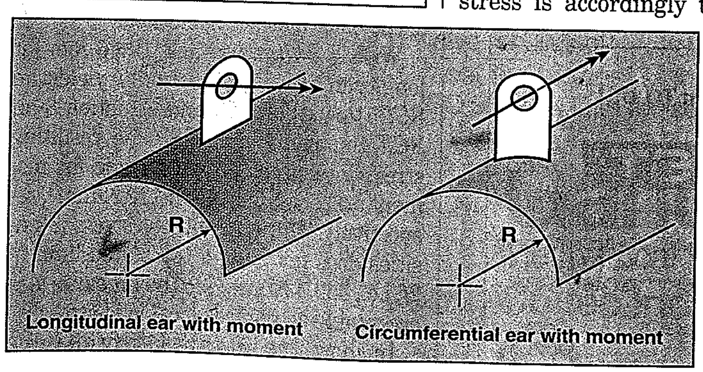


FIGURE 2. Longitudinal lifting ears are parallel to the vessel axis; circumferential ears are perpendicular to it

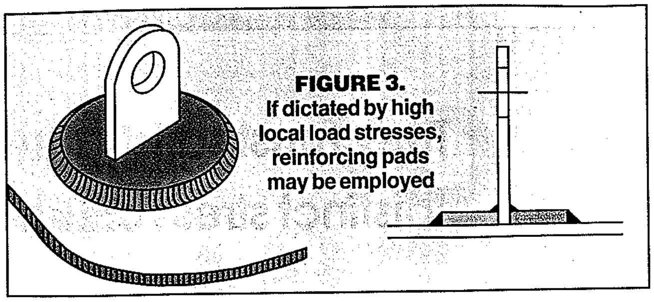
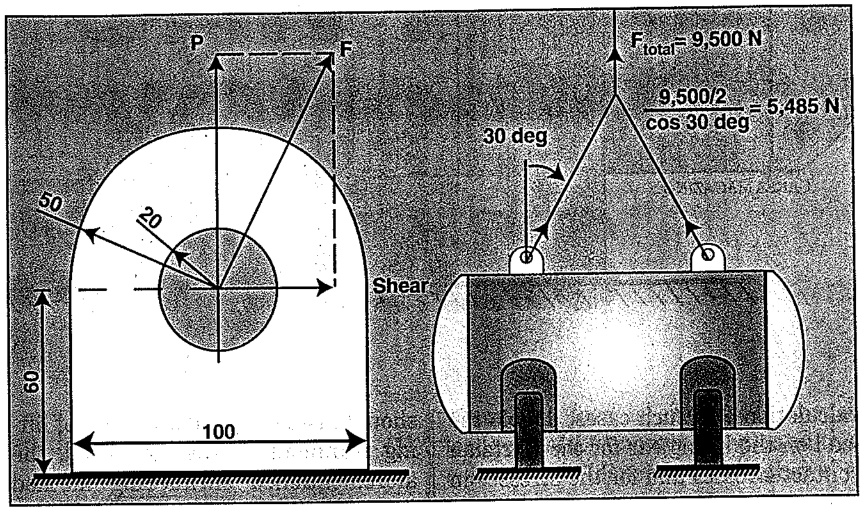


FIGURE 3. If dictated by high local load stresses, reinforcing pads may be employed

FIGURE 4 (below). Lifting ear and vessel for the numerical example in the text are shown here not to scale



with a force of (2)(5,485), or 10,970 N, to accommodate the safety factor of 2.

From Equation 1, the resulting contact stress is 402 N/mm^2. This value is permissible, as it is less than twice the yield stress.

From Equations 2 and 3, the stresses in the minimum cross section, also calculated with F at 10,970 N, are as follows:

Normal or membrane stress:

σ_{mem} = 10,970/[2(50 − 20)(6)] = 30.5 N/mm^2

Bending stress:

σ_{bend}= 1.5{[20 + (0.5)(30)]/[(30)^2(6)]} ×{10,970} =106.7 N/mm^2

These stresses satisfy the limits expressed by inequalities 4 and 5.

For the calculation of the stress intensity in the vessel's wall, the loads are (with no application of the safety factor of two): P (thrust) = 4,750 N; M (moment) = 164.5 × 10^3 N-mm, calculated (with the calculation not shown here) as the height of the ear times the shear load).

With R = 700 mm and T = 6 mm for longitudinal ears, Equation 7 yields an *SCF* (thrust) of 2.53 and Equation 8 an *SCF* (moment) of 0.628. The resulting stress intensities for thrust and for moment are 217 N/mm^2 and 112 N/mm^2, respectively. The sum of these stress intensities does not exceed the stated limit of $2f$, i.e., 333.3 N/mm^2, so the lift ear is also acceptable as regards local load stresses.

In short, the lift ears satisfy all three requirements. They are, accordingly, suitable for the hoisting operation. ■

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