IR-8-198 (IV)

DEVELOPMENT OF THE MANUFACTURING CAPABILITIES OF THE HYDROSTATIC EXTRUSION PROCESS

INTERIM ENGINEERING PROGRESS REPORT

1 September 1965 – 30 November 1965

METALLURGICAL PROCESSING BRANCH
MANUFACTURING TECHNOLOGY DIVISION
AIR FORCE MATERIALS LABORATORY
RESEARCH AND TECHNOLOGY DIVISION
AIR FORCE SYSTEMS COMMAND
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

MM PROJECT NO. 8-198

(Prepared Under Contract No. AF 33(615)-1390 by
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NOTICES

FOREWORD

This Interim Engineering Progress Report covers the work performed under Contract No. AF 33(615)-1390 from 1 September 1965 to 30 November 1965. It is published for technical information only and does not necessarily represent the recommendations, conclusions, or approval of the Air Force.

This contract with Battelle Memorial Institute of Columbus, Ohio, was initiated under Manufacturing Methods Project No. 8-198, "Development of the Manufacturing Capabilities of the Hydrostatic Extrusion Process". It is being administered under the direction of Mr. Gerald A. Gegel of the Metallurgical Processing Branch (MATB), Manufacturing Technology Division, Air Force Materials Laboratory, Wright-Patterson Air Force Base, Ohio

The program is being conducted at Battelle by the Metalworking Research Division with Mr. R. J. Fiorentino, Associate Chief, as project engineer. Others contributing to the program are Mr. W. R. Hansen, Research Metallurgist, Mr. A. M. Sabroff, Associate Chief, and Mr. F. W. Boulger, Division Chief. Mr. R. L. Jentgen, Project Leader in the Experimental Physics Division, is assisting in the fluid and lubrication studies of the program. Mr. J. C. Gerdeen, Research Mechanical Engineer, Mr. E. C. Rodabaugh, Senior Mechanical Engineer, and Mr. T. J. Atterbury, Chief of the Applied Solid Mechanics Division are contributing to the high-pressure container design study. Data from which this report has been prepared are contained in Battelle Laboratory Record Book Nos. 21990 and 23055.

ABSTRACT

Development of the Manufacturing
Capabilities of the
Hydrostatic Extrusion Process

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et al.
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The revised hydrostatic extrusion container was completed in November 1965. Experimental extrusion trials to evaluate the critical variables of the process were resumed, but the number of trials was too few to ascertain definite effects at this point. A considerable number of extrusion trials is scheduled for the next quarter.

An analytical study of several design concepts for high-pressure containers was completed. The design concepts were evaluated from the standpoint of pressure capability, probable fatigue life, size, and ease of fabrication.

PUBLICATION REVIEW

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LIST OF SYMBOLS

E_n = modulus of elasticity of component n, psi

k_n = wall ratio of component n, $k_n \equiv r_n/r_{n-1}$

K = overall wall ratio of container, $K \equiv r_N/r_o$

K′ = wall ratio of inner part of ring-fluid-segment container, $K' = r_3/r_o$

N = the total number of components in a container; N also denotes the outermost component

n = a specific component when numbered from inside out; i. e., n = 1, 2, . . . , N

p = bore pressure, psi

p_3 = fluid support pressure for the ring-fluid-segment container, psi

r_n = outside radius of component n, inches

r_{n-1} = inside radius of component n, inches

r_o = bore radius of container, inches

r_N = outer radius of container, inches

S = shear stress, psi

S_r = semirange in shear stress for a cycle of bore pressure, psi

S_m = mean shear stress for a cycle of bore pressure, psi

S_{min} = minimum shear stress during a cycle of bore pressure, psi

S_{max} = maximum shear stress during a cycle of bore pressure, psi

σ = design tensile stress of ductile steel, psi ($\sigma \leqq$ ultimate tensile strength)

σ_1 = design tensile stress of high-strength steel, psi ($\sigma_1 \leqq$ ultimate tensile strength)

$(\sigma)_r$ = semirange in tensile stress for a cycle of bore pressure, psi

$(\sigma)_m$ = mean tensile stress for a cycle of bore pressure, psi

$(\sigma)_{min}$ = minimum tensile stress during a cycle of bore pressure, psi

$(\sigma)_{max}$ = maximum tensile stress during a cycle of bore pressure, psi

α_r = semirange stress parameter for high-strength steel, $\alpha_r \equiv (\sigma)_r/\sigma_1$

α_m = mean stress parameter for a high-strength steel, $\alpha_m \equiv (\sigma)_m/\sigma_1$

σ_r = radial stress, psi

σ_θ = circumferential (hoop) stress, psi

σ_z = axial (longitudinal) stress, psi

Δ_n = interference required between cylinder, n, and cylinder, n + 1, inches

Δ_{12} = interference required between the liner, segments, and cylinder, 3, of the ring-segment and ring-fluid-segment containers, inches

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INTRODUCTION

The purpose of the present research program is to develop the manufacturing capabilities of the hydrostatic extrusion process with the aim of extruding high-quality shapes from materials of interest to the Air Force. It is a continuation of the recently completed program on Contract No. AF 33(600)-43328. The current program is divided into two phases with the following general objectives:

Phase I. Process-Development Studies

Part 1. (a) To study the effect of critical process variables on pressure requirements and surface quality in hydrostatic extrusion of AISI 4340 steel, Ti-6Al-4V titanium alloy, and 7075 aluminum alloy.

(b) To correlate all available hydrostatic-extrusion-pressure data with material properties wherever possible in order to assist direction of the experimental effort and maximize the information developed in the present program.

Part 2. To explore the hydrostatic extrudability of TZM molybdenum alloy (cast and wrought), beryllium, Cb-752 columbium alloy, powder compacts, and other materials to be selected later in the program.

Part 3. To conduct a design study for high-temperature, high-pressure hydrostatic extrusion tooling based on
(1) estimated pressure requirements for high-ratio extrusion of materials of interest to the Air Force,
(2) latest high-pressure-vessel technology, and
(3) latest tooling materials available.

Part 4. To conduct a process economic study on the construction, installation, and operation of equipment with the same operational and size requirements as the tooling developed in the previous program on Contract No. AF 33(600)-43328.

Phase II. Process-Application Studies

Part 1. To evaluate the application of the hydrostatic extrusion process for sizing and finishing conventionally hot-extruded (or rolled) structural shapes by various combinations of drawing and extruding. Primary emphasis will be on AISI 4340 steel, although some effort will be devoted to Ti-6Al-4V, 7075 aluminum, and selected refractory metals.

Part 2. To determine the feasibility of producing wire and filaments from TZM molybdenum alloy and beryllium by combinations of hydrostatic extrusion and drawing.

Part 3. To develop tooling and define process parameters necessary for the reduction of tube blanks to finish tubing from AISI 4340 and a selected columbium alloy.

The experimental study of critical process variables (Part 1 of Phase I) was interrupted by failure of the liner component of the container assembly. The failure was attributed to low-cycle fatigue. The container was redesigned to improve low-cycle fatigue life by increasing the amount of prestress on the highly-stressed liner and sleeve.

During the last quarterly period, the hydrostatic extrusion trials were resumed and the high-pressure container design study was completed.

HYDROSTATIC EXTRUSION STUDIES

Fabrication of the redesigned container assembly was subcontracted to National Forge Company of Irvine, Pennsylvania. The container, scheduled to be completed by the week of September 6, was delivered about eight weeks later on November 10, 1965. Examination of the container on arrival at Battelle revealed some small nicks and gouges on the top end face of the liner component. To avoid the possibility of these defects acting as stress raisers, the top end face of the container assembly was surface ground to remove them.

Subsequently, the hydrostatic extrusion tooling was assembled in Battelle's 700-ton vertical hydraulic press and the fluid and stem pressure measuring instruments were calibrated. At this point, unfortunately, the time remaining in the present quarterly period was enough to scarcely begin the hydrostatic extrusion study scheduled for the second series of experiments. The objective of this series was to continue the study of the critical process variables (Part I of Phase I). Although a few trials were conducted, the numbers so far were too few to ascertain definite effects of the variables studied. Thus, reporting of the results of these trials has been postponed until the next interim quarterly report.

ANALYSIS OF SEVERAL HIGH-PRESSURE CONTAINER DESIGN CONCEPTS

An analytical study of several high-pressure container design concepts has been completed. Theoretical solutions were derived for the various designs. The analyses for maximum pressure capability, residual stresses, and required shrink-fit interferences were programmed for calculation on Battelle's CDC 3400 computer.

A detailed report of the study could not be finished in time for this interim report because the description of the analyses is fairly long and because there are many significant findings to be discussed. However, the results of immediate interest are available and are presented in this report. A complete and detailed description of the analyses will be included in the next interim report.

SCOPE OF ANALYSIS

The purpose of this study is to determine the maximum pressure capability of several designs of vessels for containing fluids at the pressures encountered in hydrostatic extrusion and other hydrostatic forming processes. Containment of bore fluid pressures up to 450,000 psi at room temperature and at temperatures of 500 F and 1000 F is considered.

Four types of pressure vessel designs were analyzed in detail. These are:

(1) Multi-ring container,
(2) Ring-segment container,
(3) Ring-fluid-segment container, and
(4) Pin-segment container.

The four cylindrical containers are shown in Figure 1. A wire-wrapped (strip-wound) vessel and a controlled fluid-fill, cylindrical-layered container were also considered, but only briefly.

The multi-ring container was one of the first design modifications of the monoblock thick-walled cylinder*. An initial compressive stress at the bore is achieved by shrink-fit assembly of successive cylinders each manufactured to provide an interference fit with its mating cylinder. The multi-ring container has been analyzed on the basis of static shear strength by Manning(1, 2, 3)†.

The ring-segment container with one outer ring was patented by Poulter(4) in 1951. One intent of this design is to reduce the pressure acting upon the outer ring by using a segmented cylinder to redistribute the pressure at a larger diameter. However, the inner cylinder is always subject to the bore pressure. The external diameter of the vessel necessarily increases with increasing segment size.

*The monoblock thick-wall cylinder is the simplest type of pressure container. However, for the very high-pressure levels considered in this study it is a relatively inefficient design.

† References listed at end of report.

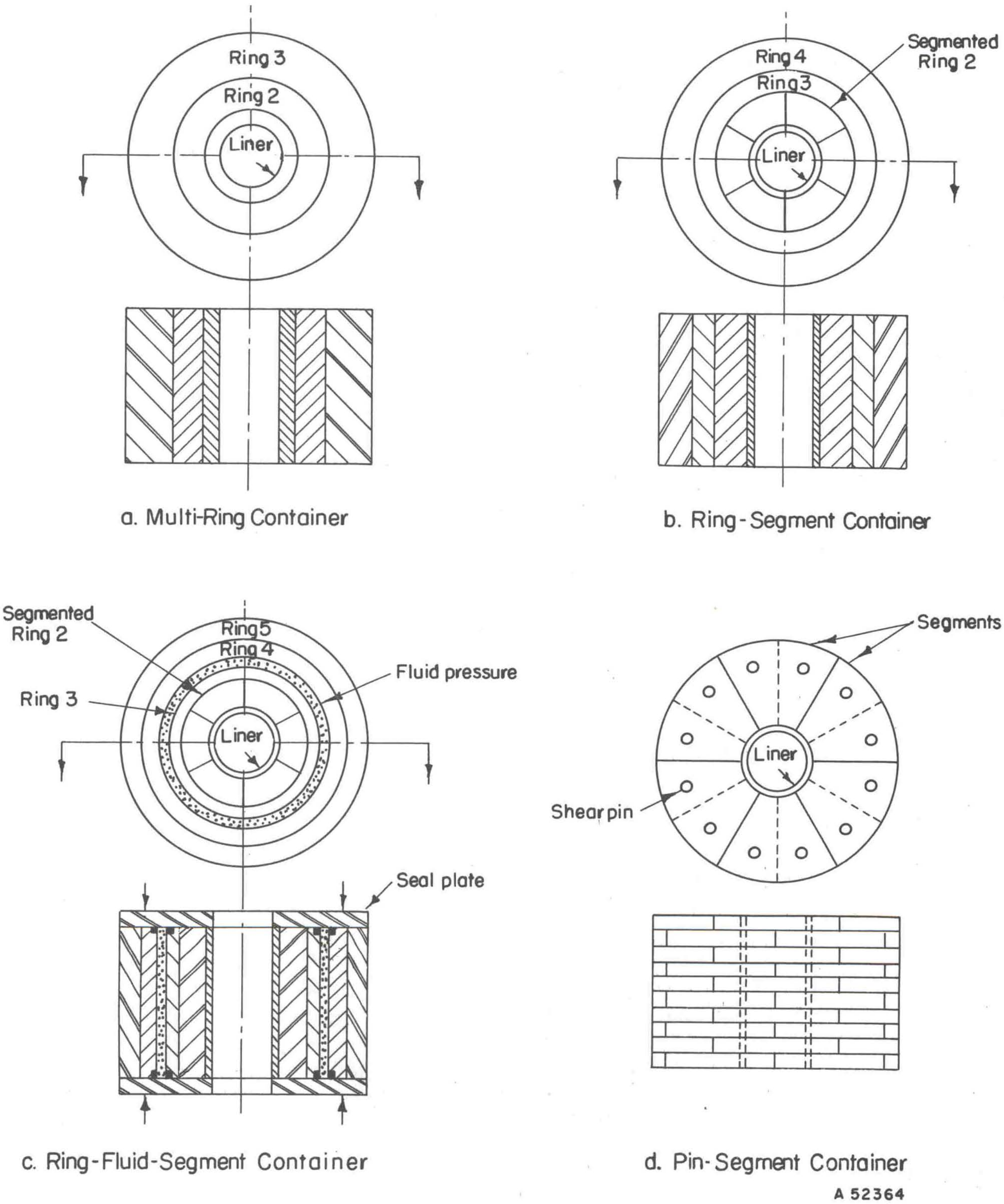

FIGURE 1. SCHEMATIC OF HIGH-PRESSURE-CONTAINER DESIGN CONCEPTS ANALYZED IN THE PRESENT STUDY

The ring-fluid-segment container makes use of the fluid-pressure support principle. This container is essentially constructed of two parts. The inner part is a ring-segment type container with one outer ring, but with a fluid support pressure, p_3, as shown in Figure 2(c). The outer part is a multi-ring container subject to an internal pressure, p_3, the support pressure for the inner part. The advantage of this design is that the fluid pressure (p_3) provides a compressive hoop stress at the bore which counteracts the tensile hoop stress resulting from the bore pressure, p. Theoretically, p_3 can be changed in proportion to the change in bore pressure in order to reduce the bore stress over an entire cycle of bore pressure. This variation of p_3 with the bore pressure is assumed in the analysis.

The origin of the ring-fluid-segment concept is not clear. Ballhausen(5) patented an approach of this sort in 1963. Another application of the same principle was patented by G. Gerard and J. Brayman(6), also in 1963. A similar design, but with additional features, was reported by F. J. Fuchs(7) in 1965.

The pin-segment design is an approach proposed by Zeitlin, Brayman, and Boggio(8). Like the ring-segment container this vessel also uses segments to reduce the pressure that must be carried by the external support. Unlike the ring-segment container, the pin-segment container has segmented disks (thin plates) rather than segmented cylinders. Also, the external supporting members in this case are pins rather than an external ring. The pins carry the reaction to the bore pressure predominantly in shear.

All four containers have one thing in common; the liner is subject to the full bore pressure. The four containers differ in the manner and in the amount they constrain the liner.

BASIS AND METHOD OF ANALYSIS

In this study the four design concepts for high-pressure containers are evaluated on the basis of a selected strength criterion for the component materials. A high-pressure container for commercial hydrostatic extrusion should, of course, be capable of repeated use without frequent failure. Therefore, it was considered essential that a fatigue strength criterion be used as the basis of evaluation in this study.

Manson and Hirschberg(9) have shown that for most materials, failure by low-cycle fatigue (life less than about 1000 cycles) involves almost entirely plastic strain. Above about 1000 cycles life the amount of plastic strain is appreciably smaller and above 100,000 cycles life the plastic strain is negligible. For the relatively high-strength materials, however, the strain is predominately elastic for lifetimes as low as 100 cycles. Because lifetimes greater than 1000 cycles are desirable in commercial applications and since high pressures require use of high-strength materials, the theory of elasticity is used in the analysis.

For the analysis, equations are derived that relate the interface pressures and the radial deformations between components. Elasticity solutions for stresses and deformations are used together with fatigue relations to determine formulas for maximum bore pressures. Stresses due to the bore pressure and shrink-fit assembly only are analyzed; no thermal gradients are assumed present. However, the effect of temperature change

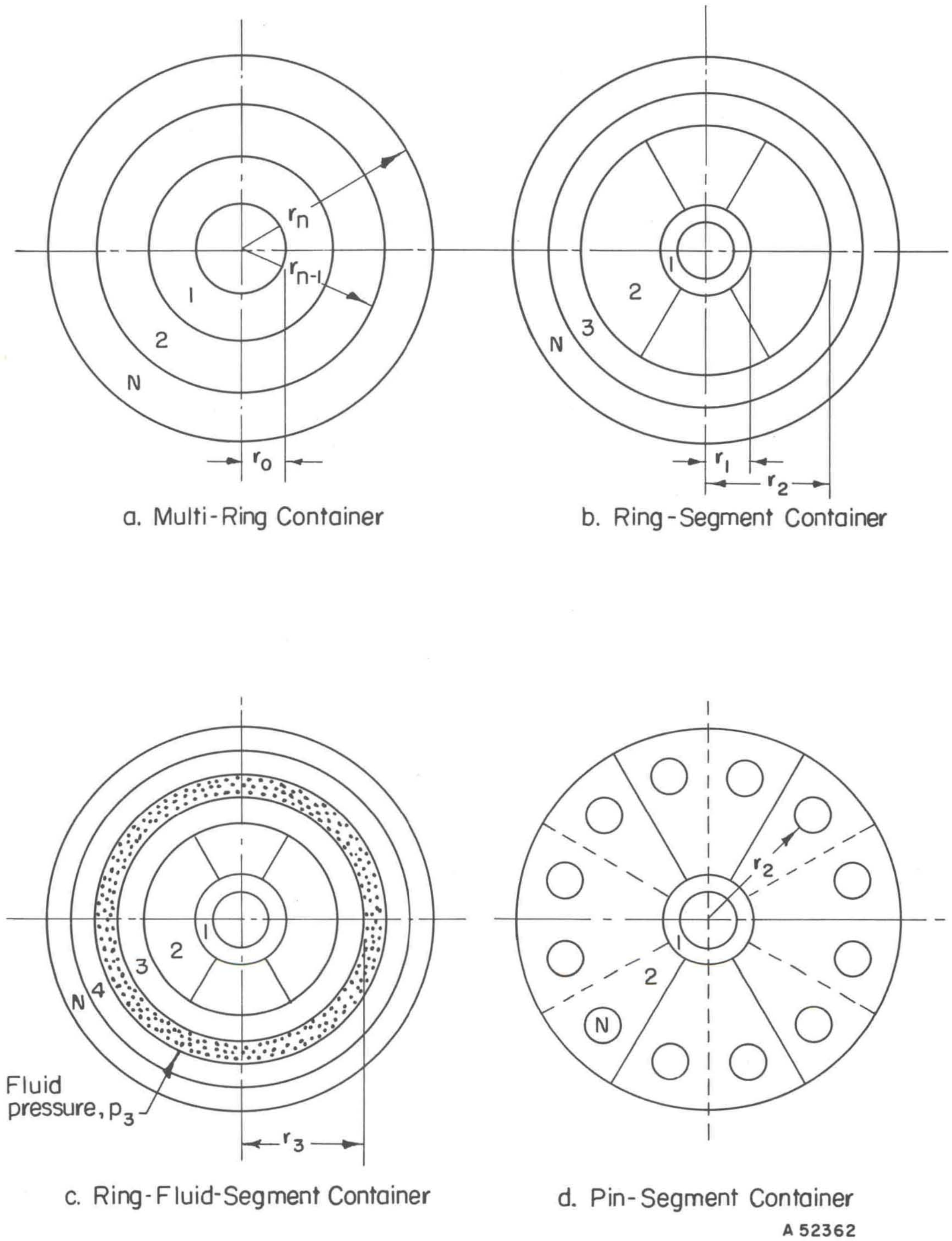

FIGURE 2. NOTATIONS USED FOR ANALYSIS OF CONTAINER DESIGN CONCEPTS

(from operating temperature to room temperature) upon the prestresses (residual stresses) is investigated. Excessive residual stresses may result because of differences in thermal expansion of the component parts of each container.

DEFINITION OF PARAMETERS

The components of each design are identified from the inside out by the numbers 1, 2, 3, . . . , N. N refers to the outermost components. As indicated in Figure 2, the components have the following radii:

$$
\begin{aligned}
r_0,\ r_1 &= \text{inner and outer radii, respectively, of component 1, the liner,} \\
r_{n-1},\ r_n &= \text{inner and outer radii, respectively, of component } n,\ n = 1, 2, \ldots, N.
\end{aligned} \tag{1}
$$

For the multi-ring container all the components are circular hollow cylinders. For the ring-segment and ring-fluid-segment containers, component 2 refers to the segments. The only exception to the notation on the radii occurs in the pin-segment design where the segment is divided for analysis into two parts and where r_2 is the radius to the inside of pins as shown in Figure 2(d).

The bore pressure is identified as follows:

$$p = \text{internal, bore pressure on liner.} \tag{2}$$

Wall ratios for each component are defined as follows:

$$k_n \equiv \frac{r_n}{r_{n-1}} \tag{3}$$

where k_n is the wall ratio for component n. The over-all diameter ratio of the container is defined as

$$K \equiv \frac{r_N}{r_o} \tag{4}$$

FATIGUE CRITERIA

Two fatigue criteria are formulated here in order that both relatively low-strength ductile materials and high-strength, more brittle materials may be used in one design. The intention is to use high-strength steels as liner materials and lower strength ductile steels for the outer cylinders in order to prevent catastrophic brittle failure.

Fatigue Criterion for Ductile Outer Cylinders

From both torsion and triaxial fatigue tests on low-strength (120 to 150 ksi ultimate strength) steels conducted by Morrison, Crossland, and Parry[10] it is concluded that a shear criterion applies. Therefore, a shear theory of failure is assumed for outer rings made of ductile steel.

In order to formulate a fatigue relation, the semirange in shear stress and the mean shear stress are needed. These stresses are defined as

$$S_r = \frac{S_{max} - S_{min}}{2} \qquad (5a, b)$$

$$S_m = \frac{S_{max} + S_{min}}{2}$$

respectively.

The Goodman fatigue relation in terms of shear stresses is assumed. This relation is

$$\frac{S_r}{S_e} + \frac{S_m}{S_u} = 1, \text{ for } S_m \geqq 0 ,$$

where S_e is the endurance limit in shear and S_u is the ultimate shear stress. For $S_u = 1/2\ \sigma_u$, where σ_u is the ultimate tensile stress, this relation can be rewritten as:

$$\frac{S_r}{S_e} + \frac{2S_m}{\sigma_u} = 1, \ S_m \geqq 0 \qquad (6)$$

The stresses S_r and S_m given by Equations (5a, b) can be calculated from elasticity solutions. In order to employ the fatigue relation (6) for general use, it is assumed that S_e can be related to S_u. This is a valid assumption as shown by Morrison, et al[10]. From the data of Reference (10), it is found that the following relation between S_e and σ_u may be assumed:

$$S_e = \frac{1}{3}\sigma_u \qquad (7)$$

Substitution of Relation (7) into (6) gives

$$3S_r + 2S_m = \sigma_u \qquad (8)$$

For design purposes this equation can be made conservative by rewriting it as

$$3S_r + 2S_m = \sigma, \text{ where } \sigma \leqq \sigma_u \qquad (9)$$

Equation (9) now has a factor of safety, σ_u/σ and can be expected to predict lifetimes for 10^6 cycles and greater for ductile steels based upon the Goodman relation and available fatigue data. (Of course, stress concentration factors due to geometrical discontinuities or material flaws would reduce the expected lifetime.)

Fatigue Criterion for High-Strength Liner

Triaxial fatigue data on high-strength steels ($\sigma_u \geqq 250$ ksi) are not available. In fact, fatigue data of any sort are very limited. Therefore, a fatigue criterion for high-strength steels under triaxial fatigue cannot be as well established as it was for the lower strength steels. The high-strength steels are expected to fail in a brittle manner. Accordingly, a maximum tensile stress criterion of fatigue failure is postulated.

Because fatigue data are limited while tensile data are available, the tensile stresses $(\sigma)_r$ and $(\sigma)_m$ are assumed to be related to the ultimate tensile strength by two parameters α_r and α_m, which are defined as follows:

$$\alpha_r \equiv \frac{(\sigma)_r}{\sigma_1}, \quad \alpha_m \equiv \frac{(\sigma)_m}{\sigma_1} \tag{10a, b}$$

where $(\sigma)_r$ is the semirange in stress, $(\sigma)_m$ is the mean stress*, and σ_1 is less than or equal to the ultimate tensile strength depending upon the factor of safety desired. In order to get some estimations of what values α_r and α_m may be, some fatigue data from the literature on rotating-beam and push-pull tests are examined. References (11), (12), (13), and (14) give such fatigue data for 18% Ni maraging, H-11, D6AC, and Vascojet 1000 and other high-strength steels having ultimate tensile strengths of 250,000 to 310,000 psi at room temperature.

The fatigue life again is found to depend on the range in stress and the mean stress, and upon the temperature. This dependence is illustrated in Figure 3 for 10^4 to 10^5 cycles life in terms of the parameters α_r and α_m. The 1000 F temperature data are for Vascojet 1000. Although α_r increases with temperature for this steel, the ultimate tensile strength decreases and the fatigue strength at 10^4 to 10^5 cycles for $\alpha_m = 0$ remains nearly constant over the temperature range of 75 F to 1000 F.

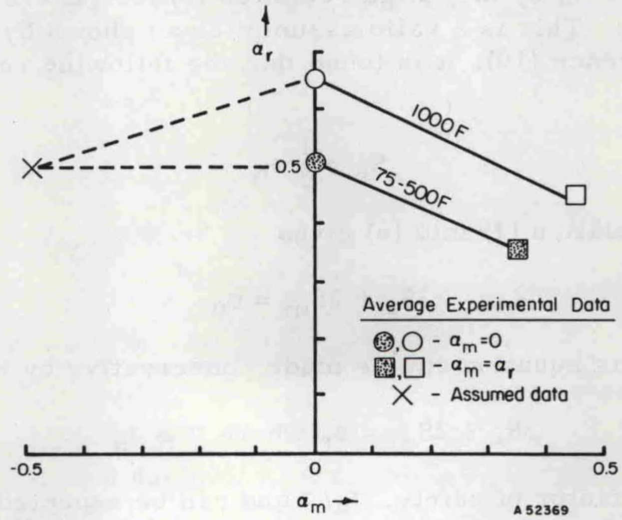

FIGURE 3. FATIGUE DIAGRAM FOR 10^4-10^5 CYCLES LIFE FOR HIGH-STRENGTH STEELS AT TEMPERATURES OF 75 F – 1000 F

α_r and α_m are defined by Equations (10a, b)

The fatigue data available are only for positive and zero mean stresses. However, there is evidence that compressive mean stress may significantly increase the fatigue strength[15, 16]. The reasons for this are thought to be that compression may reduce the detrimental effect of fluid pressure entering minute cracks or voids in the material and the compression may restrain such flaws from growing. Since the liner of a high-

*σ_r and σ_m are defined by expressions similar to Equations (5a, b) for S_r and S_m.

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 $K \to \infty$ 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.

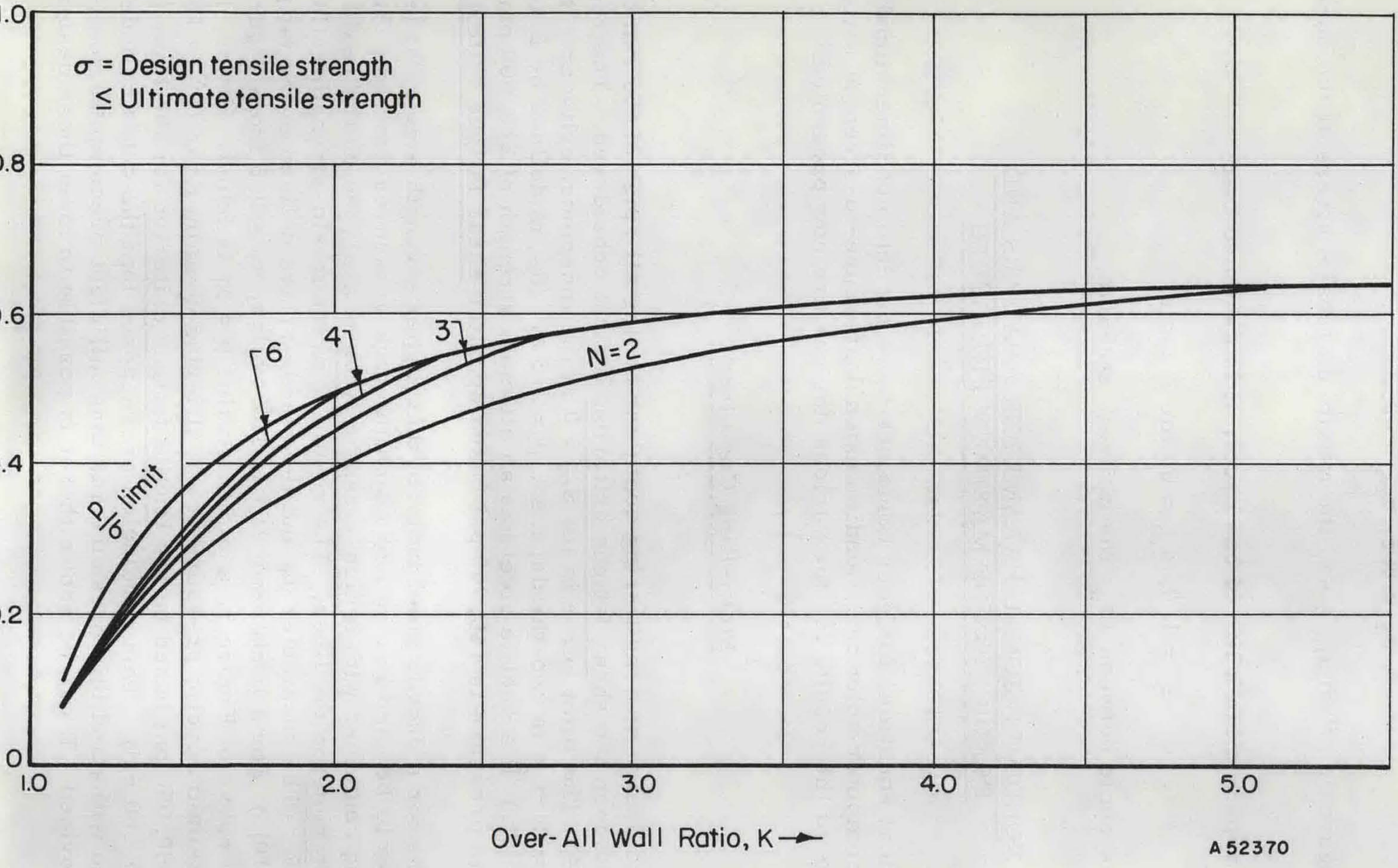

FIGURE 4. MAXIMUM PRESSURE-TO-STRENGTH RATIO p/σ IN MULTI-RING CONTAINER DESIGNED ON BASIS OF FATIGUE SHEAR STRENGTH

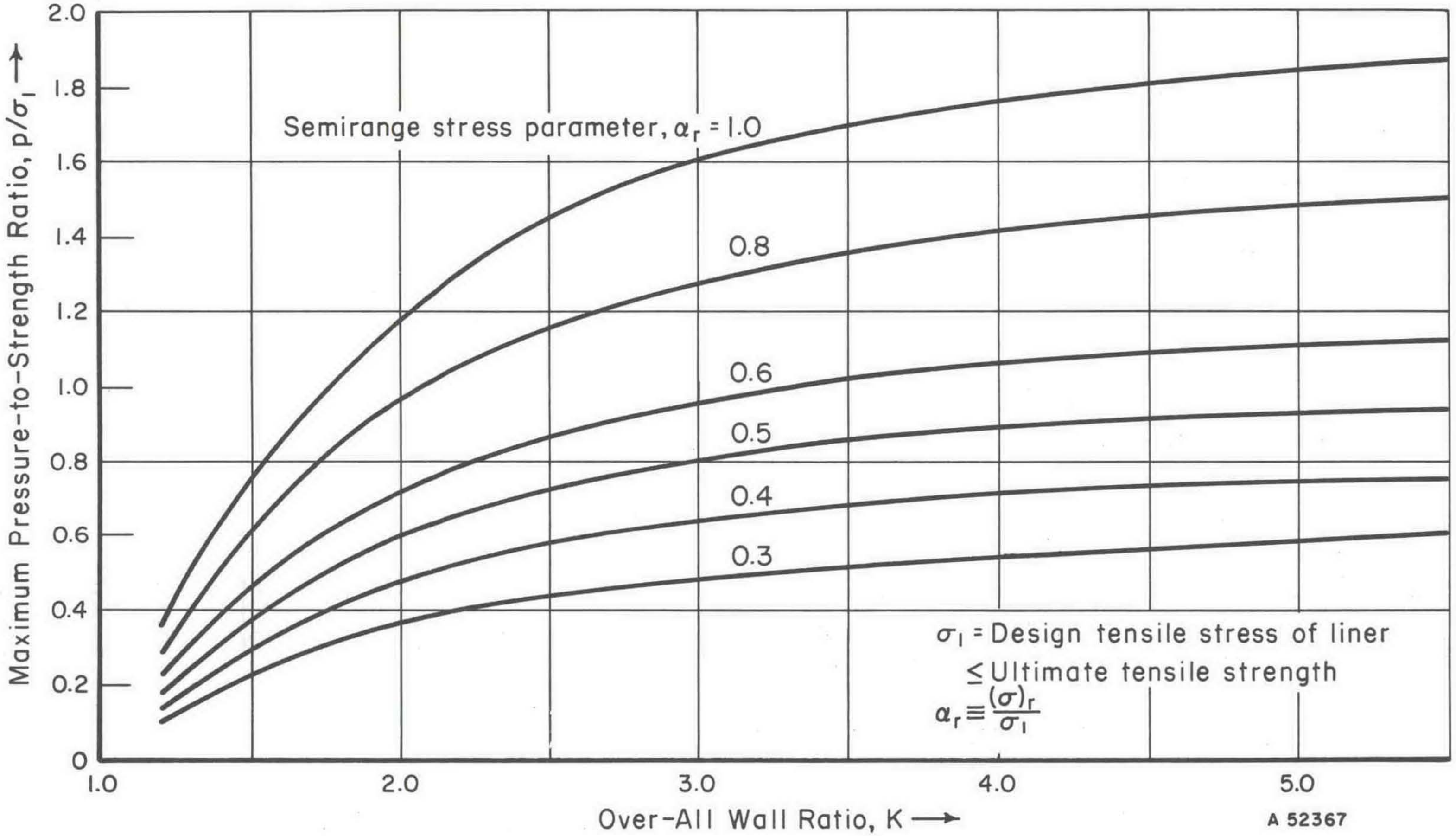

FIGURE 5. MAXIMUM PRESSURE-TO-STRENGTH RATIO IN MULTI-RING CONTAINER WITH HIGH-STRENGTH LINER BASED ON THE FATIGUE TENSILE STRENGTH OF LINER

Ring-Segment Container

The ring-segment container is also assumed to have a high-strength liner and relatively more ductile outer cylinders. The segment strength is assumed adequate. Its pressure-to-strength ratio p/σ_1 is plotted in Figure 6 for various k_1. From this figure it is evident that the ring-segment container cannot withstand as great a pressure as the multi-ring container if the over-all size is the same. This result is believed due to the fact that the segments do not offer any support to the liner – they are "floating" members between the liner and the third component, another ring. The effect is more pronounced as the segment size is increased; that is, the bore pressure capability decreases with increasing segment size. The detrimental effect of insufficient segment support to the liner can be reduced by using a high modulus material, such as tungsten carbide, for the segment material. However, in spite of this, it has been found that the reduction is not sufficient to increase the pressure capability of the ring-segment container to that of the multi-ring container for the same over-all wall ratio.

If the size and number of components of the ring-segment container are made large enough, then the pressure-to-strength ratio of this design can be made to approach that of the multi-ring container. Thus, its maximum cyclic pressure is 300,000 psi for 10^4 to 10^5 cycles life based upon an ultimate tensile strength of 300,000 psi for the liner.

Ring-Fluid-Segment Container

A high-strength liner and relatively more ductile outer cylinders are also assumed for the ring-fluid segment container. The functional dependence of the pressure-to-strength ratio of this container on the geometrical parameters is more complicated than for the other containers – mainly because of the additional parameter, the fluid support pressure, p_3. For example, Figure 7 shows that p/σ_1 decreases with segment size (k_2) for small K' but increases with segment size for larger K' ($K' \equiv k_1 k_2 k_3 = r_3/r_o$). This is an advantage over the ring-segment container (Figure 6) in which increasing segment size always has a detrimental effect. The pressure-to-strength ratio is also increased by increasing the support pressure p_3 as shown in Figure 8. (σ_1 and σ_3 are the design tensile stresses of the liner, $n = 1$, and the support cylinder, $n = 3$). With the high ratios shown, it is theoretically possible to have bore pressures as high as 1,000,000 psi in ring-fluid-segment container. However, practical limitations which are discussed later, considerably reduce the pressure capability of this design.

Pin-Segment Container

The analysis of the pin-segment container assumes a high-strength liner and lower strength, more ductile pins and segments. It is also assumed that any manufactured interference is taken up during assembly by slack between pins and holes. Therefore, the residual pressure between liner and segments is zero at room temperature but not zero at temperature if the coefficient of thermal expansion of the liner, α_1, is greater than that of the segments, α_2. In the analysis, it assumed that $\alpha_1 \geq \alpha_2$.

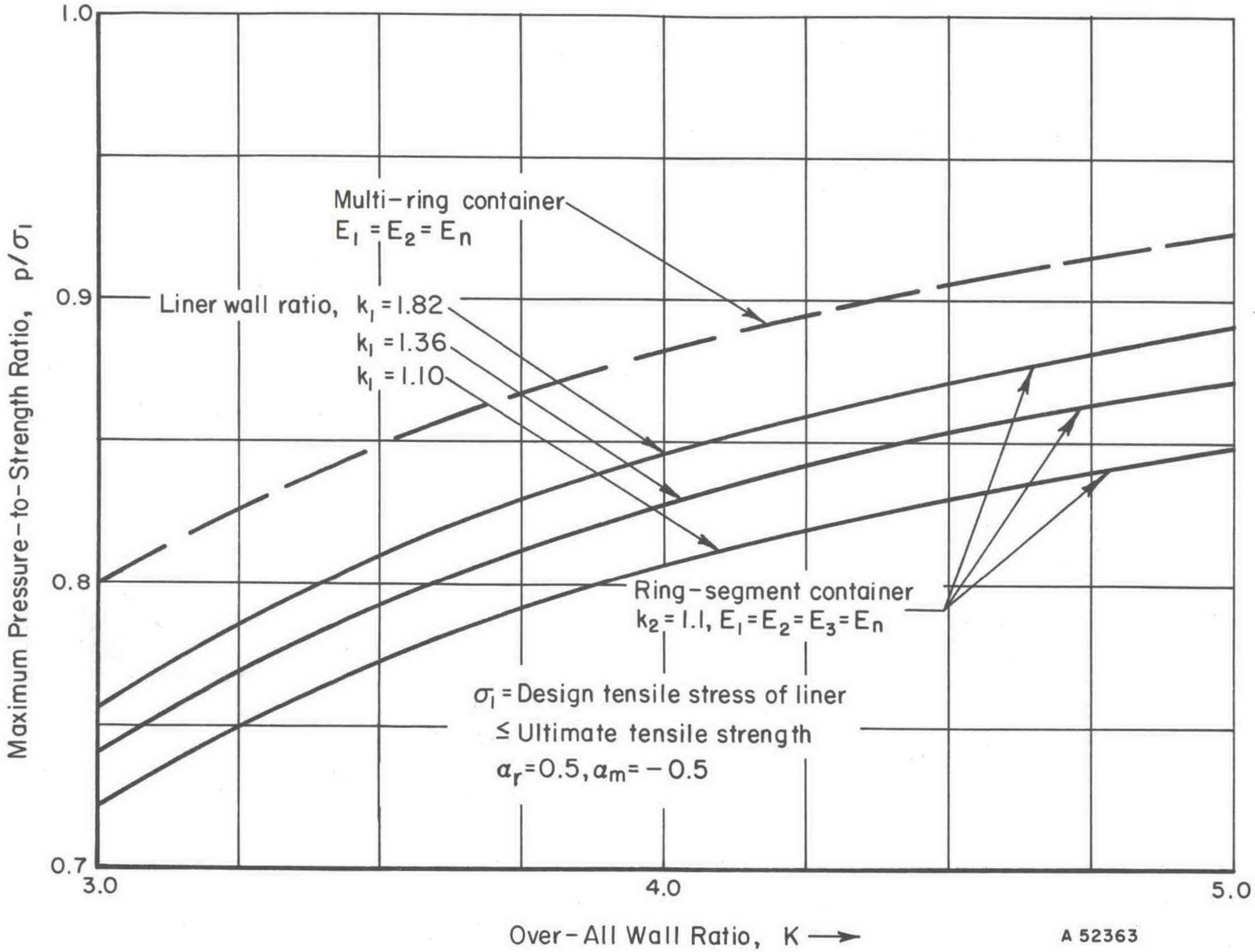

FIGURE 6. COMPARISON OF MULTI-RING CONTAINER WITH RING-SEGMENT CONTAINER FOR VARIOUS k_1

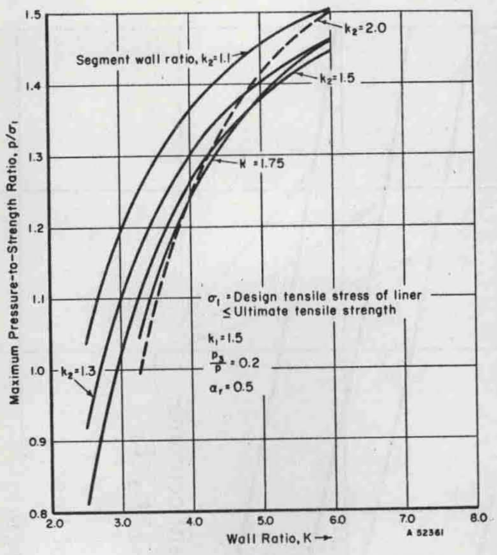

FIGURE 7. EFFECT OF SEGMENT SIZE ON THE PRESSURE-TO-STRENGTH RATIO ρ/σ_1 FOR THE RING-FLUID-SEGMENT CONTAINER

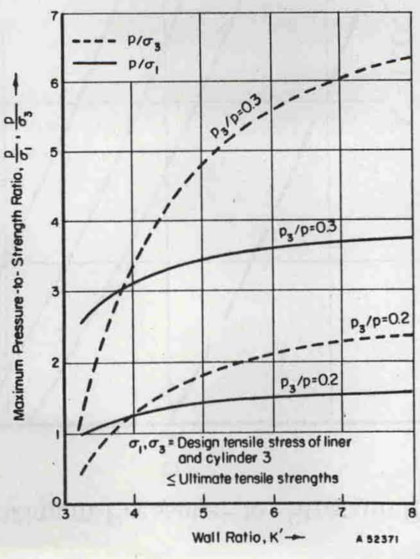

FIGURE 8. EFFECT OF SUPPORT PRESSURE ρ_3 ON BORE PRESSURE CAPABILITY FOR THE RING-FLUID-SEGMENT CONTAINER

The pressure-to-strength ratio p/σ_1 is plotted in Figure 9. Comparing this figure with Figure 5 for the multi-ring container with $\alpha_r = 0.5$, it is evident that both containers have the same limit $p/\sigma_1 \rightarrow 1$ for large wall ratios. However, $\alpha_r = 0.5$ is possible only if $\alpha_m \leqq 0$ as shown in Figure 3. Actually, $\alpha_m = +0.5$ is likely in the pin-segment container if $\alpha_r = 0.5$ because any interference is expected to be lost in taking up slack between pins and holes. In this case, then, $\alpha_r = 0.5$ would mean only one cycle life whereas $\alpha_r = 0.5$ means 10^4 to 10^5 cycles life in the multi-ring container. If this assembly problem could be eliminated by careful machining and selective fitting of pins, then theoretically, the p/σ_1 ratio of the pin-segment container could be made to approach that of the multi-ring container.

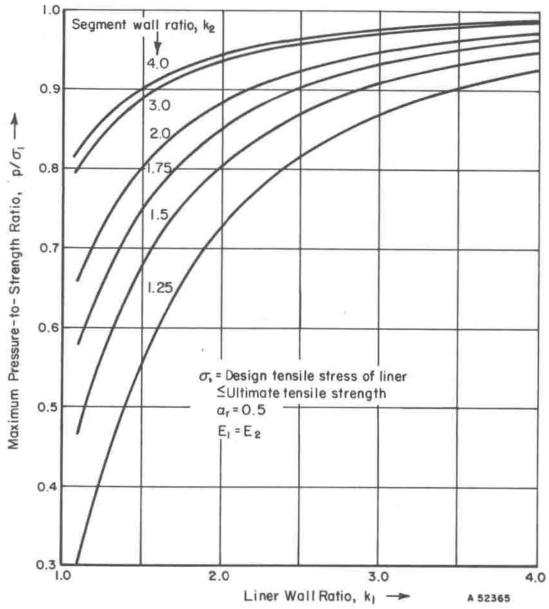

FIGURE 9. MAXIMUM PRESSURE-TO-STRENGTH RATIO, p/σ_1, FOR THE PIN-SEGMENT CONTAINER

Since no prestress has been assumed for the pin-segment container, $\alpha_r = \alpha_m = 0.35$ for 10^4 to 10^5 cycles as shown by Figure 3. For $\alpha_r = 0.35$, it is found that p/σ_1 is limited to 0.7 at best. Therefore, the maximum pressure in the pin-segment container is p = 0.7 (300,000) = 210,000 psi for 10^4 to 10^5 cycles life.

The stresses in the segments have also to be considered. It is found that high stresses develop around the pin holes. Their magnitudes decrease with increasing segment size. The shear stresses in the pins also need to be considered in order to determine the required pin size.

Strip-Wound Container

The strip-wound container uses basically the same principle as the multi-ring container. It has a cylindrical inner cylinder, the liner, under prestress, but the prestress in the liner is provided by wrapping strips or wire under tension onto the liner. It is possible to estimate the pressure-to-strength ratio of the strip-wound vessel if it is assumed that it behaves as a multi-ring container under internal pressure after the strip has been wound on. Referring to Figure 5 we see that the pressure-to-strength ratio p/σ_1 depends only on the over-all wall ratio K and α_r, the semirange stress parameter for the liner material. If K for the strip-wound vessel is taken as the ratio of the outside diameter of the last strip layer to the inner bore diameter, then Figure 5 can be used to estimate its pressure capability. Therefore, it may be concluded that the strip-wound container has a maximum pressure capability equal at best to that of the multi-ring container. However, unknown local stress concentrations and contact conditions between strips may be possible disadvantages in the strip-wound design.

Controlled Fluid-Fill Cylindrical-Layered Container

A controlled fluid-fill container, shown in Figure 10, has been proposed by Berman[16]. All the rings are assumed to be made of the same ductile material and a shear strength criterion applies. Like the ring-segment-fluid container this container also uses the fluid-pressure support principle. The advantage of this design is that residual stress limitations can be overcome by controlling the fluid pressures p_n; i. e., the pressures p_n can be reduced to zero as the bore pressure, p, is reduced to zero. There are no shrink-fits, so there are no residual stresses.

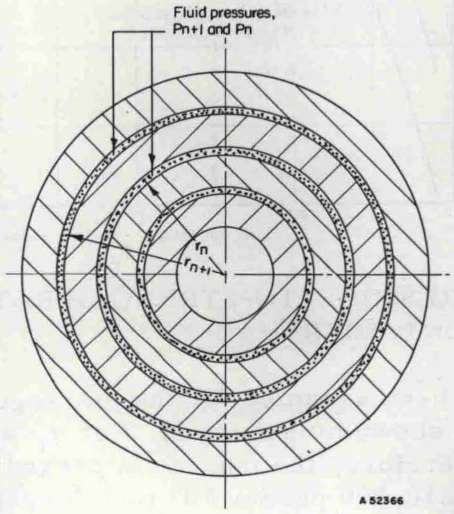

FIGURE 10. CONTROLLED FLUID-FILL CYLINDRICAL-LAYERED CONTAINER (REFERENCE (16)).

Berman's analysis was based upon static strength. A similar analysis at Battelle was based upon fatigue strength. A surprising result was found; Figure 4 applies also to this design – except for the limit curve which does not. Therefore, 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 practicable. If, for example, σ = 150, 000 psi (ultimate strength of a ductile steel), N = 8 and K = 16, then it is found that p = 240, 000 psi. Thus, for fatigue applications with bore pressures of 240, 000 psi and greater, the controlled-fluid-fill container may become too large to be practicable. Furthermore, eight rings also means there are seven annuli under fluctuating fluid pressures. Design of mechanical apparatus to supply and control all these pressures presents practical difficulties as well.

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	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 $p \leq 210,000$ 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.

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 $\leqq$ 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.

TABLE 2. LINER-BORE STRESSES AND INTERFERENCES FOR A 6-INCH BORE MULTI-RING CONTAINER WITH $K = 8.5$, $N = 5$, $k_1 = 2.0$, $k_n = 1.44$, $n \geqq 2$, $\alpha_r = 0.5$, $\alpha_m = -0.5$[a]

	Stresses at Bore of Liner[b]								
	Residual Stresses at RT			Prestresses at Temperature			Operating Stress at Pressure and Temperature		
	σ_r/σ_1	σ_θ/σ_1	S/σ_1	σ_r/σ_1	σ_θ/σ_1	S/σ_1	σ_r/σ_1	σ_θ/σ_1	S/σ_1
RT Design	0	-1.000	-0.5000	0	-1.0000	-0.5000	-0.9727	0	0.4863
500 F Design	0	-1.1230	-0.5615	0	-1.0000	-0.5000	-0.9727	0	0.4863
1000 F Design	0	-1.2998	-0.6499	0	-1.0000	-0.5000	-0.9727	0	0.4863

	Dimensionless Interference Required as Manufactured [c]	
	Between Cylinders 1 and 2 for $p = 300,000$ psi[d], $E\Delta_1/r_1p$	Between Outer Cylinders n and n + 1 $E\Delta_n/r_np$
RT Design	0.358	0.343
500 F Design	0.454	0.343
1000 F Design	0.533	0.343

(a) The k_n, K, α_r, and α_m are defined by Equations (3), (4), and (13a, b), respectively. Material data are given in Table 1. The liner is 18% Ni steel and the outer cylinders are H-11 steel.

(b) σ_r is the radial stress, σ_θ the hoop stress, S the shear stress ($S \equiv (\sigma_\theta - \sigma_r)/2$), and σ_1 is the design strength – less than or equal to the ultimate tensile strength of the liner.

(c) E is the modulus of elasticity of the outer cylinders. Δ_n is interference in inches between cylinders n and n + 1. r_n is the outer radius of cylinder n.

(d) $E\Delta_1/r_1p$, at elevated temperatures, depends on p. $\sigma_1 = 310,000$ psi is required, ($p = 0.9727\ \sigma_1$).

TABLE 3. LINER-BORE STRESSES AND INTERFERENCES FOR A 6-INCH BORE MULTI-RING CONTAINER WITH $K = 8.5$, $N = 5$, $k_1 = 2.0$, $k_n = 1.44$, $n \geq 2$, $\alpha_r = 0.5$, $\alpha_m = -0.3$[a]

	Stresses at Bore of Liner[b]								
	Residual Stresses at RT			Prestresses at Temperature			Operating Stress at Pressure and Temperature		
	σ_r/σ_1	σ_θ/σ_1	S/σ_1	σ_r/σ_1	σ_θ/σ_1	S/σ_1	σ_r/σ_1	σ_θ/σ_1	S/σ_1
RT Design	0	-0.8000	-0.4000	0	-0.8000	-0.4000	-0.9727	0.2000	0.5863
500 F Design	0	-0.9054	-0.4527	0	-0.8000	-0.4000	-0.9727	0.2000	0.5863
1000 F Design	0	-1.0505	-0.5253	0	-0.8000	-0.4000	-0.9727	0.2000	0.5863

	Dimensionless Interference Required as Manufactured[c]	
	Between Cylinders 1 and 2 for $p = 300,000$ psi[d], $E\Delta_1/r_1p$	Between Outer Cylinders n and n + 1 $E\Delta_n/r_np$
RT Design	0.217	0.304
500 F Design	0.309	0.304
1000 F Design	0.383	0.304

(a) The k_n, K, α_r, and α_m are defined by Equations (3), (4), and (10a, b), respectively. Material data are given in Table 1. The liner is 18% Ni Steel and the outer cylinders are H-11 steel.

(b) σ_r is the radial stress, σ_θ the hoop stress, S the shear stress ($S \equiv (\sigma_\theta - \sigma_r)/2$), and σ_1 is the design strength – less than or equal to the ultimate tensile strength of the liner.

(c) E is the modulus of elasticity of the outer cylinder. Δ_n is interference in inches between cylinders n and n + 1. r_n is the outer radius of cylinder n.

(d) $E\Delta_1/r_1p$, at elevated temperatures, depends on p. $\sigma_1 = 310,000$ psi is required ($p = 0.9727\ \sigma_1$).

Other Possible Material Limitations

It has been postulated that a compressive mean stress may benefit the material fatigue strength under cyclic fluid pressure. However, triaxial fatigue behavior under compressive mean stress is unknown.

Also unknown is the possible fracture of high-strength steels under large compressive stresses. Pugh and Green[17] and Crossland and Dearden[18] found for cast iron that the fracture strain and ductility (and the maximum shear stress at fracture) are increased by superimposing hydrostatic pressure. This is a favorable result, but the possibility of similar behavior for the high-strength steels should be investigated.

The effect of a brittle-ductile transition in high-strength steels on the fatigue behavior near and above the transition temperature is another factor which may need to be considered.

Huge outer cylinders are required for some of the high-pressure container designs – cylinders up to 10 to 50 feet in diameter with 1- to 3-foot wall thicknesses are necessary in some designs. As mentioned previously, fabrication of such large, forgings may be extremely difficult or impossible. Even transportation of such large forgings is another problem.

RECOMMENDATIONS

Proposed Materials Study

The possible material limitations discussed in the preceding section suggests that a materials study be conducted. The triaxial fatigue behavior of high-strength steels under compressive mean stress should be investigated. The objective of the study would be to establish a fatigue criterion for these materials. The effect of large pressures, of magnitudes one to three times the ultimate tensile strength, upon the flow and fracture characteristics of high-strength steels should also be studied. Moreover, a brittle-ductile transition in high-strength steels may influence fatigue behavior at elevated temperatures – an investigation of this factor may also be worthwhile.

Suggested High-Pressure Container

The results of the investigations on various containers have shown that fluid-pressure support is beneficial and that prestress is also beneficial in increasing the predicted fatigue strength under cyclic pressure loading. Use of high-strength steels for the liners of the containers was also found necessary. Although the controlled fluid-fill design, Figure 10, uses the fluid-support principle, the required size and complexity of the fluid-fill apparatus for fatigue application makes this design impracticable. Use of shrink-fit to provide compressive prestress can reduce the required size and the number of pressure annuli as the ring-fluid-segment design indicates. Although the latter design has the benefit of prestress from shrink-fit, it requires large interferences because of large deformations of the segments and large outer cylinders because the segments offer no hoop support.

A suggested design which appears to minimize the problems introduced by segments is shown in Figure 11. It is made up of two multi-ring units and a fluid-pressure support annulus. Three rings are shown in each part in Figure 11, but the number of rings can be varied to give the best design. For example, for containers having small bores, one ring is sufficient in the inner part. It is easily shown (using the tensile fatigue criterion for the inner ring) that a cyclic bore pressure of 450,000 psi is possible with one inner ring of wall ratio, $k_1 = 2.0$ and a support pressure of 250,000 psi. A multi-ring container for the outer part can be designed for 10^4 to 10^5 cycles at 250,000 psi as shown in this study.

It may be that the fluid-support pressure should not be reduced to zero with the bore pressure but reduced to some minimum value in order to provide some prestress in the outer cylinder of the inner part. Controlling the pressure in one annulus does not present as many difficulties as it does in the controlled fluid-fill container design where there are many annuli.

The suggested design can be analyzed using analyses similar to those used in this study. It is suggested that this be done.

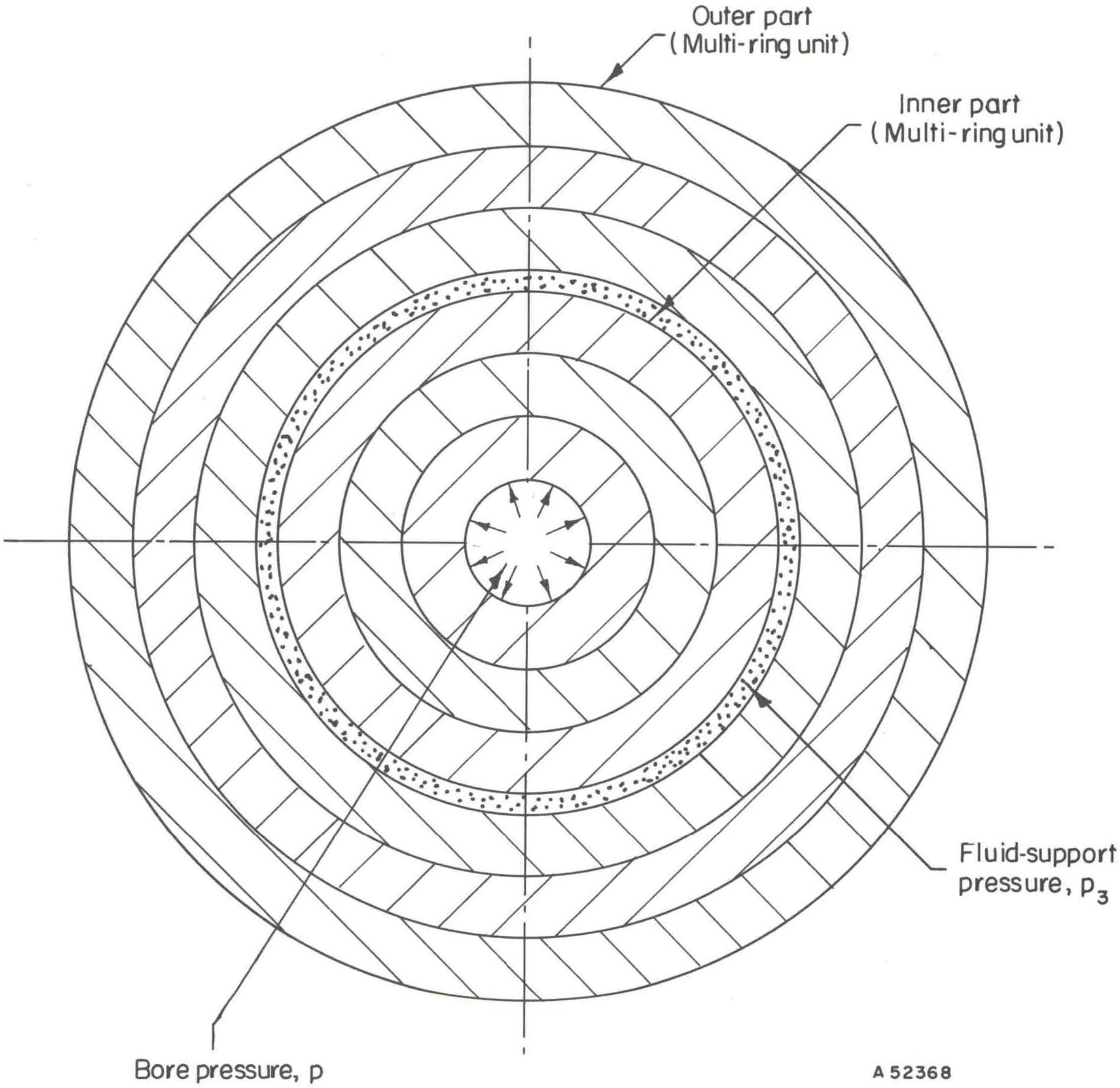

FIGURE 11. SUGGESTED FLUID-SUPPORT MULTI-RING CONTAINER FOR HIGH PRESSURE

The design involves the combined use of interference-fit multi-ring construction with fluid-pressure support.

FUTURE WORK

It is anticipated that, during the next quarter, the experimental portion of the hydrostatic extrusion program will be fully resumed. In addition to studies of the critical process variable, the hydrostatic extrudability of relatively more difficult-to-work materials such as TZM molybdenum alloy and beryllium will be evaluated. In addition, finishing of shapes, fabrication of wire, and extrusion of tubing will have begun.

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