ENGINEERING DESIGN

A GLOBAL VIEW OF DUPONT POLYMERS

Pressure Vessel Design for Engineering Plastics

HOOP STRESS MAY NOT BE ALL YOU NEED TO KNOW—FINITE ELEMENT ANALYSIS DEMONSTRATES IMPORTANCE OF THE TYPE OF END CLOSURE SPECIFIED

By Peter A. Tuschak

Engineering plastics have been used for pressure vessel applications for a long time. Lighter bodies, ballcock valves, and spray paint containers (Figure 1) are just a few examples of successful developments in this area.

Significant advances in resin development are now creating still further opportunities. Plastics of increasing stiffness and toughness have been commercialized. New amorphous plastics, suitable for blow molding sizable parts, have also been developed, opening the way to one step manufacturing of these products. The above activity creates new opportunities for pressure vessels in engineering plastics.

This article describes the results of studies made by DuPont on stress distributions in plastic pressure vessels and of the codes and standards that regulate their design.

Many designers use nominal hoop stress formulas —

$S = PR/t$ for thin walled vessels, or

$$S = P \times \frac{R_o^2 + R_i^2}{R_o^2 - R_i^2} \text{ for thick walled vessels}$$

as a basis for vessel design. We studied a typical pressure vessel, Figure 2, using finite element stress analysis on a digital computer and found that, due to geometrical discontinuities, stresses as high as three to six times the nominal hoop stress can occur. The results of a typical case are shown in Figure 3. The section where the cylindrical vessel and the torispherical head join is magnified, and lines of constant stress (stress contours) through the section are shown. The maximum stress is approximately six times the nominal hoop stress and it occurs on the inside surface of the knuckle radius.

Figure 1. Pressure vessels such as these have long been produced in engineering plastics. Stiffer and tougher resins now open up even more opportunities for larger, more critical applications.

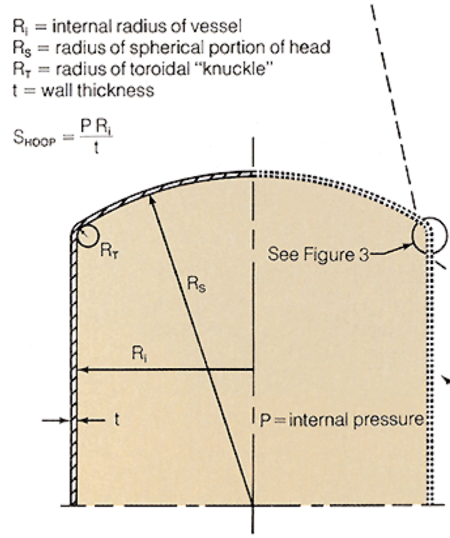

Figure 2. Cross section of pressure vessel analyzed using the ansys finite element program.

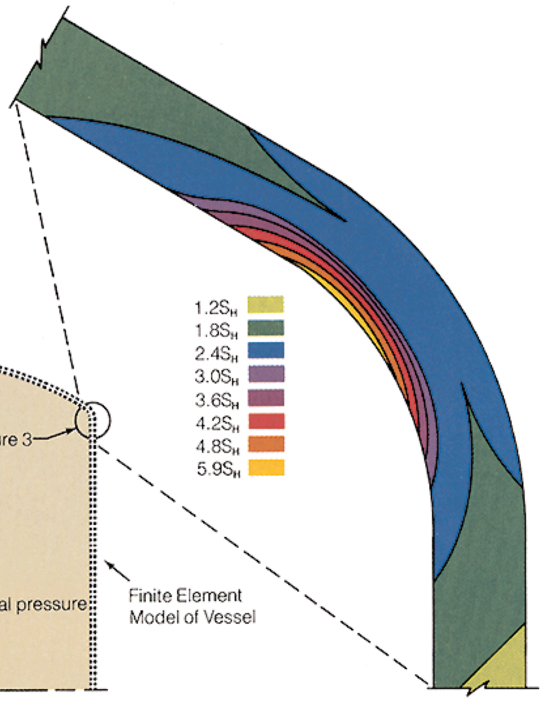

Figure 3. Stress distribution knuckle section of typical pressure vessel S_H = nominal hoop stress

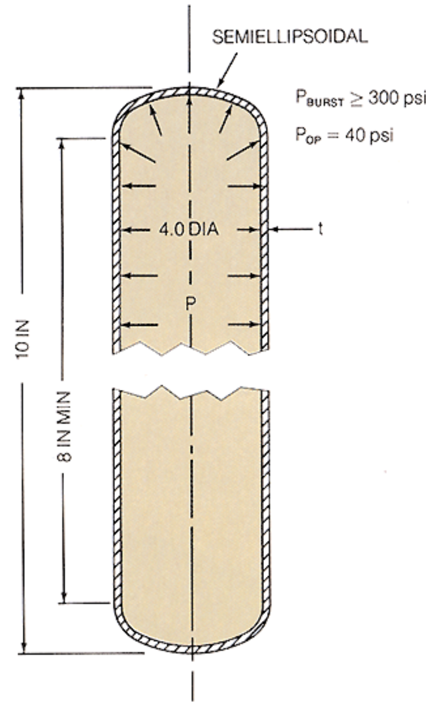

Figure 4. Dimensions of sample pressure vessel of Delrin® 500

Generous Safety Factors Required

Some of the industrial codes and standards that regulate pressure vessel design recognize the existence of stress concentrations and require the use of generous safety factors. The American Society of Mechanical Engineers, as an example, recommends that operating pressure in plastic vessels be no more than one sixth of the burst pressure, as determined by hydrostatic testing. The Technical Services Laboratory (TSL) design group established certain guidelines for pressure vessel design based in part on the ASME and other industrial codes.

Consider a pressure vessel with requirements as follows: The internal diameter of the vessel to be 102 mm (4 0 in), the overall length not to exceed 254 mm (10 in), with the cylindrical portion to be at least 203 mm (8 in) long, as in Figure 4. The material must be FDA approved. The operating pressure will be 276 kPa (40 psi), intermittently applied, with burst pressure at least 2068 kPa (300 psi). The operating temperature to be 23°C (73°F) at 50 percent relative humidity (RH). The first step in the design procedure is to determine a tentative wall thickness: Using the nominal hoop stress formula we can write

$$S_H = \frac{P\,R}{t}$$

Based on our finite element study and other relevant information, we can assume that the maximum stress will be

$$S_{max} = KS_H$$

where K is the stress concentration factor.

The magnitude of K depends on the type of end closure we select. We saw in our finite element example that for a torispherical head, K=6. Hemispherical and semiellipsoidal heads produce lower stress concentration factors. Since the dimensional requirements preclude a hemispherical closure—it would allow only a 152 mm (6 in) straight section—we'll select a 2:1 semi-ellipsoidal head. With this choice the stress concentration factor becomes four to one, i.e.

$$S_{max} = 4 \times S_H$$

Now,

$$t = \frac{4 \times P \times R}{S_{max}}$$

Because of the need for an FDA approved material we'll choose Delrin® 500 acetal resin, for which

$$S_{max} = 68948 \text{ kPa } (10{,}000 \text{ psi})$$

at 23°C (73°F). Thus,

$$t = \frac{4 \times 2068 \times 50.8}{68948} = 6.1 \text{ mm, or}$$

$$t = \frac{4 \times 300 \times 2}{10{,}000} = 0.24 \text{ in.}$$

with a wall thickness of 6.1 mm (0.24 in), the predicted burst pressure is 2068 kPa (300 psi) as required. Using the ASME guidelines adopted by TSL, we will now postulate the operating pressure to be less than one sixth of the burst pressure, i.e.

$$P_{op} \leq \frac{P_{BURST}}{6} = 345 \text{ kPa } (50 \text{ psi})$$

Since the required operating pressure is 276 kPa (40 psi), the design meets the requirements. The maximum stress at the operating pressure becomes

$$S_{(max.\ op)} = \frac{PR}{t} \times K = \frac{276 \times 50.8 \times 4}{6.1} = 9191 \text{ kPa, or}$$

$$\frac{40 \times 2 \times 4}{0.24} = 1333 \text{ psi}$$

and the safety factor is now

$$\text{s.f.} = \frac{S_{(max)}}{S_{(max\ op)}} \quad \begin{matrix} = \dfrac{68948}{9191} \\ = \dfrac{10{,}000}{1333} \end{matrix} \quad = 7.5$$

The above data were obtained by analysis only. The next, and perhaps most important step in the design is to mold a prototype vessel. Analysis does not take into account weld lines, gate size, effect of gate locations, etc.; thus, actual testing must be performed to establish the burst pressure. In the present case, since the vessel must survive a large number of cycles, it will undergo fatigue testing first. This consists of cycling the pressure from zero to 276 kPa (40 psi), 100,000 times at 23°C (73°F) and 50 percent RH.

The cycling test will be followed by the hydrostatic burst test. The same specimen that was cycle tested will be burst. If the burst pressure is equal to or greater than 2068 kPa (300 psi), our job is finished. On the other hand, if the burst pressure is less than 2068 kPa, we must modify the design, or lower the required operating pressure. If a design modification is made, a new prototype will be tested to make sure the desired performance is achieved. Therefore, pressure vessel design is an iterative procedure and the designer should remember that final testing of the product is the ultimate proof of performance.

While this article presents a rational approach to pressure vessel design with engineering plastics, it has not been possible to cover all aspects of vessel design. For example, long term pressurization might require considerations of material creep in determining maximum allowable stress. Also, the question of pressure relief devices was not addressed. These, and other considerations are continually under study by the TSL design group and the lessons learned are available for application to customer problems on a continuing basis.

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