In order that each ring have the same shear stress under static pressure, Berman finds that the same relation, Equation (33) (first found by Manning[5]), applies for the controlled fluid-fill container that also applies for the multi-ring container designed for static shear strength. If this result is used in a shear fatigue analysis (assuming ductile materials), then Equation (33) can be interpreted as the maximum shear stress developed during a cycle of pressure, i.e.,

$$(S)_{max} = \frac{p}{N} \frac{K^{2/N}}{(K^{2/N}-1)} \tag{73}$$

If the pressures p_n are reduced to zero, then the minimum shear stress during a cycle of pressure is zero. Therefore, the semirange and mean shear stresses are equal,

$$S_m = S_r = \frac{pK^{2/N}}{2N(K^{2/N}-1)} \tag{74a, b}$$

where S_m and S_r are defined in Equations (8a, b).

If Equation (74a, b) are substituted into the fatigue relation, Equation (12), there results

$$\sigma = \frac{5p}{2N} \frac{K^{2/N}}{(K^{2/N}-1)} \tag{75}$$

It is surprising that this result, Equation (75), is the same as Equation (43) plotted in Figure 11, the result of the shrink-fit analysis, except now the limit Equation (45) no longer applies. Therefore, now p/σ can be made as large as desired simply by increasing N. The only problem is that the required N or K may be too large to be practical. For example, assume σ = 150,000 psi (ultimate strength of a ductile steel), N = 8 and K = 16. Calculating p we find that p = 240,000 psi. Thus, it is concluded that for fatigue applications under high pressure the controlled-fluid-fill multi-ring container becomes too large to be practical. Eight rings also means there are seven annuli under fluctuating pressures. (The magnitudes of these pressures are all different and are given by an equation similar to Equation (41).) Design of mechanical apparatus to supply and control all these pressures presents practical difficulties also.

DESIGN REQUIREMENTS AND LIMITATIONS FOR HIGH-PRESSURE CONTAINERS

As already indicated, the theoretically predicted maximum pressure capability for the four containers considered in detail in the present study are as follows for 10^4 to 10^5 cycles life:

Container	Maximum Pressure, p, psi
Multi-ring	300,000
Ring-segment	300,000
Ring-fluid-segment ($p_3/p = 0.3$)	~1,000,000
Pin-segment	210,000

These predictions are based on an ultimate tensile strength of 300,000 psi for the liner and 200,000 psi for the outer cylinders or components, and apply to any operating temperature provided these are the strengths at temperature.

For liners with ultimate tensile strengths much greater than 300,000 psi, the theoretical maximum pressure capability of the various designs may be improved appreciably. This is true if it can be assumed that the higher strength materials would exhibit the same fatigue behavior as that shown in Figure 9 for steels with ultimate tensile strength ranging from 250,000-310,000 psi at room temperature. (Tensile strengths of 410,000 psi have been reported for AISI M50 steel. If the previous assumption is correct, then a multi-ring or ring-segment container with an M50 liner would have a theoretical maximum pressure capability of 410,000 psi. However, these containers may require that some of ductile outer cylinders have ultimate tensile strengths greater than 200,000 psi.)

Possible Manufacturing and Assembling Limitations

It is important to note that the theoretical pressures given in the above tabulation may not be achievable for each design because of practicable design limitations. For example, the outside diameters required for designs having 6- and 15-inch bore diameters are as follows:

Container	Outside Diameter, inches	
	6-inch Bore Design	15-inch Bore Design
Multi-ring	51.0	127.5
Ring-segment	60.0	150.0
Ring-fluid-segment	229.5	573.5
Pin-segment	90.4	180.2

It may be impossible to obtain steel cylinders in such large sizes (10- to 50-foot diameters) with ultimate strengths of 200,000 psi, and it may be impossible to machine and transport these large cylinders. Also heat treatment of heavy sections may be a problem. This may not be the case for pin-segment container, however. In this instance, it may be possible to forge the large steel pins (18.2 inches and 45.4 inches in diameter respectively, based on a design shear stress of 50,000 psi in fatigue for the pins) and the segments (thick plates). This indicates an advantage of the pin-segment design for vessels with $p \leqq 210,000$ psi.

A pin-segment arrangement may also be used to advantage as a replacement for the outer cylinder in the other container designs. This would help overcome the difficulties associated with the large steel cylinders. A wire wrap or strip wrap could also be used to this advantage as a replacement to outer cylinders.