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 3 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	Outside Diameter, inches: 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 these large cylinders. 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.

The limitations in some of the designs due to large-diameter outer cylinders may be partially overcome by using the autofrettage process to provide some additional prestress at the liner bore. The process introduces compressive prestresses by plastic deformation of the bore. This approach could reduce the size and number of outer rings that otherwise would be needed to achieve the total prestress by shrink fitting alone. In fact, the autofrettage process could be used to improve the size efficiency of all the design concepts considered. However, if autofrettaging is employed, then high-strength steels with appreciable amounts of ductility should be selected for the liner because the process requires plastic deformation of the bore.

In addition to the potential problem of cylinder size, the theoretical pressures may not be possible to achieve because excessive interferences may be required for shrink-fit assembly. The maximum interferences required for the designs with the above theoretical pressures are as follows:

Container	Maximum Interference Required, inch/inch
Multi-ring	$\Delta_1/r_1 = 0.0036$
Ring-segment ($k_2 = 1.1$, $\frac{E_2}{E_1} = 3.0$)	$\Delta_{12}/r_1 = 0.0028$
Ring-fluid-segment ($k_2 = 2.0$)	$\Delta_{12}/r_1 = 0.0164$
Pin-segment	None, except for a small amount to take up slack during assembly

For the multi-ring container, the interference required between the liner and cylinder 2 as manufactured is $\Delta_1/r_1 = 0.0036$ in./in. This is a reasonable value and corresponds to a temperature difference of 400 to 500 F for assembly. However, the interference as manufactured is not always the same as the interference as assembled. Suppose that the multi-ring container is assembled ring by ring from the inside out. Each ring expands as it is shrunk on and the assembly interference progressively increases beyond the manufactured interference. Formulas for the assembly interference can also be derived. Derivations will be given in the subsequent report.

The interference required for the ring-fluid-segment container is $\Delta_{12}/r_1 = 0.0164$ in./in. This interference requirement is severe, if not impossible, especially when one considers assembling not only the liner and cylinder 3, but also a number of segments all at the same time. (Δ_{12} is the interference required between the liner, segments, and cylinder 3. Δ_{12} is also the assembly interference as well as the manufactured interference since the liner, cylinder 3, and the segments must be assembled simultaneously.) The large magnitude for Δ_{12} is primarily due to large radial elastic deformation of the segments under pressure. This is a distinct disadvantage for the containers having segments in their designs. Another potential disadvantage of these designs is the possible problem of gouging the liner with the corners of the segments if the components are assembled by pressing.

The severe interference requirements imposed by the segments are reduced if the segment size (k_2) is reduced and if a higher modulus material is used for the segments. These effects are shown above for the ring-segment container which has a lower interference requirement, i.e., $\Delta_{12} = 0.0028$ in./in.