From "Pressure Vessels Design and Practice – Somnath Chattopadhyay"

2.4 Design by rule

By following design-by-rule methods, the designer simply follows the rules laid out in the procedures for components such as nozzles, heads, flanges, and so on. This procedure has the great advantage of simplicity and consistency but has several limitations. For example, there are cases when the loadings and geometries are such that the procedure cannot be applied effectively. Some of the rules are based on elastic stress analysis with some limitations on maximum stress. Some are based on shakedown concepts without specifically considering stress ranges, while others are based on limit load concepts with suitable shape factors. Design-by-rule methods were used in earlier ASME design codes (Sections I and VIII).

Generally speaking, design-by-rule methods of design are based on experience and tests. This process requires the determination of design loads, the choice of a design formula and the selection of an appropriate stress allowable for the material used. The procedure provides the information on required vessel wall thickness as well as the rules of fabrication and details of construction. These rules do not typically address thermal stresses and fatigue. The fatigue issues are considered covered by the factors of safety.

From "Process Equipment Design – Brownell & Young"

13.7 DESIGN OF CYLINDRICAL SHELLS UNDER INTERNAL PRESSURE

The equations for determining the thickness of cylindrical shells of vessels under internal pressure are based upon a modified membrane-theory equation. The development of this equation is described in the following chapter (see Eq. 14.34). The modification empirically shifts the thin-wall equation (see Eq. 3.14) to approximate the "Lamé" equation for thick-walled vessels (see Fig. 14.5). The equation may be written in either of the following forms:

$$t = \frac{pr_i}{fE - 0.6p} = \frac{pr_0}{fE + 0.4p} \qquad (13.1)$$

or

$$p = \frac{fEt}{r_i + 0.6t} = \frac{fEt}{r_0 - 0.4t} \qquad (13.2)$$

where t = minimum required thickness of the shell exclusive of corrosion allowance, inches
p = design pressure, or maximum allowable working pressure, pounds per square inch
E = welded-joint efficiency (see Table 13.2)
f = maximum allowable stress, pounds per square inch (see Table 13.1 or Appendix D).
r_i = inside radius of the shell, inches
r_0 = outside radius of the shell, inches

If the thickness of the shell exceeds 50% of the inside radius, or when the pressure exceeds $0.385fE$, the Lamé equation should be used to calculate the vessel-shell thickness (see Chapter 14). The following forms of the Lamé equation are given by the code (11).

With the pressure p known,

$$t = r_i(Z^{1/2} - 1) = r_0\left(\frac{Z^{1/2} - 1}{Z^{1/2}}\right) \tag{13.3}$$

where

$$Z = \frac{fE + p}{fE - p} \tag{13.4}$$

When t is known,

$$p = fE\left(\frac{Z - 1}{Z + 1}\right) \tag{13.5}$$

where

$$Z = \left(\frac{r_0 + t}{r_0}\right)^2 = \left(\frac{r_0}{r_i}\right)^2 = \left(\frac{r_0}{r_0 - t}\right)^2 \tag{13.6}$$