Thus, it is seen that some theoretical 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 practicable 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:

Container	Bore Diameter, inches	Outside Diameter, inches	Number of Components, N	Maximum Pressure, p, psi
Multi-ring	6	51.0	5	300,000
	15	72.0	7	275,000
Ring-segment ($k_2 = 1.1$, $E_2/E_1 = 3.0$)	6	60.0	6	290,000
	15	72.0	8	265,000
Ring-fluid-segment ($p_3/p = 0.3$)	6	72.0	10	286,000
($k_2 = 2.0$) ($p_3/p = 0.2$)	15	72.0	5	118,000
Pin-segment	6	72.0	3	195,000
	15	(a)	--	--

(a) OD $\leq$ 72.0 not possible for 10^4-10^5 cycles life and $\alpha_r = \alpha_m = 0.35$.

It is evident that lower maximum pressures are now predicted, particularly for the 15 inch bore designs. The reduction in pressure capability is due only to the restriction in outside diameter for the multi-ring, ring-segment, and pin-segment containers. However, both the outside diameter and interference limitations reduce the predicted pressure for the ring-fluid segment container. The reduction for this container is severe and is caused by three effects. The first is excessive deformation of the segments for $k_2 = 2.0$. The other effects are coupled; reducing the outside diameter while maintaining the design pressure increases the interference required, but limiting the interference causes a reduction in maximum pressure because the interference depends upon the pressure.

Residual Stress Limitations

A container designed for a specific cyclic pressure requires certain residual stresses (prestresses) at operating temperature. It is also important, however, to check the residual stresses at room temperature because of differences in thermal expansion.

Residual stresses are calculated for the multi-ring container as an example. The specific container design discussed here is the one considered in the foregoing section for a bore diameter of 6 inches. Calculations are performed for design applications at room temperature, 500 F, and 1000 F. The material data assumed are given in Table 1. The liner material is assumed to be 18 per cent Ni maraging steel, and the outer cylinders are assumed to be made of modified H-11 steel. The differences in thermal expansion for these materials are likely to be the largest expected among the steels that could be used.

TABLE 1. ELEVATED-TEMPERATURE DATA FOR 18% Ni MARAGING STEEL AND H-11 STEEL[a]

	70 F	500 F	1000 F
	Modulus of Elasticity, psi		
18% Ni Maraging	26.5×10^6	23.0×10^6	18.7×10^6
H-11	30.0×10^6	27.4×10^6	22.8×10^6
	Coefficient of Thermal Expansion, in./in.		
18% Ni Maraging	5.6×10^{-6}	5.6×10^{-6}	5.6×10^{-6}
H-11	7.12×10^{-6}	7.25×10^{-6}	7.37×10^{-6}

(a) Poisson's ratio taken as constant, $\nu = 0.3$, for both materials.

Results are given in Table 2. The range and mean stress parameters were $\alpha_r = 0.5$ and $\alpha_m = -0.5$, respectively. The results show that the excessive residual stresses at room temperature occur for the multi-ring container having a required prestress, $\sigma_\theta = -\sigma_1$ at 500 F and 1000 F; i.e., the residual stress $\sigma_\theta < -\sigma_1$ at room temperature, where σ_1 is the design stress and $\sigma_1 \leqq$ ultimate tensile strength. The reason for this is the larger interferences required for elevated-temperature application as shown in Table 2. Larger interferences are necessary for high-temperature applications because the outer rings expand more than the liner due to the differences in thermal expansions as shown in Table 1. On the other hand, reduction of the temperature from operating temperature to room temperature causes the outer rings to tend to contract more than the liner. The liner resists the contraction and the residual interface pressures are increased, thereby increasing the magnitude of the residual hoop stress at the bore.

If the multi-ring container is to be used at 500 F and 1000 F with the material properties given in Table 1, then the prestress requirement, $\sigma_\theta = -\sigma_1$ at temperature ($\alpha_m = -0.5$) has to be relaxed. Accordingly, calculations of residual stresses and interferences are rerun for $\alpha_m = -0.3$ (prestress $\sigma_\theta = -0.8\ \sigma_1$ at temperature). These results are shown in Table 3. With $\alpha_m = -0.3$, excessive residual stresses at room temperature are avoided for the 500 F design. However, for operation at 1000 F, $\alpha_m > -0.3$ is necessary since $\sigma_\theta < -\sigma_1$ at room temperature for the 1000 F design with $\alpha_m = -0.3$.

Decreasing the interference fit (from those in Table 2 to those in Table 3), in order to avoid excessive residual stresses at room temperature, increases $(\sigma_\theta)_{max}$ from 0 to positive values. As pointed out in the latter part of the Fatigue Criteria section, zero to small $(\sigma_\theta)_{max}$ is expected to be beneficial in preventing the detrimental effect of fluid pressure from entering voids in the material. Therefore, if excessive residual stresses are to be avoided in containers designed for high temperatures, and if $(\sigma_\theta)_{max}$ is to be kept small, then the thermal coefficients of expansion of the component parts of the container should be more closely matched than those of Table 1. Preferably the coefficient of thermal expansion should be larger for the liner than for the outer cylinders; this would cause a reduction rather than an increase in residual stresses upon decreasing the temperature from operating temperature to room temperature.