pressure container can be precompressed by shrink-fit assembly, an important factor in triaxial fatigue may be the prestress that can be initially provided. Therefore, for 10^4 to 10^5 cycles triaxial fatigue life, α_r and α_m are assumed to be

$$\alpha_r = 0.5, \ \alpha_m = -0.5 \tag{11}$$

as indicated in Figure 3. With $\alpha_m = -\alpha_r$ the maximum tensile stress at the bore would be zero.

In order to approximate a life of one cycle, it is assumed that

$$\alpha_r = 1.0, \ \alpha_m = 0, \ \text{for 1 cycle} \tag{12}$$

which represents a cycle between $\pm\sigma_u$, the ultimate strength.

NONDIMENSIONAL PARAMETER ANALYSIS AND PREDICTION OF MAXIMUM PRESSURES

The theoretical equations for each container were put into nondimensional form and programmed for computer solution. Nondimensional pressure-to-strength ratios were determined. Some of the results for the various designs are now presented.

Multi-Ring Container

The optimum design of a multi-ring container having all rings of the same ductile material and based upon the shear fatigue criterion is first considered. The results are plotted in Figure 4. The limit curve is for $S_m = 0$ in the innermost cylinder. The limiting value of p/σ for $K \rightarrow \infty$ is two-thirds; i.e., $p = 2/3\ \sigma$. (S_m is defined by Equation (5b) and σ by Equation (9).) If a ductile steel has an ultimate strength of 210,000 psi, then the maximum cycle pressure is 140,000 psi based upon the shear fatigue criterion.

These results for a ductile steel indicate that higher strength steels, at least for the liner, will have to be used in order to reach the high pressures desired. Accordingly, a multi-ring container with a high-strength liner is analyzed on the basis of the tensile fatigue criterion for the liner. The resulting pressure-to-strength ratios are plotted in Figure 5. (The parameter α_r and the stress σ_1 are defined in the fatigue relation, Equation (10).) For a lifetime of 10^4 to 10^5 cycles, $\alpha_r \approx 0.5$ from Figure 3. From the $\alpha_r = 0.5$ curve of Figure 5, a limiting value $p = \sigma_1$ is found. Hence, it is concluded that the maximum cyclic pressure in a multi-ring container for 10^4 to 10^5 cycles of fatigue life is 300,000 psi based on the tensile fatigue criterion for the liner (σ_1 is assumed to be 300,000 psi). This conclusion presupposes that the outer cylinders can also be designed to withstand this pressure and that sufficient precompression ($-0.5 \leqq \alpha_m \leqq 0$) can be provided. It can be shown that it is possible to meet these design requirements.

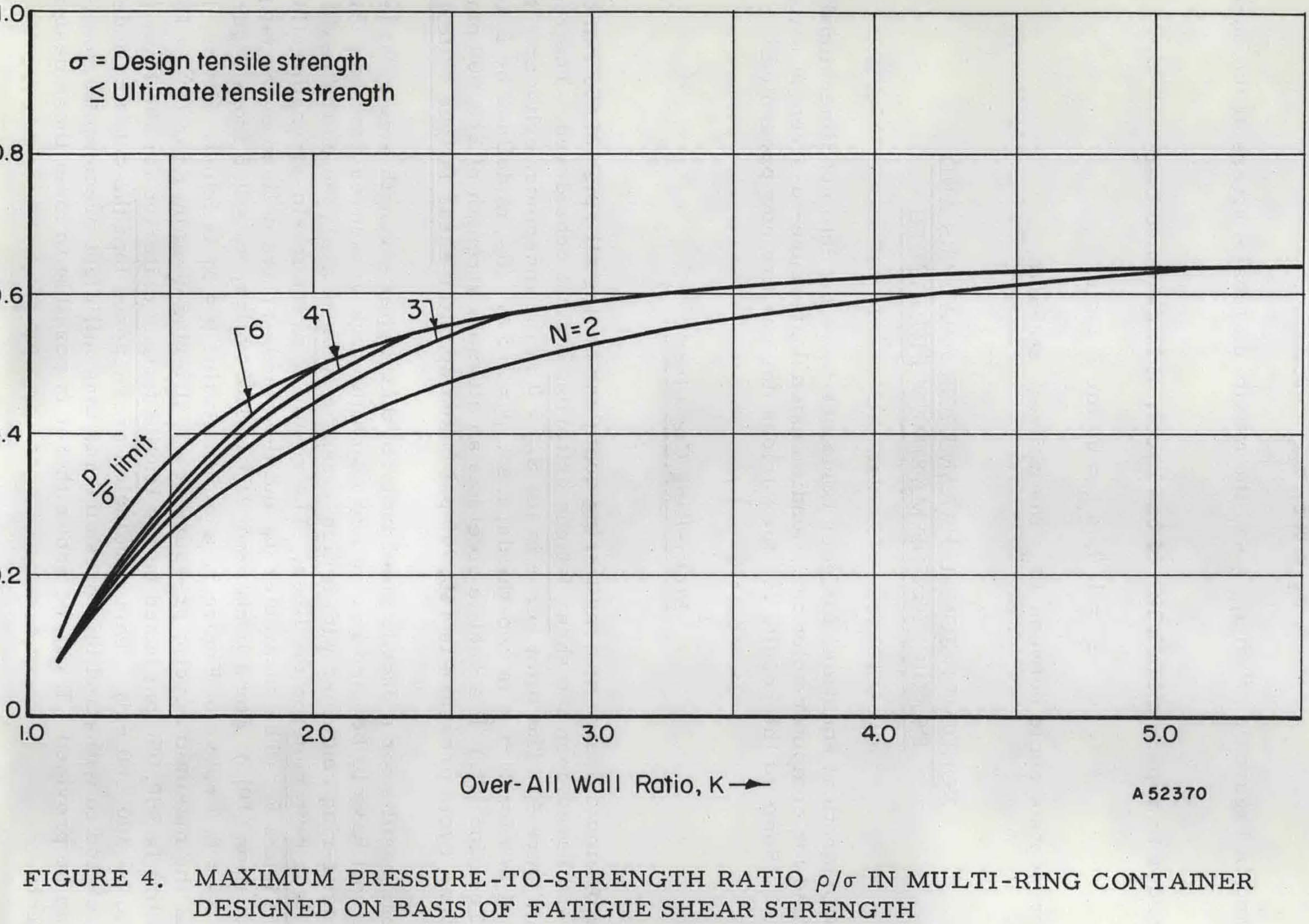

FIGURE 4. MAXIMUM PRESSURE-TO-STRENGTH RATIO p/σ IN MULTI-RING CONTAINER DESIGNED ON BASIS OF FATIGUE SHEAR STRENGTH