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 such 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.

The limitations in some of the designs due to large-diameter outer cylinders may also 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 are as follows:

Container	Maximum Pressure, p, psi	Maximum Interference Required, inch/inch
Multiring	300,000	$\Delta_1/r_1 = 0.0036$
Ring-segment ($k_2 = 1.1$, $\frac{E_2}{E_1} = 3.0$)	300,000	$\Delta_{12}/r_1 = 0.0028$
Ring-fluid-segment ($k_2 = 2.0$)	450,000	$\Delta_{12}/r_1 = 0.0129$
Pin-segment	210,000	None, except for a small amount to take up slack during assembly
Ring-fluid-ring (Example 2)	450,000	$\Delta_1/r_1 = 0.0080$

For the multiring 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 it 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 multiring 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 are given in Appendix II.

The interference required for the ring-fluid-segment container is $\Delta_{12}/r_1 = 0.0129$ 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 shown as follows: from Equation (19a) it is found that

$$\frac{E_2 (u_1 - u_2)}{r_1 p_1} = 0.69 \text{ for } k_2 = 2 \text{ and } p_2 = p_1/k_2 \quad ,$$

where u_1 and u_2 are the radial displacements of the segment and r_1 and r_2, respectively. From a computer calculation for the ring-fluid-segment container, p_1 at pressure ($\sigma_r = -p_1$ at r_1), is found to be $p_1/\sigma_1 = 2.2$. Thus,

$$\frac{E_2 (u_1 - u_2)}{r_1 \sigma_1} = 2.2\,(0.69) = 1.518 \quad .$$

For $p/\sigma_1 = 2.87$ and $p = 450,000$ psi, $\sigma_1 = 157,000$ psi. Hence, $\dfrac{u_1 - u_2}{r_1} = 0.00795$ in./in.

for $\sigma_1 = 157,000$ psi and $E_2 = 30 \times 10^6$ psi, and it is evident that large interference, $\Delta_{12} = 0.0129$ in./in., is required to overcome large deformation of the segments under pressure. This is a 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. A further factor that must be considered in the design of segments is bending deformation. This is discussed in Appendix I.

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 that has a lower interference requirement; i.e., $\Delta_{12} = 0.0028$ in./in. However, selection of a high modulus material must be done with care because tensile stresses do develop in the segments as shown in Appendix I and many high-modulus materials have low tensile strengths.

Thus, it is seen that some <u>theoretical</u> container designs for high pressure may be impossible to fabricate because of the large outside diameters and interferences required. In order to obtain a more realistic evaluation of the various design concepts, predictions of pressure capability are made for more <u>practicable</u> design requirements, i.e., outside diameters limited to 72 inches and the interferences limited to 0.007 in./in. maximum. These predictions are as follows for 10^4-10^5 cycles life: