64-12

ASD TR 8-111 (III)

ASD INTERIM REPORT 8-111 (III)
APRIL 1964

III (III)

ELECTRO-SPARK EXTRUDING

J.H. WAGNER

REPUBLIC AVIATION CORPORATION
MANUFACTURING RESEARCH

CONTRACT: AF 33(657)11265
ASD PROJECT: 8-111

INTERIM PROGRESS REPORT
1 JANUARY 1964 to 1 APRIL 1964

The effects of shock loading on a thick walled cylinder at an internal pressure of 200,000 psi hydrostatic pressure are analysed. A container geometry is proposed.

BASIC INDUSTRY BRANCH
MANUFACTURING TECHNOLOGY LABORATORY

AERONAUTICAL SYSTEMS DIVISION
UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

ASD TR 8-111

ASD INTERIM REPORT 8-111 (III)
April, 1964

ELECTRO-SPARK EXTRUDING

J. Wagner

Republic Aviation Corporation
Manufacturing Research Department

Contract AF33(657)11265
ASD Project 8-111

Interim Technical Engineering Report No. 3
1 January 1964 - 1 April 1964

Republic Aviation Corporation Report No. RAC 2288

The effects of shock loading on a thick walled cylinder at an internal pressure of 200,000 psi hydrostatic pressure are analysed. A container geometry is proposed.

BASIC INDUSTRY BRANCH
MANUFACTURING TECHNOLOGY LABORATORY
Aeronautical Systems Division (AFSC)
United States Air Force
Wright-Patterson Air Force Base, Ohio

ABSTRACT-SUMMARY
Interim Technical
Progress Report No. 3

ASD INTERIM REPORT 8-111 (III)
April, 1964

PORT 8-111 (III)

ELECTRO-SPARK EXTRUDING

J. Wagner

Republic Aviation Corporation

The spark gap discharge phenomena as applied to experimental extrusion are reviewed. The equations derived for a thick walled cylinder under transient internal loading are applied to check the stress levels encountered in two proposed experimental extrusion devices. The strength requirements of the equipment are reviewed and analysed with both static and dynamic loading. A container concept is presented.

ental extrusion
linder under
evels encountered
ength requirements
c and dynamic

NOTICE

When Government drawings, specifications or other data are us for any purpose other than in connection with a definitely related Gove ment procurement operation, the United States Government thereby in no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied t said drawings, specifications, or other data, is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation or conveying any rights or permission to manufacture, use or sell any patented invention that may in any way b related thereto.

The work reported in this document has been made possible thro the support and sponsorship extended by the Air Materiel Command ur Contract No. AF33(657)11265. It is published for technical informatic only and does not necessarily represent recommendations or conclusi of the sponsoring agency.

Qualified requesters may obtain copies of this report from Defe Documentation Center, Cameron Station, Alexandria, Virginia.

Copies of ASD Technical Reports should not be returned to the Aeronautical Systems Division unless return is required by security considerations, contractual obligations, or notice on a specific docum

NOTICE

When Government drawings, specifications or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data, is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation or conveying any rights or permission to manufacture, use or sell any patented invention that may in any way be related thereto.

The work reported in this document has been made possible through the support and sponsorship extended by the Air Materiel Command under Contract No. AF33(657)11265. It is published for technical information only and does not necessarily represent recommendations or conclusions of the sponsoring agency.

Qualified requesters may obtain copies of this report from Defense Documentation Center, Cameron Station, Alexandria, Virginia.

Copies of ASD Technical Reports should not be returned to the Aeronautical Systems Division unless return is required by security considerations, contractual obligations, or notice on a specific document.

FOREWORD

This Interim Technical Progress Report covers the work performed under Contract AF33(657)11265 from January 1, 1964 to April 1, 1964. It is published for technical information only and does not necessarily represent the recommendations, conclusions or approval of the Air Force.

This contract with Republic Aviation Corporation of Farmingdale, New York, was initiated under ASD Manufacturing Technology Laboratory Project 8-111, "Electro Spark Extruding." It is administered under the direction of Mr. T.S. Felker of the Basic Industry Branch MATB, Manufacturing Technology Laboratory, Aeronautical Systems Division, Wright-Patterson Air Force Base, Ohio.

Mr. J.H. Wagner of the Manufacturing Research Department, Republic Aviation Corporation is the engineer in charge of the project. Mr. Gunther Pfanner is cooperating in the research.

The transient wave analysis presented in this report was done by Mr. B.P. Leftheris of Re-entry Simulation Laboratory, Research Division.

The primary objective of the Air Force Manufacturing Methods Program is to increase producibility, and to improve the quality and efficiency of fabrication of aircraft, missiles and components thereof. This report is being disseminated in order that methods and/or equipment developed may be used throughout industry, thereby reducing costs and giving "MORE AIR FORCE PER DOLLAR."

Your comments are solicited on the potential utilization of the information contained herein as applied to your present or future production programs. Suggestions concerning additional manufacturing methods development required on this or other subjects will be appreciated.

PUBLICATION REVIEW

Approved by: Robert W. Hussa
Robert W. Hussa, Ass't. Chief
Manufacturing Rsch. Engineer

Approved by: T. F. Imholz
T. F. Imholz, Chief
Mfg. Rsch. Engr.

CONTENTS

INTRODUCTION

Electro-spark extrusion is a metalforming concept involving recovery of the energy derived from a rapid discharge of capacitor stored electrical power by reduction of a metal billet through an appropriate die. One method, the electrohydraulic reduction of the billet, places the extrudable material in an ultra-high pressure fluid environment with the rapid energy discharge occurring across a suitable electrode gap. This short time duration gap discharge in a properly designed high strength container causes a shock-pressure wave to strike the billet face and, upon reflection, to initiate particle acceleration in the billet material and to release to the billet face some of the intrinsic kinetic and pressure energy available as a consequence of known wave reflection parameters. The sum of these phenomena when added to the potential energy present in the high fluid pressure environment have been shown by mathematical analysis to be ample under carefully controlled conditions to cause a metal billet to yield and to extrude a given length for the time duration of one electrical pulse. The problem of maintaining extrusion then properly becomes one of providing sufficient discrete pulse discharges with the correct time interval to allow a constant velocity extrusion.

In addition, the capacitor discharge equipment can be alternately employed to produce impulsive electromagnetic forces upon the billet in the hydrostatic container. In this method, the rapid discharge of the stored electrical energy produced high magnetic fields about an appropriately constructed conductive coil. By induction, to a conductive piston-disc interposed between the coil and the billet, repulsive magnetic forces rapidly accelerate the disc and thereby drive a pressure wave against the billet.

As an alternative, where the billet material is sufficiently conductive, the electromagnetic repulsion may be coupled to the billet face directly in order to induce the acceleration forces required to extrude the billet.

A third method of capacitor stored energy utilization is to recover the energy created by the electrohydraulic discharge and to transfer it by mechanical means to the billet. Displacement of the billet is accomplished mechanically by the use of a piston-ram on the billet material in a suitable container. The pressure pulse or electromagnetic energy created by a series of discharges impinges on the face of the piston causing it to move forward to extrude the billet.

The advantages offered by the foregoing approach to electrohydraulic extrusion are: (1) elimination of billet container wall friction, (2) reduction of billet die surface friction, (3) creation of pressures considerably beyond those obtained hydrostatically, (4) reduced container size and cost.

To determine the production potential for extruding a steel alloy by the use of capacitor stored energy, the Aeronautical Systems Division of Wright Patterson Air Force Base has awarded Contract No. AF33(656)-11265 to Republic Aviation Corporation. Extrusion will be attempted in the course of the program by conducting experiments with capacitor discharge equipment to develop the most suitable techniques for metal reduction by several electrohydraulic methods. The program consists of two phases as follows:

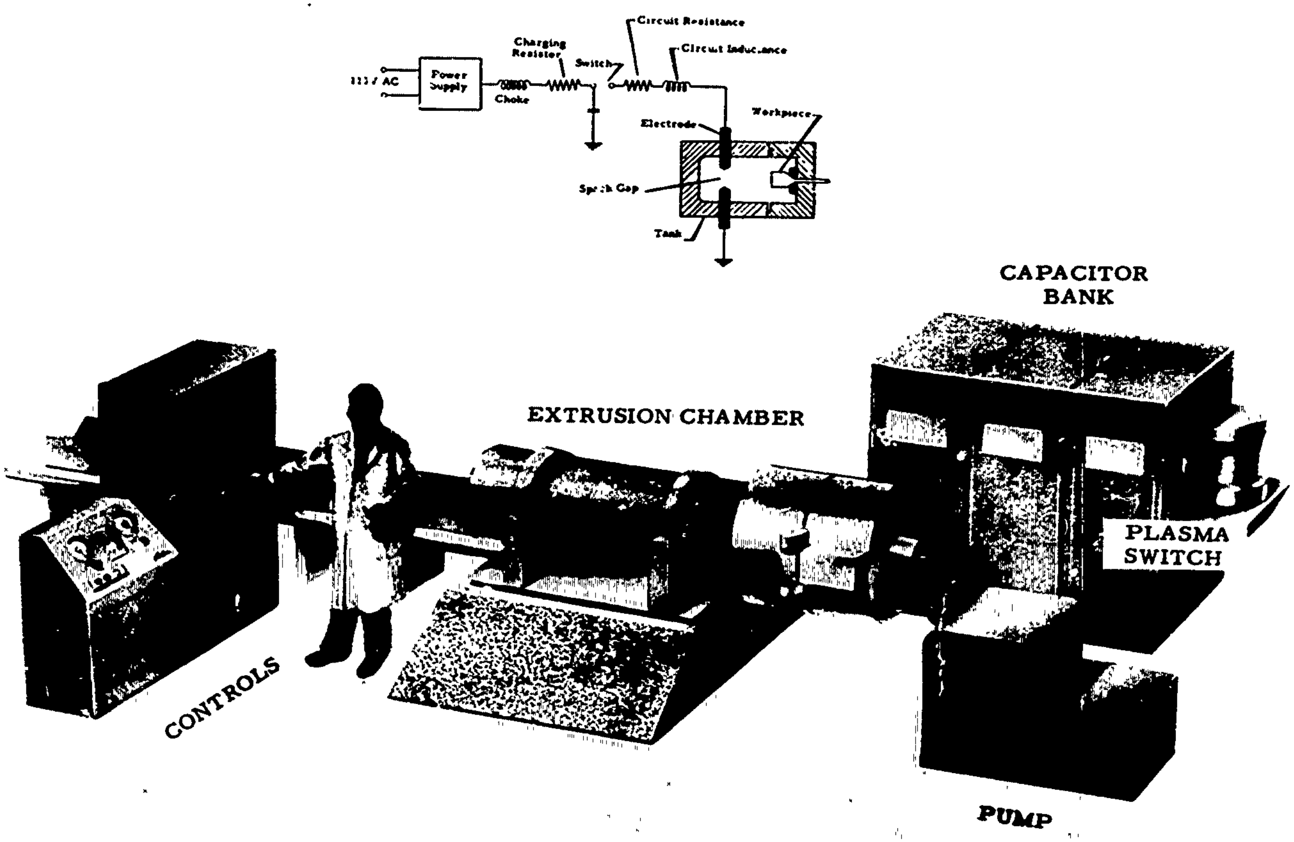
Phase I - Design of the Extrusion Equipment
Phase II - Development of the Extrusion Process

The approaches mentioned in the foregoing are to be thoroughly investigated for suitability toward attaining the objectives of the program; i.e., extrusion of a 2 inch diameter to a 1/4 inch round.

Mathematical treatment of the electrohydraulic pulse extrusion case (Reference 1) produced a descriptive equation containing a combination of dynamic and static variables generally similar to the well known equation used for static extrusion ($P = K \ln \frac{A}{a}$). In the static extrusion case, the pressure is raised until extrusion starts; while in the pulse extrusion case, the pulse propagates through the billet imparting a motion to the billet. The extrusion produced for one pulse depends upon the pressure amplitude, while the final extruded length depends upon the rate the pulses are generated.

From the foregoing, it is apparent that achievement of the total energy necessary to extrude a target section requires multiple electrical discharges. The time between discharges should be minimum to reduce the overall extrusion time and avoid stop-start extrusion. In order to optimize these requirements, it is necessary to modify the 156,000 joule Republic Aviation capacitor discharge facility to include the necessary equipment to allow faster charging rates and to design a switching arrangement and duty cycle timer to favor the rapid cyclical release of stored energy at precisely timed intervals.

The theory of unsteady waves, (Reference 1) used to develop working equations for computing the radial and hoop strains in hollow cylinders under transient internal loads indicated that pressure vessel design requirements to contain dynamic forces superposed over high hydrostatic pressures was possible and practicable. This analysis pointed up desirable container dimensional geometry and indicated an order of magnitude of mechanical properties for the materials in the critical areas of the chamber. These criteria have been formulated into general design requirements and are presented in this report.



ELECTRO-SPARK EXTRUSION EQUIPMENT AND PROCESS CONCEPT

FIGURE 1

DESIGN OF A PRESSURE VESSEL FOR ELECTROHYDRAULIC EXTRUSION

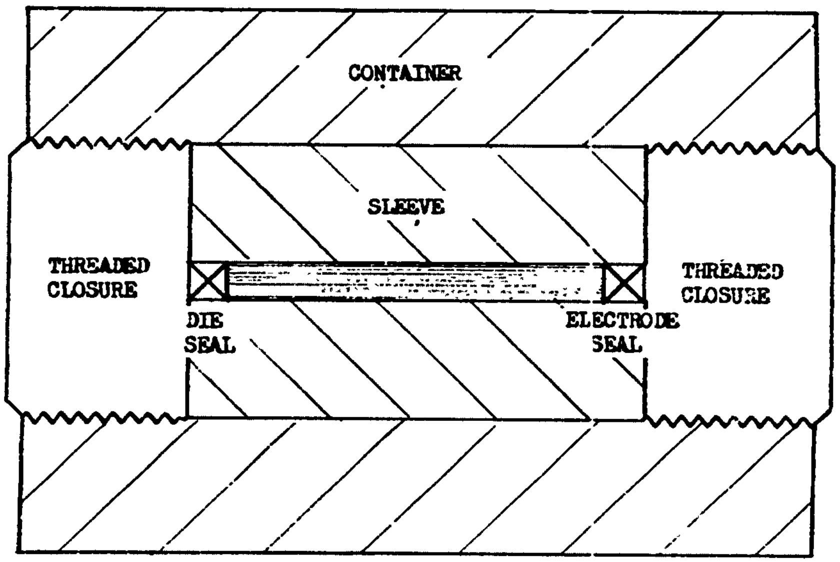
A. Basic Equipment Design

1. The Spark Discharge Extrusion Concept

To accomplish extrusion, power is initially supplied to a capacitor bank and the stored charge held off by a suitable switch until released to the high pressure device containing an open gap electrode, an extrusion die and a metal billet. The billet is maintained under high hydrostatic pressure defined by the mechanical properties of the particular extrudable material used and the degree of metal reduction required. Once the storage of energy is completed, the discharge event begins when a trigger switch (vacuum gap) is ionized and current flows into the circuit, thereby ionizing the high fluid pressure gap between the electrodes (refer to Figure 1). A spark channel is abruptly created and the vapor products in the channel expand into a spherical "gas bubble." The inertia of the circumjacent fluid coupled with the extremely high velocity of expansion of the "gas bubble" produces a compression shock wave in the surrounding liquid. This compressed water layer then travels through the liquid as a shock pressure front at approximately acoustic velocity with a time duration of approximately 20 microseconds.

The energy released by the gap discharge consists of kinetic energy plus the energy due to the increased pressure in the traveling wave. At the billet face the pressure approximately doubles because of the reinforcement of the incident and reflected waves. The shock pressure wave splits into a reflected pulse back into the fluid and a transmitted pulse forward into the billet material. The billet attains, by the same phenomena as the fluid, a particle velocity sufficient to cause longitudinal movement for a length proportional to the energy released.

The very benefits derived from the shock pressure pulse in extrusion (i. e., particle velocity sufficient to cause plastic strain and movement of the billet)become a limiting factor when applied directly to the structure in the pressure vessel walls. As the cylinder is initially under high



SINGLE SHRINK SHROUD ASSEMBLY
FIGURE 2

internal static pressures the effects of the superposed dynamic loading become critical. Results of the analysis completed in subject Reference 1, indicated a practical container geometry is attainable to withstand the explosive forces from an electrical gap discharge event.

The illustration (Figure 1) shows the spark discharge extrusion process and equipment in a conceptual manner. A capacitor bank to supply the energy, a high pressure chamber, pumping system and associated controls are shown.

2. Effect of Shock Loading in Container Cavity Dimension

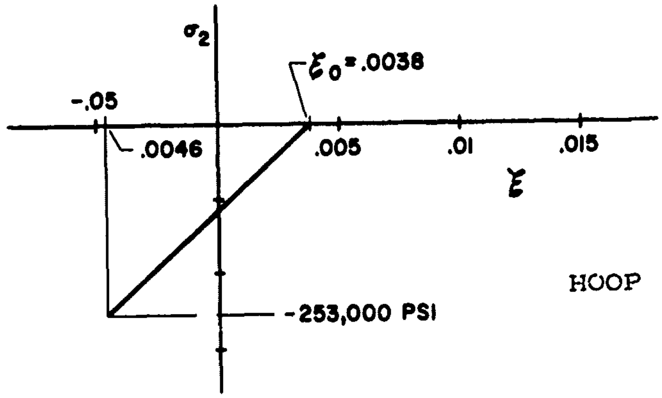
Equations have been previously derived and calculations prepared in Reference 1 for the general problem of a thick-walled cylinder under the influence of a transient internal load where the unbalanced forces from a detonative capacitor discharge set up velocity gradients in part of the cylinder while the remainder is completely at rest. The problem has been formulated and explicit solutions given for elastic and elasto-plastic cases. These specialized interpretations are presented in Appendix I for reference.

With the aforementioned analysis in mind, a tentative pressure vessel construction (Figure 2) using a single shrink shroud of 4340 steel (24 inch O. D.) over an 18% Ni. maraging steel liner (2 inch I. D.) with a static internal pressure of 200,000 psi was derived using the customary Lamé equations. A desirable residual compressive hoop stress resulting from the combination of shrink fit assembly and cold work fabrication procedures in the most highly stressed portion of the apparatus, (the inside diameter; $r_o = 1$ inch) was required to withstand safely hydrostatic internal pressures to the maximum amount of 200,000 psi.

The question of a serviceable design to contain the forces released by the discharge of electrical capacitor stored energy in addition to the loading imposed on the equipment by the high order of hydrostatic pressure presented a problem to which no exact solution could be found. For this reason, the mathematical analysis, Reference (1), was prepared to predict the effects of detonative shock loads on a container using normal construction materials of significant mechanical strength values.

This analysis indicated that residual hoop stresses result in the container walls from reaction of the metal to the passing of a maximum detonative shock pressure pulse. The maximum response occurs at the fluid metal interface at the internal radius (r_o = 1 inch) where the pressure pulse is greatest. The material has been shown by the analysis to experience first plastic flow in the outward direction and then inward, thus giving rise to the possibility of secondary yielding.

Reference to Case 1 in Appendix I, the hoop strain for zero stress (permanent set) is 0.0036. The residual strain after unloading is 0.00463 (compression). The diagram for this set of conditions indicates that a permanent set to a residual compressive stress level at the bore

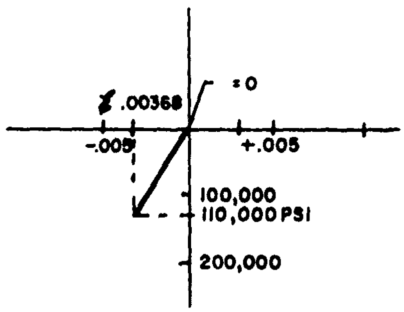


of 253,000 psi. This is a value greater than the ultimate strength of the liner material (250,000 psi) being considered without the addition of hydrostatic pressure. Therefore, the assembly shown in Figure 2 is proven unsatisfactory, because of the high degree of deformation present at the critical location r_o = 1 inch from the action of the passing of the shock wave.

As the reverse yielding at r_o = 1 inch has been shown to be beyond the mechanical property values of 18% Ni. maraging steels, it is of interest to find a radius where the severity of the shock wave has been sufficiently diminished so as to present a less drastic effect on the walls of the pressure container. A plot, Figure 3 of the particle velocity against the radius in the container wall indicates that the particle velocity attenuates rapidly from its initial critical value of u = 2864 inches per second. It is reasonable to assume that a wall radius can be found where the effect of the particle velocity on the container walls is reduced to the degree that severe deformations do not occur from the loading-unloading cycle of the transient wave. The curve changes slope between the values r = 2 and r = 3 inches. At r = 3 inches, a 40% attenuation has diminished the particle velocity to a value of 1654 inches per second. This value of r_o = 3 inches was selected for analysis.

The calculations for an r_o = 3 inches are outlined in Appendix I, Case (2).

The hoop strain imposed at r_o = 3 inches from the action of the loading wave where the pressure has attenuated to 248, 280 psi has been shown using the concept of elastic recovery to be equal to zero, thus indicating no permanent set in the hoop direction from the loading wave action. Due to the unloading there is elastic recovery strain in the amount of 0. 00368 (compression). The equivalent stress for this strain value is 110, 000 psi (compression). The significance of this is that a permanent



residual hoop stress (compression) due to the action of the wave remains at the inside diameter of the chamber (r_o = 3 inches). If there were no

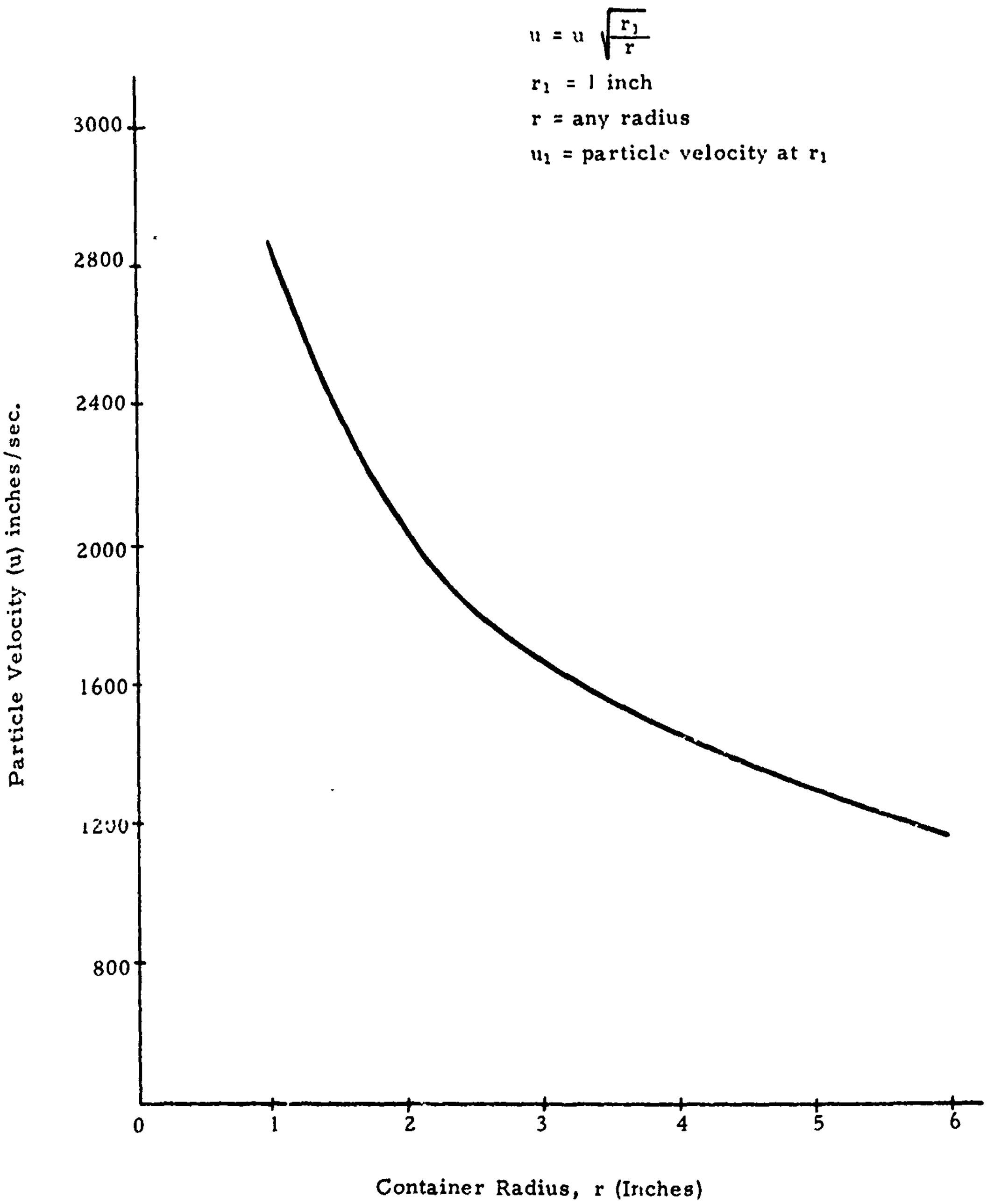

$u = u_1 \sqrt{\frac{r_1}{r}}$
r_1 = 1 inch
r = any radius
u_1 = particle velocity at r_1
Particle Velocity (u) inches/sec.
3000
2800
2400
2000
1600
1200
800
0
1
2
3
4
5
6
Container Radius, r (Inches)

FIGURE 3

considerations, other than hoop stress, the pressure vessel at $r_o = 3$ inches could elastically withstand dynamic stresses from -110,000 psi to the yield point of the material, 250,000 psi or a total of 360,000 psi in the hoop direction.

It must be remembered that the analysis outlined in the Appendix I indicates the condition in the pressure vessel walls at $r_o = 3$ inches. This case is for the same wave (at a later period in time) as was analyzed for $r_o = 1$ inch. The particle velocity and the pressure, of course, are attenuated from passage through the material of the pressure vessel walls. Although the calculations show a small residual compressive stress of 30,000 psi, this value can be disregarded if we consider $r_o = 3$ inches to be the container cavity dimension at the fluid-metal interface. (This will be made further evident in the following section.)

3. Combined Effect of Internal Detonative and Hydrostatic Loading

Because the shock wave loading by itself has been previously shown to behave elastically between 110,000 and the yield point of the material in the wall at $r_o = 3$ inches, it becomes of interest to subject the stresses produced from a combination of the dynamic and static loading to analysis to see if the coupling can be accomplished without failure in the wall. The calculations for this are outlined in Appendix II.

The work indicates that a compressive pre-stress in the hoop direction of approximately 133,000 psi must be incorporated in the pressure vessel construction by some method to insure that failure will not occur at $r_o = 3$ inches as a result of the combining of the hydrostatic and dynamic loading. This order of compression stress is to be obtained by the use of careful shrink fit procedures in addition to appropriate autofrettage of the inner liner prior to assembly. As an interference shrink fit construction will be used. The interference must be calculated commensurate with ordinary assembly procedures. In addition, the residual stress level after shrink and at pressure must be maintained so that this area will not be subject to failure during usage.

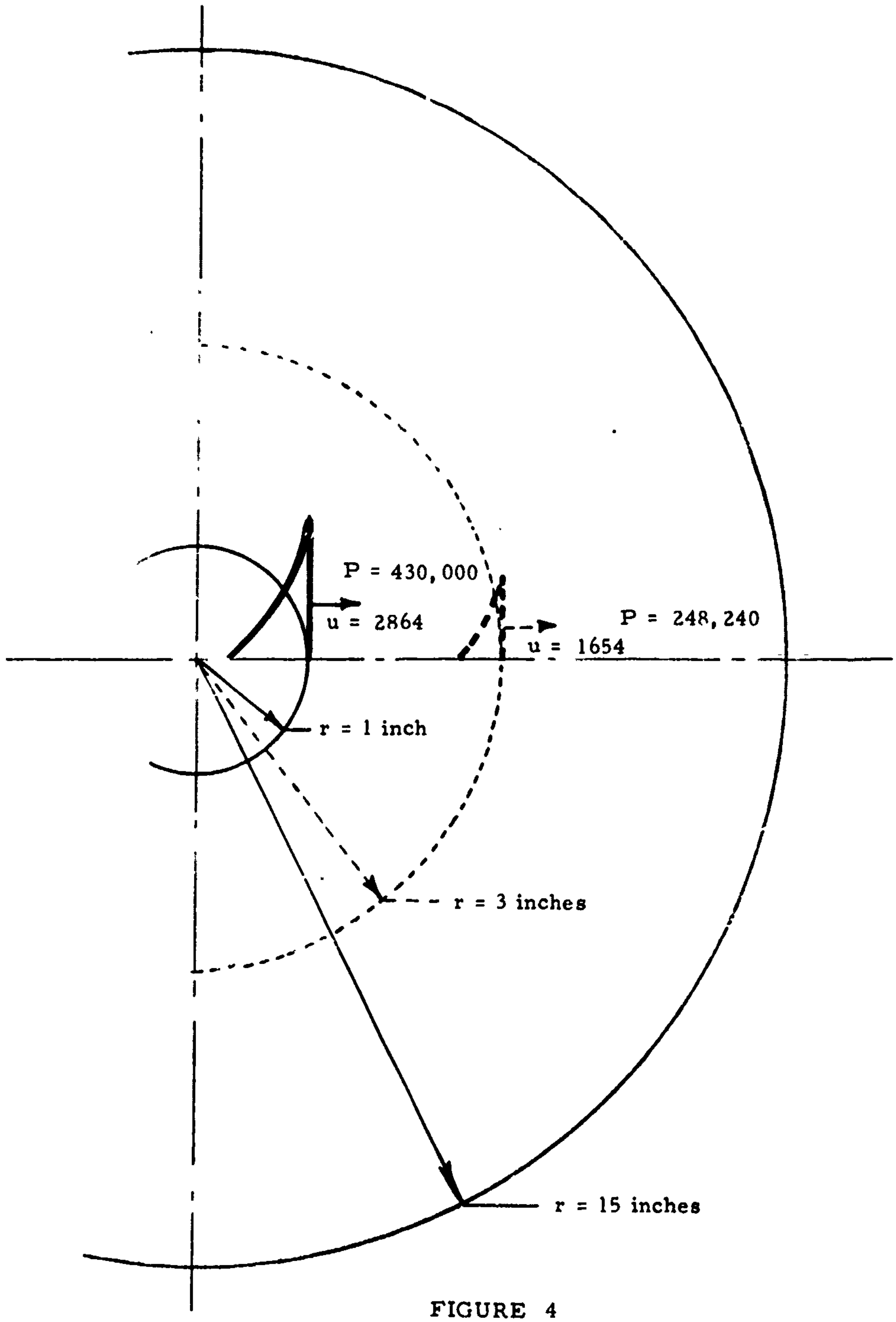


FIGURE 4

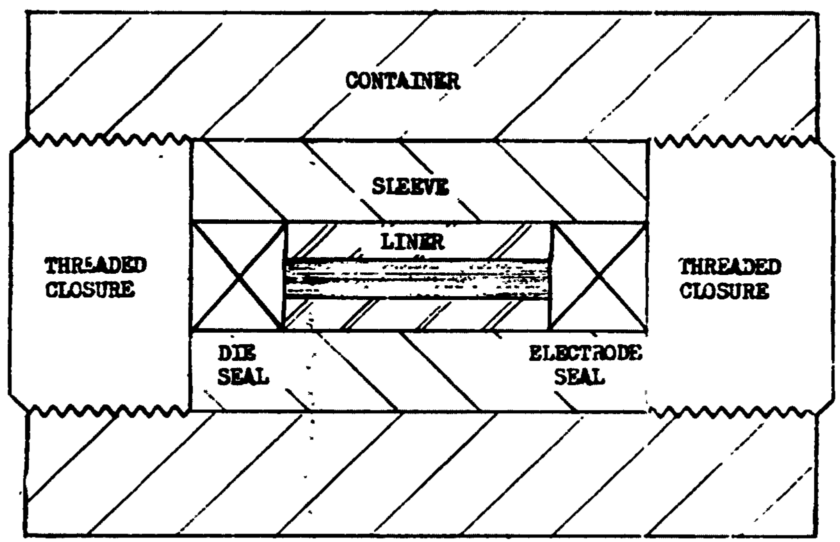
4. Replaceable Liner Construction

The analyses and calculations presented for r_o = 1 inch and r_o = 3 inches container dimensions have considered the decay of one transient detonative shock wave in a cylinder from a point at the pressurized fluid-metal interface outward for a distance of three inches. Figure 4 shows the respective condition of one wave at r_o = 1 inch and r_o = 3 inches at two different instants of time. If we should consider a chamber only of cavity diameter of 6 inches (r_o = 3 inches) the pressure upon reflection would again be 430,000 psi and the effect on the container wall would be too severe to contemplate for the proposed experiments. If we, however, take the maximum pressure (430,000 psi) at the 2 inch diameter and allow the effects (particle velocity and pressure) to attenuate by some means, the consequences of the event at the 6 inch diameter interface can be tolerated.

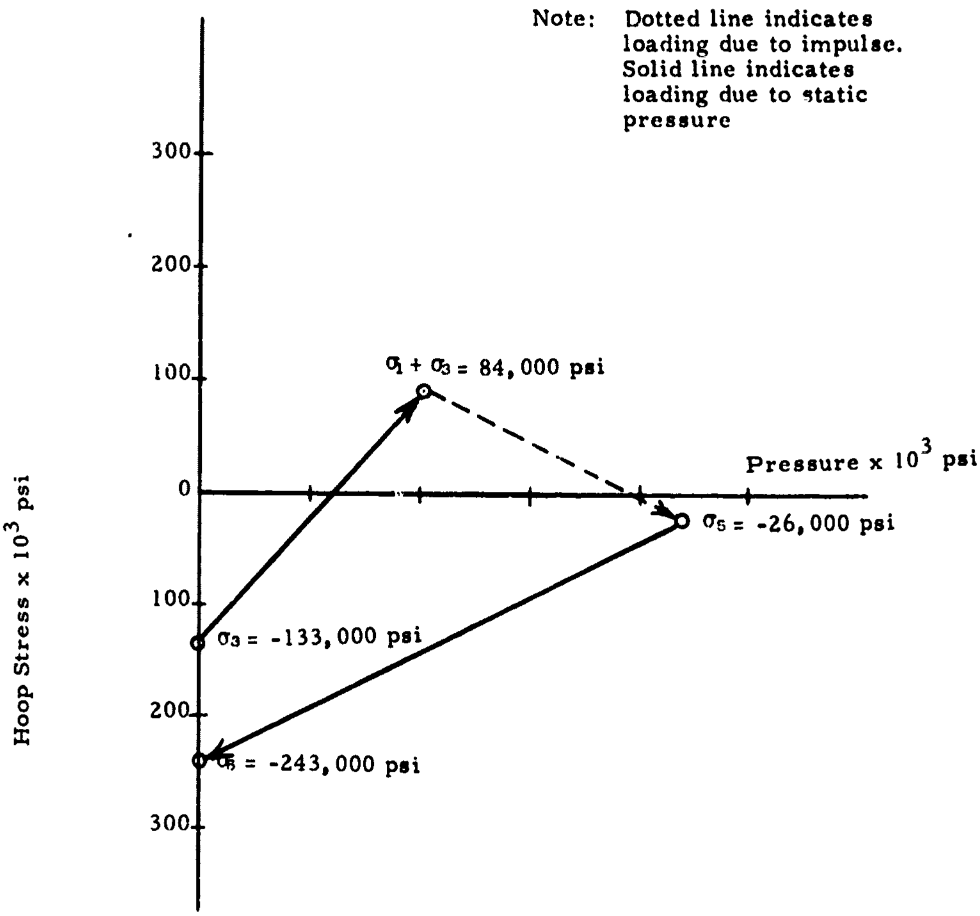
The means proposed to mitigate the severity of the shock pulse on the wall material is the insertion of a removable liner 2 inch I. D. x 6 inch O. D. in the pressurized container. This liner will be completely restrained by the hydrostatically pressurized fluid in the radial, tangential and axial direction. Such a construction is shown in Figure 5.

It is anticipated that over a period of time after a specific number of maximum duty cycles, the replaceable liner should fail. When this occurs, the liner can be removed and a replacement provided.

The initial static and dynamic loading cycle is depicted graphically in Figure 6. σ_3 is the initial prestress at P = 0. Upon the insertion of pressure into the cavity, the stress is raised to $\sigma_1 + \sigma_3$. The first maximum discharge resets this stress pattern to the value at σ_5 and upon release of pressure, the residual hoop stress returns to the point of σ_6. This is then the permanent working stress in the cavity of the pressure chamber.



PROPOSED PRESSURE VESSEL CONSTRUCTION
FIGURE 5



STRESS DISTRIBUTION DUE TO STATIC AND DYNAMIC LOADING

FIGURE 6

STATUS OF THE WORK

The analyses derived for the effect of detonative shock loading in a thick walled cylinder hydrostatically loaded have been completed and the conclusions derived therefrom have been applied to the design of a suitable piece of equipment to be used in the extrusion experiments.

All vendor proposals for equipment have been reviewed for conformance to specification and suitability of costs, and based upon this review, an agreement has been entered into with Harwood Engineering Company, Walpole, Massachusetts to develop with Republic Aviation Corporation, and to manufacture the equipment required for the proposed experiments. Harwood has started fabrication of a 200,000 psi high pressure pumping system to displace 20 in ^{3}/min. Engineering has been completed in the design of the pressure vessel and work has commenced with the assembly drawings. These should be ready for detailing early in the next reporting period.

Work has proceeded with the electrical considerations necessary to alter the Republic Aviation Corporation 156,000 joule capacitor bank to attain suitable charging rates and to release energy to the extrusion chamber at the repetition rates that are anticipated to be necessary to accomplish extrusion. This work will continue for the next reporting period.

REFERENCE

1. "Electro-Spark Extruding," Republic Aviation Corporation Report No. 2027, (ASD TR 8-111 (II), 1 January, 1964

APPENDIX I

CALCULATION FOR STRAIN DEFORMATION DUE TO TRANSIENT WAVE

I Statement of the Problem

Consider a thick-walled cylinder with σ_y = 250,000 psi and the following dimensions (the cylinder is filled with water):

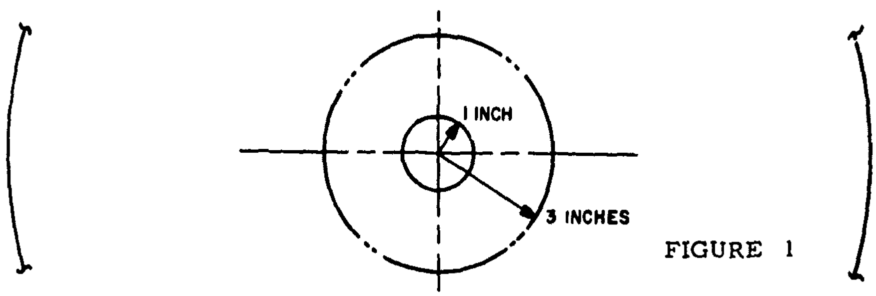
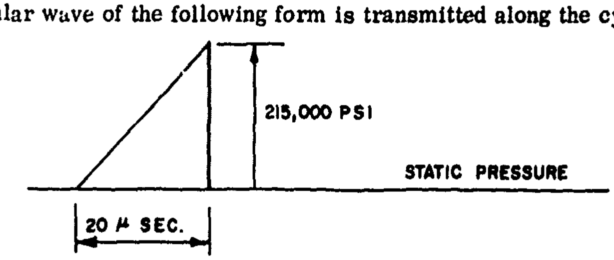


FIGURE 1

A triangular wave of the following form is transmitted along the cylinder.



This wave reflects upon arrival on a closed end. Find the resulting strains r_o = 1 inch and r_o = 3 inches.

The pressure of the reflected wave front is approximately twice the pressure of the incident wave front. This is considered the highest possible internal pressure; thus, the calculations will be performed for the reflected region only.

To illustrate the wave decay, two cases are analyzed; Case 1, where r_o = 1 inch, and Case 2, where r_o = 3 inches. Hence $P = -2\,(215,000) = -430,000$ psi at r_o = 1 inch. Using equation (23), we have

$$u_1 = -\frac{430,000\,g}{\rho_o C_o} = -2864.6 \text{ in./sec}$$

where $\rho_o = 0.29 \text{ lb/in.}^3$

$C_o = 200,000 \text{ in./sec}$

$g = 32.2 \text{ ft/sec}^2$

From equation (34), the velocity at $r_o = 3$ inches (chosen arbitrarily) is

$$u_3 = u_1 \sqrt{\frac{r_1}{r_3}} = -2,864.6 \sqrt{\frac{1}{3}} = -1654.0 \text{ in./sec}$$

Thus the pressure at $r_o = 3$ inches is given by

$$P_3 = \sqrt{\frac{1}{3}} \times P_1 = -248,280 \text{ psi}$$

A. CASE 1

1. <u>Loading Wave</u>

From equation (28) the density ratio at $\bar{t} = 0$ is given by

$$\zeta = \frac{\rho_o}{\rho} = \frac{C_o}{C_o - u} = \frac{2.00 \times 10^5}{2.00 \times 10^5 - 2864.6} = .98568$$

The maximum σ_1 at the start of plastic deformation is given by

$$\sigma_1 = \frac{\sigma_y}{1-\mu} = -\frac{250,000}{0.7} = -357,143 \text{ psi}$$

where $\sigma_y = -250,000$ psi

$1-\mu = 0.7$

Thus $\zeta = 1 - \frac{357,143}{30 \times 10^6}(1-2 \times 0.3^2) = 1 - .0119 \times .82 = .99042$

at $\sigma_y = -357,143$ psi

As soon as the material begins stretching out, the loading becomes plastic. Considering $\bar{E} = 0$ from equation (38), we have

$$\omega = \sqrt{\bar{\sigma}_y \cdot \zeta} = \sqrt{.00833 \times .98568} = \sqrt{.0082106} = .09061$$

where $\bar{\sigma}_y = \dfrac{250,000}{30 \times 10^6} = .00833$

Also, $\bar{u}_1 = \dfrac{u}{C_o} = -\dfrac{2864.6}{2 \times 10^5} = -.01432$

Substituting in equation (39), we have

$$\mathscr{E} = \cosh .09061\bar{t} - \frac{.01432}{.09061} \sinh .09061\bar{t} - 1$$

Since $\bar{u} = \dfrac{d\mathscr{E}}{d\bar{t}}$, we have

$$\bar{u} = .09061 \{\sinh .09061\bar{t} - .15800 \cosh .09061\bar{t}\}$$

First, the time $\bar{t}_o$, where $\bar{u} = 0$, is found by plotting $\bar{u}$ versus $\bar{t}$. (See Figure 2). The $\mathscr{E}$ versus $\bar{t}$ plot is shown in Figure 3: it is shown that as $\bar{u}$ approaches zero, $\mathscr{E}$ approaches a constant value of 0.0125. The radial strain is given by $\epsilon_1 = \zeta \dfrac{r_o}{r} - 1$: The ϵ_1 versus $\bar{t}$ plot is shown in Figure 4: ϵ_1 approaches the value of .0264.

2. Unloading Wave

The duration of the triangular wave is 20 μsec: its equivalent rectangular profile will have a duration of 10 μsec. Hence,

$$\bar{t} = \frac{C\ t}{r_o} = \frac{2.0 \times 10^5 \times 10 \times 10^{-6}}{1} = 2.0$$

It is shown in Figure 2, however, that the loading process terminates at $\bar{t} = 1.75$. Thus, when the unloading wave occurs, the material is in equilibrium.

For the unloading wave, we have the following constants (equation (53)):

$\bar{E} = 1$

$\zeta = 1$: The material returns to its initial volume.

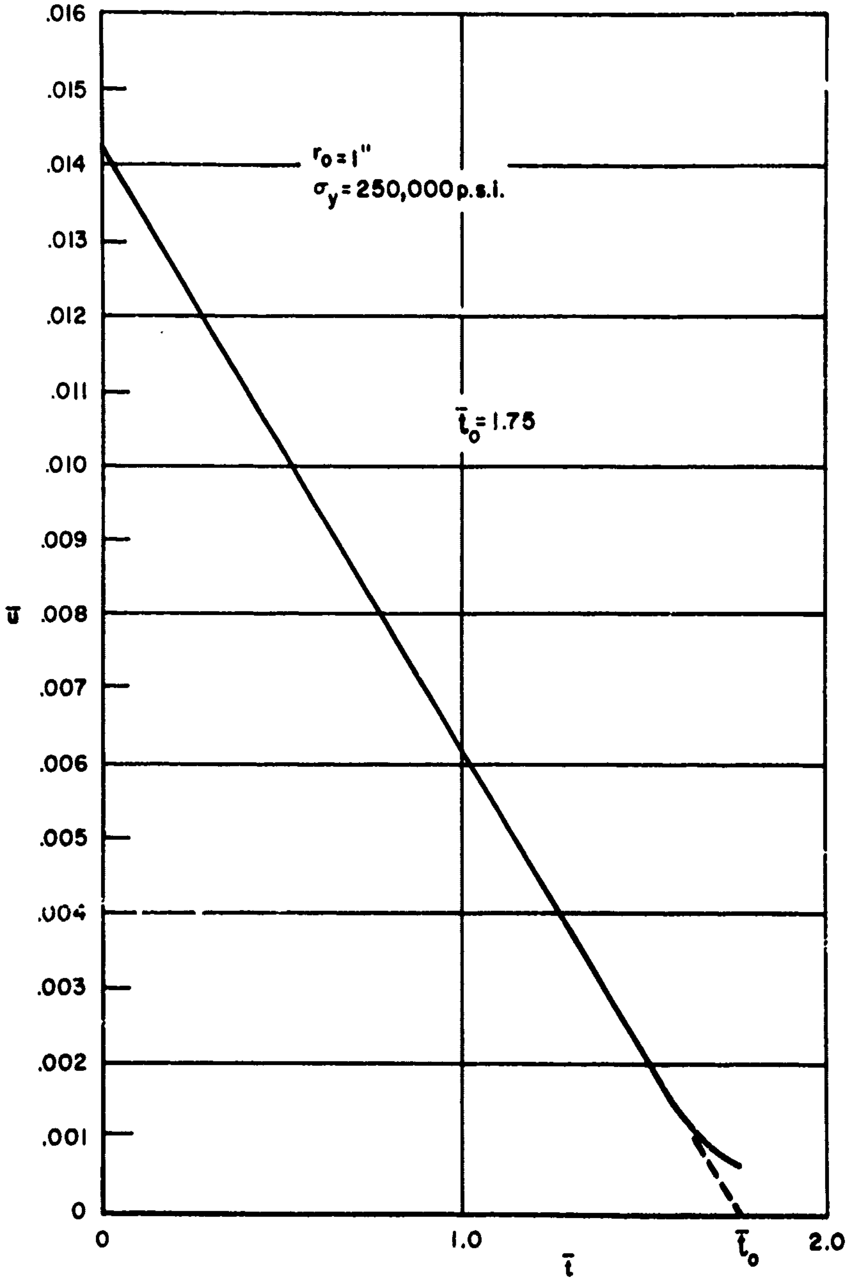

.016
.015
.014
.013
.012
.011
.010
.009
.008
.007
.006
.005
.004
.003
.002
.001
0
$\bar{u}$
$r_0 = 1''$
$\sigma_y = 250{,}000$ p.s.i.
$\bar{t}_0 = 1.75$
0
1.0
2.0
$\bar{t}_0$
$\bar{t}$

FIGURE 2

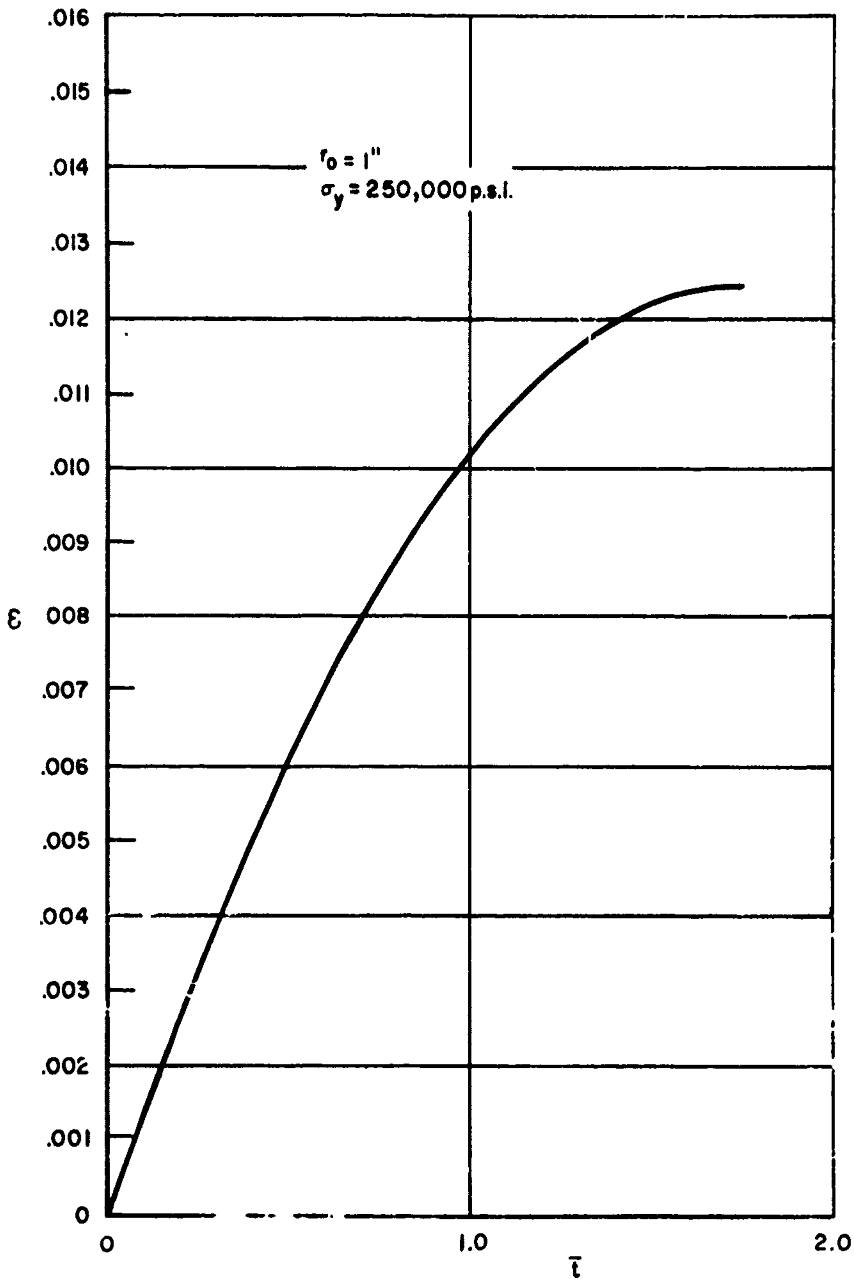

$r_0 = 1''$
$\sigma_y = 250{,}000$ p.s.i.
.016
.015
.014
.013
.012
.011
.010
.009
ε
.008
.007
.006
.005
.004
.003
.002
.001
0
0
1.0
2.0
$\bar{t}$

FIGURE 3

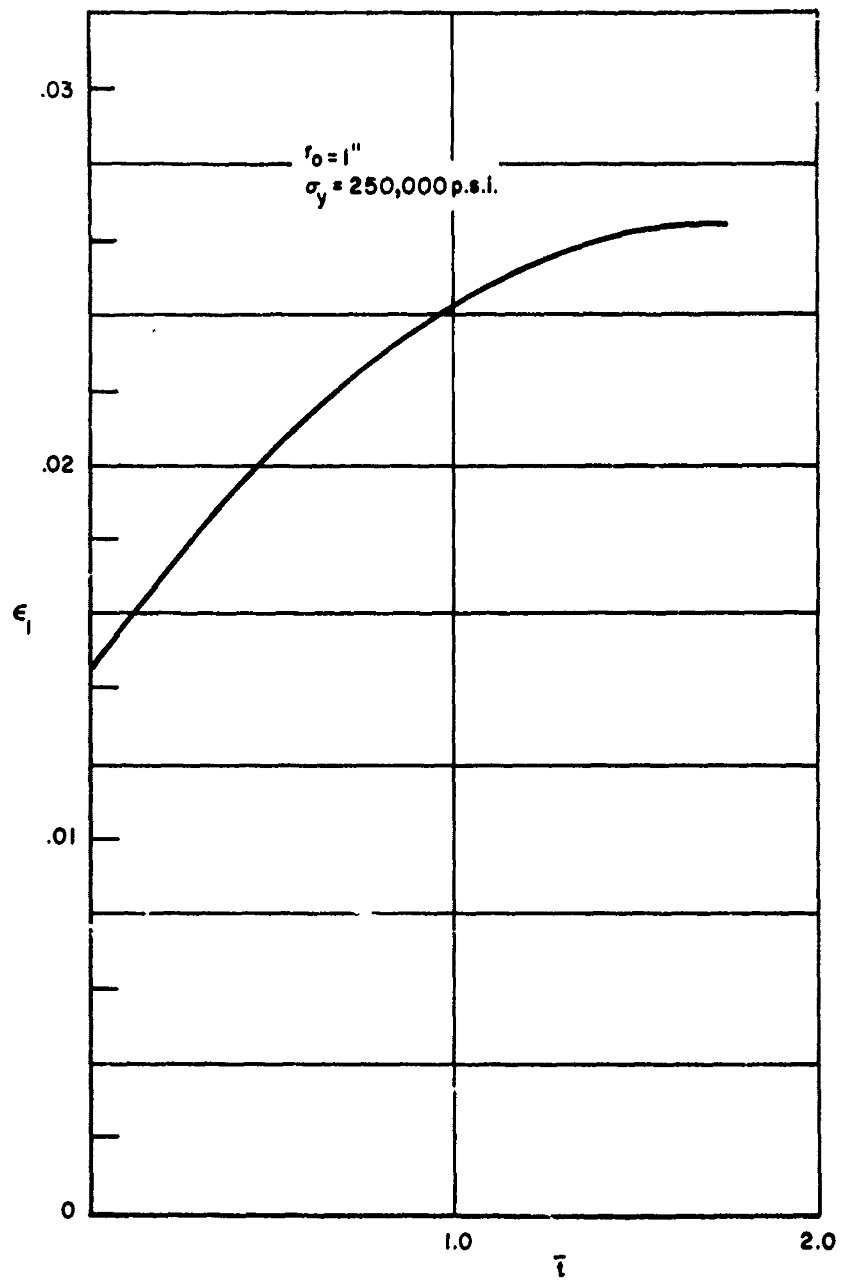


FIGURE 4

Hence $\alpha = \frac{1}{2} + \mu = 0.5 + 0.303 = .803$

where $\mu = 0.303$. N.B. The value of σ_1 after unloading is zero

$\beta = 1 - \mu = 0.697$

$\gamma = 0.303$

$u_o = .01432$

$\xi = -1.0000 - \bar{u}_o = -1.0000 - .01432 = -1.01432$

$\psi = \sqrt{\frac{\alpha^2}{4} + 4\beta} = 1.71697$

$\frac{\psi}{2} = .8585$

Thus $\frac{\gamma}{\beta} = \frac{.303}{0.697} = .43472$ and $\frac{\alpha}{2} = 0.4015$

Hence $\xi + \frac{\alpha}{2}(\frac{\gamma}{\beta}) = -1.01432 + 0.4015(.43472)$

$= -1.01432 + .17454 = -.83978$

and $Y = \dfrac{\xi + \frac{\alpha}{2}(\frac{\gamma}{\beta})}{\frac{\psi}{2}} = -\frac{.83978}{.8585} = -.978195$

Also, $\frac{\psi}{2} \times \frac{\gamma}{\beta} = .8585 \times .43472 = .37321$

$\frac{1}{2} \times \frac{\gamma}{\beta} = .43472 \times .5 = .21736$

$\frac{\alpha}{2} Y = -.4015 \times .978195 = -.392745$

$\frac{\gamma}{\beta} \times \frac{\alpha}{2} = .43472 \times .4015 = .17454$

$\frac{Y}{2} = -\frac{.978195}{2} = -.489097$

and $\frac{1}{2}\frac{\gamma}{\beta} - 1 = -1 + .21736 = -.78264$

Hence $\bar{u} = .27686 \sinh .8585\bar{t} - .79696 \cosh .8585\bar{t} + .78264 e^{.4015\bar{t}}$

and $\mathscr{E} = e^{-.4015\bar{t}} \{.43472 \cosh .8585\bar{t} - 1.01432 \sinh .8585\bar{t}\} - .43472$

In the plot of $\bar{u}$ versus $\bar{t}$, Figure 5, the time when the velocity becomes zero is shown as $\bar{t}_o = 0.026$ ($t = .26 \times 10^{-6}$ sec). Substituting in the equation of $\mathscr{E}$ we find the residual hoop strain, $\mathscr{E} = .01713$. Equations (51) and (52) were derived, however, with $\mathscr{E}_o = 0$ at $\bar{t} = 0$ (with reference to the equilibrium conditions after the loading).

To find the residual hoop strain after unloading, we have:

$$\Delta \mathscr{E} = \mathscr{E}_{unloading} - \mathscr{E}_{loading}$$

$$= .01713 - .0125 = \underline{\underline{.00463}} \text{ (compression)}$$

In order to find the residual hoop stress we use the principle of elastic recovery. Thus, the volumetric strain e_v is given by:

$$e_v = \epsilon_1 + \mathscr{E}_{unloading}$$

where ϵ_1 is the radial linear strain change upon relief and $\mathscr{E}$ is the corresponding hoop strain. We know, however, the radial unloading stress: i.e., $\Delta\sigma_1 = 430,000$ psi. Hence, $\epsilon_1 = \frac{430,000}{30 \times 10^6} = .0143$.

The value of e_v is also known from the loading conditions:

$$-e_v = \frac{\Delta V}{V_o} = -\frac{\rho}{\rho_o} + 1 = +.01813$$

Thus, the hoop strain for zero stress (permanent set) is given by

$$\mathscr{E}_o = e_v - \epsilon_1 = .01813 - .0143 = \underline{\underline{.0036}}.$$

The corresponding permanent set in the radial direction is $\epsilon_o = .026 - .0143 = \underline{\underline{.012}}$. In the calculations for the unloading wave, it was assumed that upon relief the recovery was instantaneous: thus $\frac{r}{r_o} = 1$ at $t = 0$.

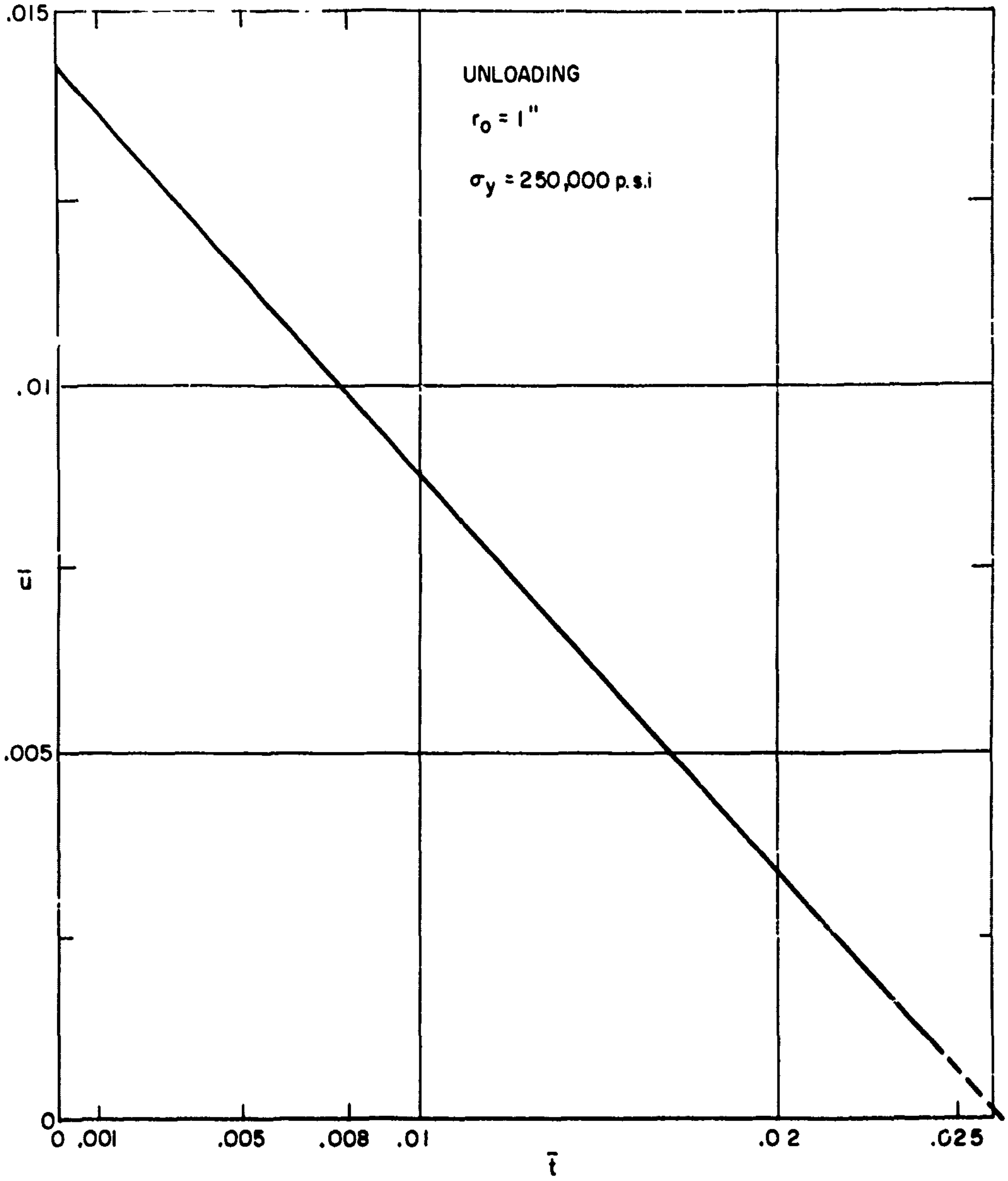
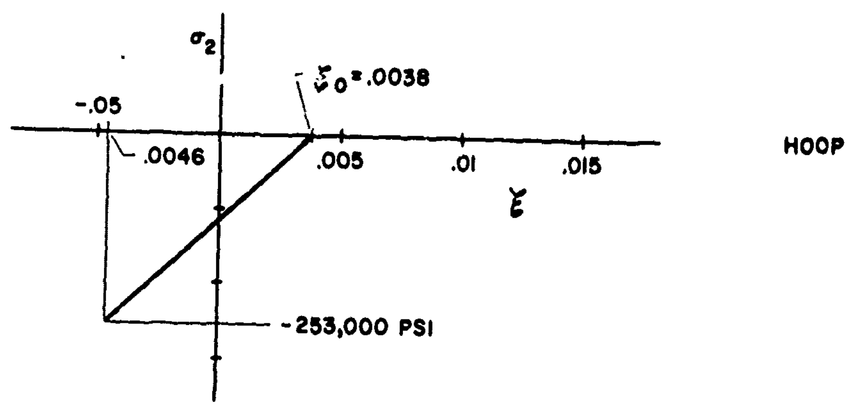

UNLOADING
$r_0 = 1''$
$\sigma_y = 250{,}000$ p.s.i
.015
.01
.005
0
$\bar{u}$
0 .001
.005
.008
.01
.02
.025
$\bar{t}$

FIGURE 5

Then, for $\bar{t} > 0$, $\frac{\bar{r}}{r_o}$ decreases, changing both hoop and radial strains according to to the following equations:

$$\xi = 1 - \frac{r}{r_o} \quad \text{and} \quad \epsilon_1 = \zeta \frac{r_o}{r} - 1.$$

Hence, the material will be under the following strain conditions :



Without pursuing this case any further, it is easily concluded that secondary yielding will take place. This is undesirable for most designs.

B. CASE 2

1. Loading Wave

The velocity at $r_o = 3$ inches was given as:

$$u_3 = -1654.0 \text{ and } \bar{u}_3 = \bar{u}_o = -\frac{1654}{2\times10^5} = -.00827$$

$$\zeta = \frac{C_o - u}{C_o} = 1 - \bar{u}_o = .99173, \text{ and } \sigma_{1(\text{elastic})} = \frac{250,000}{0.7} = 357,143 \text{ psi}$$

Hence, $\omega = \sqrt{.00833 \times .99173} = \sqrt{.0082611} = .0908906$

$$\therefore \quad \frac{u_o}{\omega} = - \frac{.00827}{.0908906} = - .0909885$$

Hence $\quad \mathscr{E} = \cosh .0908906\bar{t} - .0909885 \sinh .0908906\bar{t} - 1$

and $\quad \bar{u} = .0908906 \{\sinh .0908906\bar{t} - .0909885 \cosh .0908906 \}$

The duration of the wave, on the other hand, is given by

$$\bar{t} = \frac{10 \times 10^{-6} \times 2 \times 10^5}{3} = \frac{2}{3}$$

where

$$t = 10 \times 10^{-6}$$

$$C_o = 2 \times 10^5$$

$$r_o = 3 \text{ inches}$$

The $\bar{u}$ versus $\bar{t}$ and $\mathscr{E}$ versus $\bar{t}$ plots are shown in Figures 6 and 7, respectively. Another case of $\sigma_y = 150,000$ psi is also shown for comparison.

With the value of $\bar{t}$ at $r_o = 3$ inches being smaller than at $r_o = 1$ inch, it is shown that the unloading wave occurs before the material achieves equilibrium.

2. Unloading Wave

Following the same procedure as for $r_o = 3$ inches, with

$$\bar{u}_o = 0.00827 - .00275^* = .00552$$

$$\mathscr{E} = e^{-.4015\bar{t}} \{.43472 \cosh .8585\bar{t} - .96794 \sinh .8585\bar{t} \} - .43472$$

and $\quad \bar{u} = - .27793 \sinh .8585\bar{t} + .78816 \cosh .8585\bar{t} - .78264 e^{.4015\bar{t}}$

The plot of $\bar{u}$ versus $\bar{t}$ is shown in Figure 8. Comparing this unloading with that of $r_o = 1$ inch, we see that, at $r_o = 3$ inches, equilibrium is achieved in about half the time.

---

* velocity from the loading process

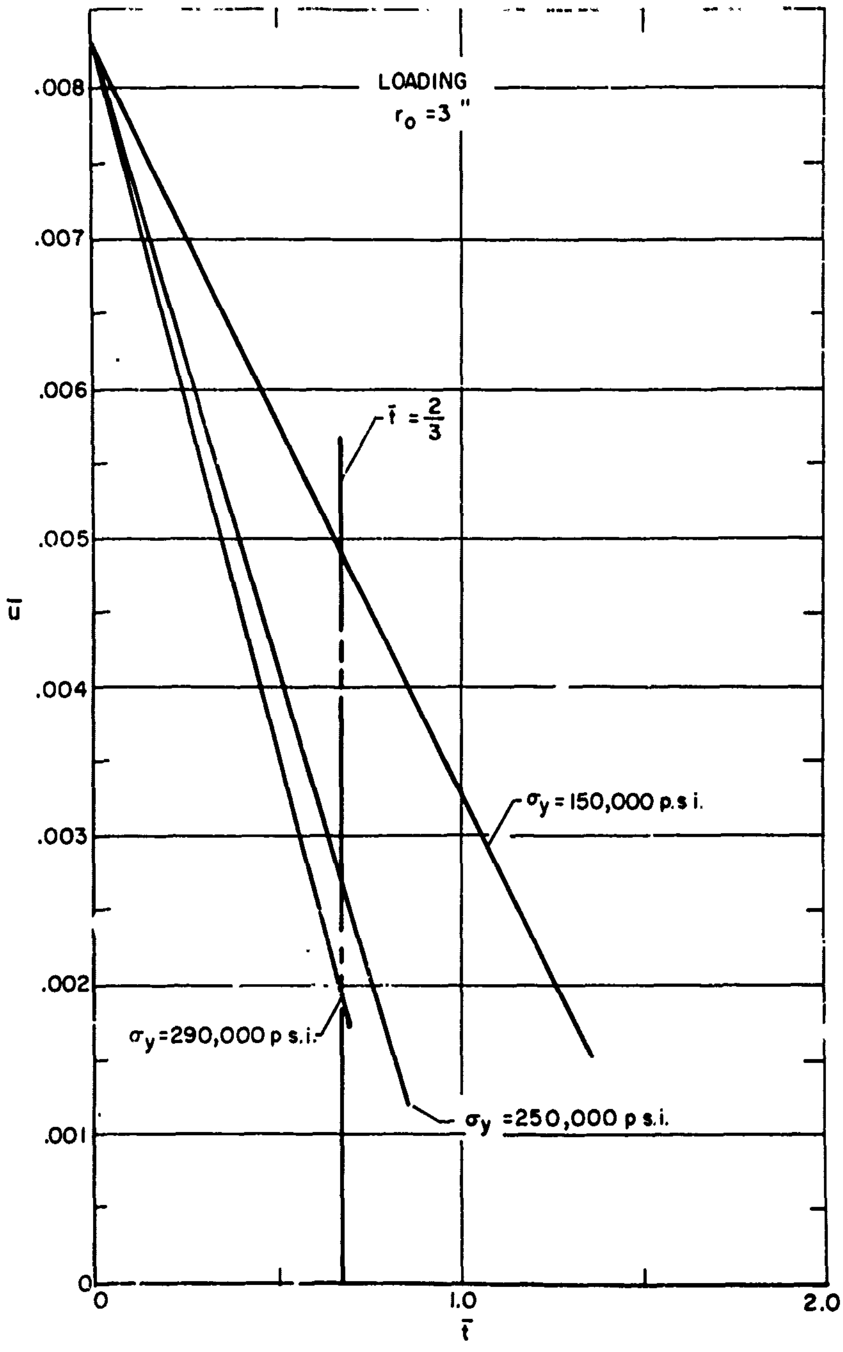

LOADING
$r_o = 3''$
.008
.007
.006
.005
.004
.003
.002
.001
0
$\bar{u}$
$\bar{t} = \frac{2}{3}$
$\sigma_y = 150{,}000$ p.s.i.
$\sigma_y = 290{,}000$ p.s.i.
$\sigma_y = 250{,}000$ p.s.i.
0
1.0
2.0
$\bar{t}$

FIGURE 6

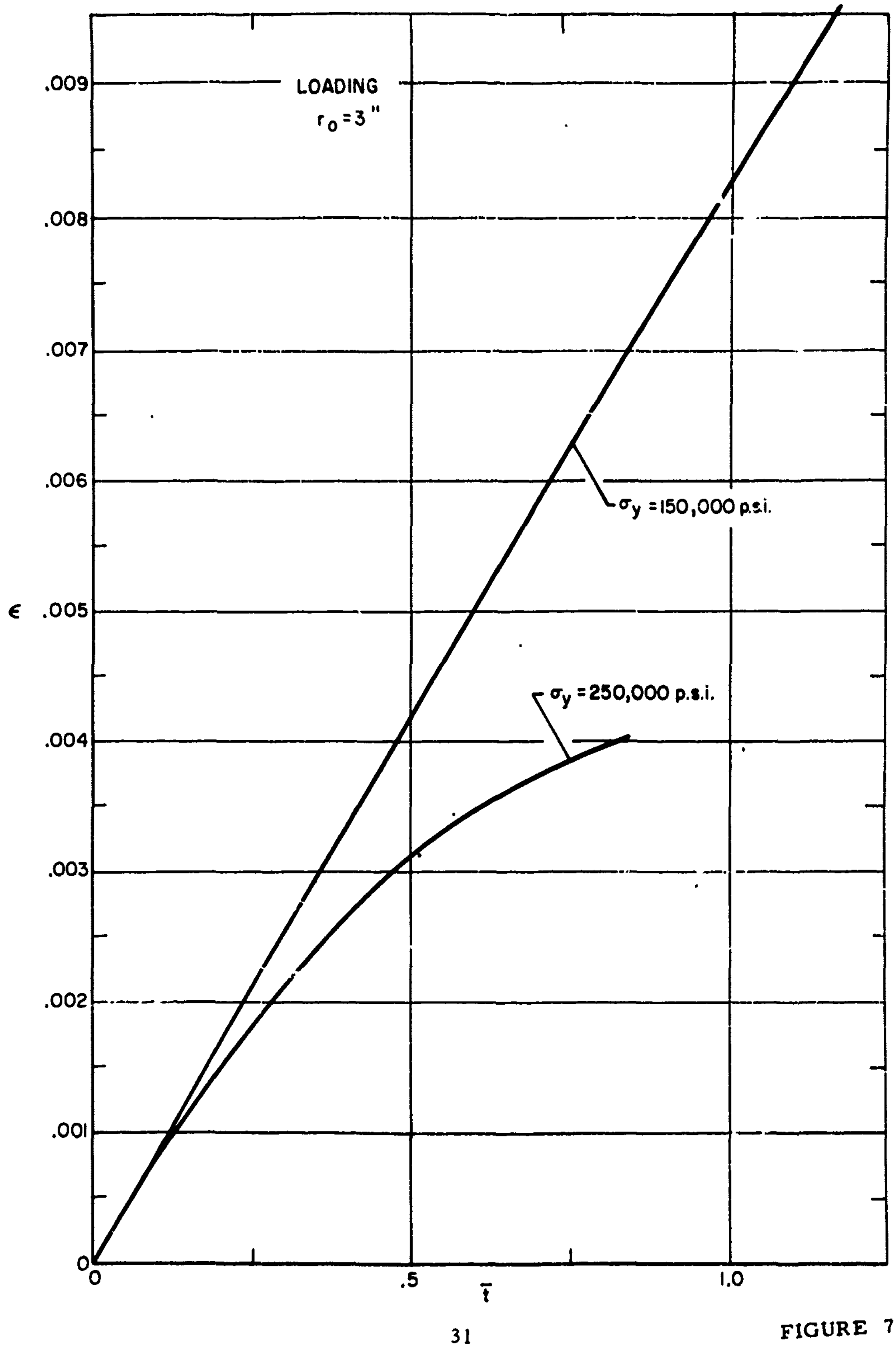

LOADING
$r_0 = 3''$
$\sigma_y = 150,000$ p.s.i.
$\sigma_y = 250,000$ p.s.i.
ϵ
.009
.008
.007
.006
.005
.004
.003
.002
.001
0
0
.5
1.0
$\bar{t}$

FIGURE 7

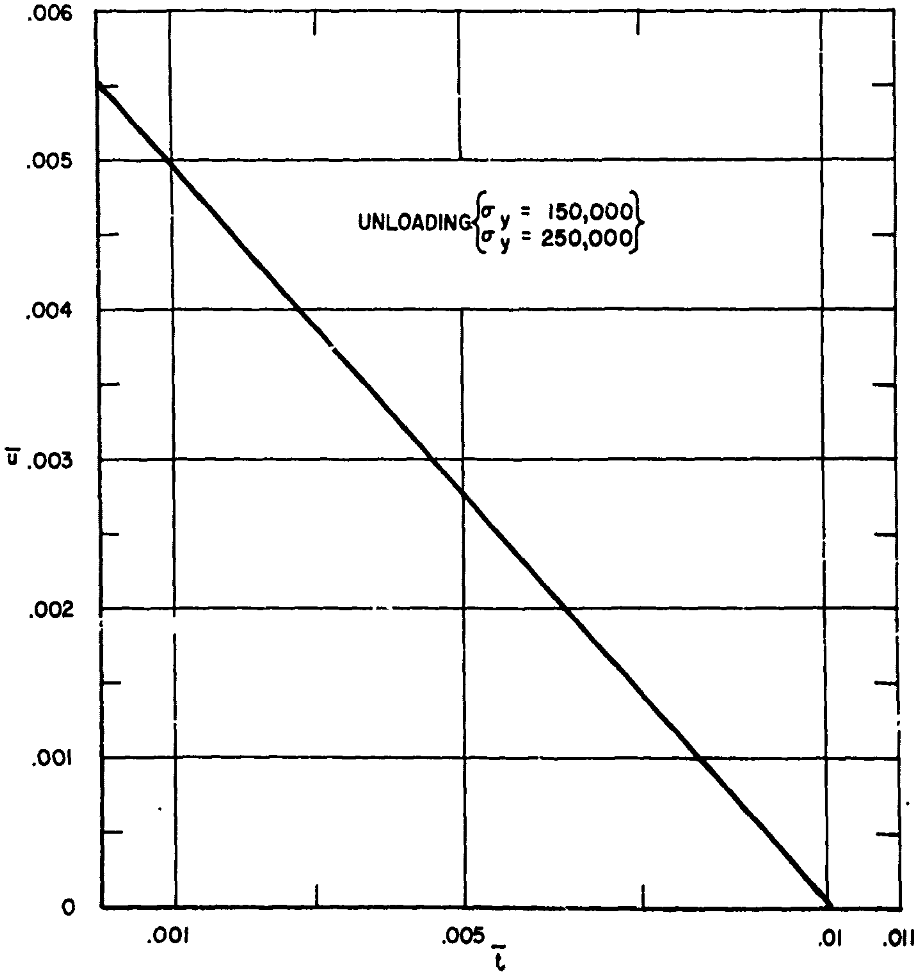

.006
.005
.004
$\bar{u}$ .003
.002
.001
0
UNLOADING $\begin{Bmatrix} \sigma_y = 150{,}000 \\ \sigma_y = 250{,}000 \end{Bmatrix}$
.001
.005
$\bar{t}$
.01
.011

FIGURE 8

The corresponding value of $\mathscr{E}$ at $\bar{t} = .01$ is $\mathscr{E} = .00727$. The $\mathscr{E}$ value at $\bar{t} = 0.667$ for loading is $\mathscr{E} = .00365$. Hence, the resulting strain is (.00733 - .00365) or $\mathscr{E} = \underline{.00368}$ (compression).

To find the resulting stresses, the permanent set in both the hoop and radial directions must be found. Employing the concept of elastic recovery again, we have

$$e_v = \mathscr{E}_o + \epsilon_1$$

$$\epsilon_1 = \frac{248,280}{30 \times 10^6} = .008276$$

Furthermore, $-e_v = +\dfrac{\Delta V}{V_o} = -\dfrac{\rho}{\rho_o} + 1 = -.99173 + 1 = .00827.$

Thus, $\mathscr{E}_o = 0$: This result indicates that there is no permanent set in the hoop direction. In the radial direction, $\dfrac{r}{r_o} = 1 + \mathscr{E} = 1 + .00365 = 1.00365$ from loading. Hence $\dfrac{r_o}{r} = \dfrac{1}{1.00365} = .99636$. Also, $\dfrac{\rho_o}{\rho} = .99173$.

$\therefore \quad \epsilon_1 = .99636 \times .99173 - 1 = .98812 - 1 = -.01188$ from loading.

During the unloading, there is the elastic recovery where $\epsilon_1 = .00827$: then, there is the further change of ϵ_1 due to the $\dfrac{r}{r_o}$ change.

Hence $\quad \epsilon_1 = \dfrac{r_o}{r} \cdot 1 + .00827$

$$\frac{r}{r_o} = 1 - \mathscr{E} = 1 - .00727 = .99373$$

$\therefore \quad \dfrac{r_o}{r} = 1.0063$

and $\quad \epsilon_1 = .0063 + .00827 = .01457$

Thus, change in $\epsilon_1 = .01457 - .0119 = .00265$ (tension). The value of ϵ_1 when the radial stress is zero is $-.01190 + .00327 = -.0036$.

Residual radial stress $= -.0036 + .0026 = -.001$ or $-.001 \times 30 \times 10^6 = -30,000$ psi (compression).

The residual hoop stress $= .00368 \times 30 \times 10^6 = 110,400$ psi (compression). From the secondary yielding criterion, we have

$$\sigma_2 - \sigma_1 < \sigma_y$$

or $-30,000 + 110,000 = 80,000$ psi. Since σ_y was assumed at 250,000 psi, the material will not go through secondary yielding.

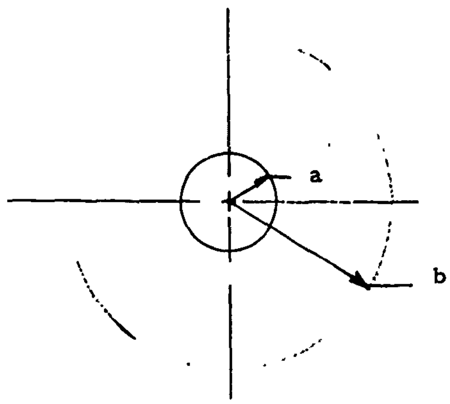
APPENDIX II

CALCULATIONS SHOWING COMBINED EFFECT OF IMPULSE AND STATIC LOADING

I. Introduction

The combined effect of the action of a transient shock wave over an internal hydrostatic pressure in thick walled cylinder is analysed in the following to verify the structural integrity of a container. The method of superposition of stress is used.

If we are given a cylinder with the following dimensions:



a = 3"
b = 15"
w = b/a = 5"
P = internal pressure (200,000 psi

For the purpose of discussion, we define the following variables at an internal radius of $r_o = a = 3$ inches:

σ_1 = hoop stress under internal hydrostatic pressure alone

σ_2 = radial stress under internal hydrostatic pressure alone

σ_3 = residual hoop stress in bore at zero pressure

σ_4 = residual hoop stress resulting from an impulse

$\sigma_5 = \sigma_3 + \sigma_4$ = resultant hoop stress after release of internal pressure due to passage of first maximum shock impulse

$\sigma_6 = \sigma_1 + \sigma_3 + \sigma_4$ = combined hoop stresses under static pressure after passage of impulses

II Calculation for Stress from Internal Pressure Alone

The equations of Lamé may be used to calculate the elastic stresses in a thick walled cylinder, internally loaded. The relationship for the hoop stress at the bore states that,

$$\sigma_1 = P \frac{w^2+1}{w^2-1}$$

Then, for a wall thickness w = 5 and an internal pressure of

$P = 200,000$ psi

$$\sigma_1 = 200,000 \times \frac{26}{24} = 217,000 \text{ psi (tension)}$$

By definition,

$$\sigma_2 = -P = -200,000 \text{ psi (compression)}$$

From the secondary yielding criterion, we have

$$\sigma y > \sigma_2 - \sigma_1$$

or -200,000 - 217,000 = -417,000 psi. Since σ_y was assumed at 250,000 psi, the material will go through secondary yielding.

However, to reduce the stress to a point under the 250,000 psi yield stress of the material, we must impose a negative hoop stress in the bore prior to loading. To bring the bore to a zero hoop stress at an interr pressure of 200,000 psi let,

$$\sigma_3 = -217,000 \text{ psi}$$

Then using the Tresca Criterion, we have

$$\sigma y > \sigma_2 - \sigma_1$$

or -200,000 + 0 = -200,000 psi. This value is suitably below the required 250,000 yield stress. Therefore, the material will not fail and we can superpose the dynamic loading to predict the final state.

III Continued Dynamic and Static Load

Previous analysis (Appendix I) has shown that the residual hoop stress σ_4 after discharge (peak pressure, P = 248,280 psi and time duration t = 20 μsec is,

$$\sigma_4 = -110,000 \text{ psi}$$

The combined hoop stresses (σ_6) under static pressure after the initial capacitor discharge will then be

$$\sigma_6 = (\sigma_1 + \sigma_3) + \sigma_4$$

$$= 217,000 - 217,000 + (-110,000)$$

$$= -110,000 \text{ psi (compression)}$$

the radial stress,

$$\sigma_2 = -P = 200,000 \text{ psi (compression)}$$

From Tresca's Criterion, we have,

$$\sigma_y > \sigma_2 - \sigma_1$$

or -200,000 - (-110,000) = -90,000 psi. Since σ_y is again 250,000 psi yielding will not occur in the bore from the combined stresses due to dynamic and static forces.

IV Release of Pressure

Upon the release of the static pressure, the permanent hoop stress imposed by the first discharge event will reverse by the amount of σ_1. Then the residual stress with internal pressure P = 0 will be,

$$\sigma_6 = \sigma_3 + \sigma_4$$
$$= -217,000 - 110,000$$
$$= -317,000 \text{ psi (compression)}$$

Since σ_6 exceeds the 250,000 psi yield strength of the material under the condition of no internal pressure, the bore will yield radially inward. This is an unsatisfactory stress condition for cylinder design.

V Solution of the Problem

Because yielding has occurred at the bore with an initial hoop stress, σ_3 = 217,000 psi (compression) a σ_3 must be found that will allow elastic behavior of the material within the 250,000 psi yield strength wall material being considered.

Let us calculate a new residual hoop stress σ_3 in the bore at no pressure that will allow the desired elastic action within the limits of the 250,000 psi yield strength material. Then

$$\sigma_8 = \sigma_1 + \sigma_3 + \sigma_4$$

$$240,000 = 217,000 + \sigma_3 + (-110,000)$$

$$\sigma_3 = -133,000 \text{ psi.}$$

Therefore, the desired value of pressures at the bore σ_3 = 133,000 psi (compression)

The combined stress (σ_8) at a pressure P = 200,000 after a maximum energy impulse is then

$$\sigma_8 = \sigma_1 + \sigma_3 + \sigma_4$$

$$= 217,000 + (-133,000) + (-110,000)$$

$$= -26,000 \text{ psi (compression)}$$

$$\sigma_2 = -P = -200,000\text{psi (compression)}$$

Then using the Tresca Criterion,

$$\sigma_y > \sigma_2 - \sigma_1$$

or -200,000 -(-26,000) = -174,000. With the σ_y = 250,000 psi material, yielding will not occur in the bore.

To f .d the residual hoop stress upon release of the 200.000 psi static pressure P goes to zero, and the hoop stress is diminished by the amount σ_1 = 217,000 psi. Then

$$\sigma_5 = \sigma_3 + \sigma_4$$

$$\sigma_5 = 133,000 + (-110,000)$$

$$\sigma_5 = 243,000 \text{ psi}$$

Therefore the bore diameter at r_o = a = 3 inches has attained a residual hoop stress (compression) of 243,000 psi which is within the required 250,000 psi yield strength of the material. The pressure vessel can be operated safely in the prescribed limits of 200,000 psi static pressure and a peak shock pulse of 248,280 psi for 20 μ duration.

DISTRIBUTION LIST

ASD (ASRCT-10)
Wright-Patterson AFB, Ohio (5)

ASD (ASW, Mr. Hazen A. Miles)
Wright-Patterson AFB, Ohio

ASD (ASOP, Mr. J.M. Troyan)
Wright-Patterson AFB, Ohio (2)

ASD (ASNPFP, Mr. R. Farrington)
Wright-Patterson AFB, Ohio

ASD (ASRCE-31A, Miss B. Parker)
Wright-Patterson AFB, Ohio

AFSC (SCR-2, Mr. C. W. Kniffin)
Andrews AFB
Washington 25, D.C.

Allegheny Ludlum Steel Corp.
Research Center
Attn: Mr. R.K. Pitler
Chief Rsch. Metallurgist
High Temperature Alloys
Brackenridge, Pennsylvania

Aluminum Company of America
ALCOA Building
Attn: Mr. R. W. Andrews, Jr.
Pittsburgh 19, Pennsylvania

Baldwin-Lima-Hamilton Corp.
Press Department
Attn: Dr. F. J. Kent
Engineering Manager
Philadelphia 42, Penna.

Battelle Memorial Institute
Metal Working Division
Attn: Mr. A. M. Sabroff
Assistant Chief
505 King Avenue
Columbus 1, Ohio

Bureau of Naval Weapons (CS-23)
Navy Department
Attn: Mr. S.E. Sanfillippo
Washington 25, D.C.

ASD (ASRCM-34, Mr. S. Inouye)
Wright-Patterson AFB, Ohio

ASD (ASRCM-10, Mr. W.G. Ramke)
Wright-Patterson AFB, Ohio

ASD (ASRCE, Mr. J. Teres)
Wright-Patterson AFB, Ohio

ASD (ASRC, Dr. A.M. Lovelace)
Wright-Patterson AFB, Ohio

FTD (TD-E2b)
Wright-Patterson AFB, Ohio

Aerospace Industries Association
of America, Inc.
Attn: Mr. S.D. Daniels
Director of Technical Services
1725 De Sales Street, NW
Washington 6, D.C.

Allegheny Ludlum Steel Corporation
Attn: Extrusion Plant
Watervliet, New York

Defense Documentation Center (DDC)
Cameron Station
Alexandria, Virginia, 22314 (20)

Babcock & Wilcox Company
Attn: Mr. James Barrett
Beaver Falls, Pennsylvania

Barogenics, Inc.
Attn: Mr. Alexander Zeitlin
Vice President
51 East 42nd Street
New York 17, N.Y.

Bureau of Naval Weapons (PID-2)
Navy Department
Washington 25. D.C.

California Institute of Technology
Jet Propulsion Laboratory
Attn: Mr. I. W. Newlan
4800 Oak Grove Drive
Pasadena 3, California

IIT Research Institute
Metals Research Department
Attn: Mr. Frank A. Crosely
3350 South Federal Street
Chicago 16, Illinois

National Academy of Science
National Research Council
Div. of Engineering & Industrial Resources
Attn: Mr. E. V. Bennett
Washington 25, D.C.

NORAIR Division
Northrop Corporation
Attn: Mr. J. A. Van Hammersveld
1001 East Broadway
Hawthorne, California

Nuclear Metals, Inc.
Attn: Mr. Klein, Vice President
West Concord, Massachusetts

North American Aviation, Inc.
Attn: Plant Engineering Library
43 East Fifth Avenue
Columbus 16, Ohio

Oregon Metallurgical Corp.
Attn: Mr. Frank Vandenburgh
Vice President
P.O. Box 484
Albany, Oregon

Pressure Technology Corp. of Amer.
Attn: Mr. Alfred Bobrowski, Pres.
453 Amboy Avenue
Woodbridge, New Jersey 07095

Republic Aviation Corporation
Attn: Mr. A. Kastelowitz
Director Manufacturing Research
Farmingdale, L.I., N.Y.

Reynolds Metals Company
918 16th Street, N.W.
Washington 6, D.C.

Lombard Corporation
Youngstown, Ohio

P.R. Mallory & Company, Inc.
Attn: Mr. A.S. Doty, Director
Technical Services Laboratories
Indianapolis 6, Indiana

Curtiss-Wright Corporation
Metals Processing Division
Attn: Mr. A.D. Roubloff
Chief Engineer
760 Northland Avenue
P.O. Box 13
Buffalo 15, New York

National Aeronautics & Space Admin.
Lewis Research Center
Attn: Mr. George Mandal, Chief, Library
21000 Brookpark Road
Cleveland 25, Ohio

North American Aviation, Inc.
Attn: Mr. Walter Rhineschild
International Airport
Los Angeles 9, California

Oak Ridge National Laboratory
Metals & Ceramics Division
Attn: Mr. C. W. Thurber
P.O. Box X
Oak Ridge, Tennessee

Reactive Metals Products
Attn: Mr. G. W. Cleveland
Sales Engineer
Niles, Ohio

Republic Steel Corporation
Research Center
Attn: Mr. R. W. Kollar
6801 Breckville Road
Cleveland 31, Ohio

Union Carbide Nuclear Company
Attn: Mr. Paul E. Wilkinson
P.O. Box Y
Oak Ridge, Tennessee

The Cleveland Twist Drill Company
Attn: Mr. Robert P. Lesher
Development Engineer
P.O. Box 6656
Cleveland 1, Ohio

Climax Molybdenum Co. of Mich.
Refractory Metal Division
14410 Woodrow Wilson Blvd.
Detroit 38, Michigan

Commanding Officer
Watertown Arsenal Laboratories
Attn: Mr. S.V. Arnold
Associate Director
Watertown 72, Massachusetts

Mr. James W. Conrad
Technical Consultant
1526 Denniston Avenue
Pittsburgh 17, Pennsylvania

General Dynamics/Convair
Attn: Mr. J.H. Famme, Director
Manufacturing Development
P.O. Box 1950
San Diego 12, California

General Electric Company
Attn: Mr. J.K. Keeler
Manager, Engineering
21800 Tungsten Road
Cleveland 17, Ohio

Harvey Aluminum, Inc.
Attn: Mr. C.A. Moudry
Technical Director
19200 South Western Avenue
Torrance, California

Jones & Laughlin Steel Corp.
Attn: Mr. Robert S. Orr
Commercial Research Librarian
3 Gateway Center
Pittsburgh 30, Pennsylvania

Canton Drop Forging & Mfg. Co.
Attn: Mr. Chandis Brauchler, Pres.
2100 Wilett Avenue
Canton, Ohio

Dow Chemical Company
Metallurgical Laboratory
Attn: Dr. T.E. Leontis
Assistant to the Director
Midland, Michigan

E.I. du Pont de Nemours & Co., Inc.
Pigments Department - Metal Products
Attn: Dr. E.M. Mahla
Technical Asst. to Director-Metal Prod.
Wilmington 98, Delaware

Crucible Steel Company of America
Attn: Dr. Walter Finlay
Assistant Vice President-Technology
P.O. Box 88
Pittsburgh 30, Pennsylvania

General Dynamics/Astronautics
Attn: Mr. V.C. Mellquist
Chief of Applied Manufacturing
Research & Process Development
P.O. Box 1128 (Zone 290-00)
San Diego 12, California

General Electric Company
Aircraft Gas Turbine Division
Attn: Mr. C.J. Wile
Engineering Manager
Metallurgical Engineering Operations
Cincinnati 15, Ohio

H.M Harper Company
Lehigh Avenue & Oakton Street
Morton Grove, Illinois

Union Carbide Stellite Company
Kokomo, Indiana

Kawecki Chemical Company
Attn: Mr. Edwin V. Bielecki
Research Manager
Boyertown, Pennsylvania

Ling-Temco-Vought, Inc.
Attn: Library 1-63101
P.O. Box 5907
Dallas 22, Texas

Stanford Research Institute
Department of Metallurgy
Menlo Park, California

Superior Tube Company
Development Metallurgy Section
P.O. Box 191
Norristown, Pennsylvania

Universal-Cyclops Steel Corp.
Refractomet Division
Attn: Mr. Charles Mueller
General Manager
Bridgeville, Pennsylvania

Vanadium Corporation of America
Attn: Mr. C.N. Cosman
Metallurgical Engineer
Graybar Building
420 Lexington Avenue
New York 14, N.Y.

Westinghouse Electric Corporation
Materials Manufacturing Dept.
Attn: Mr. F.L. Orell
Section Manager, Develop. Contracts
Blarisville, Pennsylvania

Wyman-Gordon Company
Attn: Mr. Arnold Rustay
Vice President & Technical Dir.
Grafton Plant
Worcester Street
North Grafton, Massachusetts

Stauffer Metals Company
1201 South 47th Street
Richmond, California

United States Steel Corporation
Products Development Division
525 William Penn Place
Pittsburgh, Pennsylvania

U.S. Atomic Energy Commission
Technical Information Services Extension
Attn: Mr. Hugh Voress
P.O. Box 62
Oak Ridge, Tennessee

Wah Chang Corporation
Attn: Mabel E. Russell, Librarian
P.O. Box 366
Albany, Oregon

Wah Chang Corporation
Attn: Mr. K.C. Li, Jr.
100 Church Street
New York 7, New York

Wolverine Tube
Division of Calumet & Hecla, Inc.
Attn: Mr. F.C. Eddens, Manager
Special Metals, New Products Division
17200 Southfield Road
Allen Park, Michigan

National Forge Company
Attn: Mr. James R. Becker
Project Engineer
Press Form Products
Irvine, Warren County, Penna.

AD

Republic Aviation Corporation
Manufacturing Research
Farmingdale, New York

ELECTRO SPARK EXTRUDING
by: G. Pfanner, J. H. Wagner,
April 1964 pages
Project 8-111 Third Interim Report
(Unclassified) Contract AF33(657)-11265

Unclassified Report

UNCLASSIFIED

1. Electric Discharges Applications
2. Shock Waves Applications
3. Metals - Deformation
4. Capacitors Applications

UNCLASSIFIED

AD

Republic Aviation Corporation
Manufacturing Research
Farmingdale, New York

ELECTRO SPARK EXTRUDING
by: G. Pfanner, J. H. Wagner,
April 1964 pages
Project 8-111 Third Interim Report
(Unclassified) Contract AF33(657)-11265

Unclassified Report

UNCLASSIFIED

1. Electric Discharges Applications
2. Shock Waves Applications
3. Metals - Deformation
4. Capacitors Applications

UNCLASSIFIED

AD

Republic Aviation Corporation
Manufacturing Research
Farmingdale, New York

ELECTRO SPARK EXTRUDING
by G. Pfanner, J. H. Wagner.
April 1964 pages
Project 8-111 Third Interim Report
(Unclassified) Contract AF33(657)-11265

Unclassified Report

UNCLASSIFIED

1. Electric Discharges Applications
2. Shock Waves Applications
3 Metals - Deformation
4. Capacitors - Applications

UNCLASSIFIED

AD

Republic Aviation Corporation
Manufacturing Research
Farmingdale, New York

ELECTRO SPARK EXTRUDING
by: G. Pfanner, J. H. Wagner,
April 1964 pages
Project 8-111 Third Interim Report
(Unclassified) Contract AF33(657)-11265

Unclassified Report

UNCLASSIFIED

1. Electric Discharges Applications
2. Shock Waves Applications
3. Metals - Deformation
4. Capacitors - Applications

UNCLASSIFIED

AD

The work in this report presents both static and dynamic considerations as applied to the design of a thick walled extrusion device. A design concept is derived and checked.

UNCLASSIFIED

I Pfanner, G
II Wagner, J. H.
III Republic Aviation Corp.
IV Contract AF33 (657)11265
V ASD Project 8-111
VI Manufacturing Technology Division

UNCLASSIFIED

AD

The work in this report presents both static and dynamic considerations as applied to the design of a thick walled extrusion device. A design concept is derived and checked.

UNCLASSIFIED

I Pfanner, G
II Wagner, J. H.
III Republic Aviation Corp.
IV Contract AF33 (657)11265
V ASD Project 8-111
VI Manufacturing Technology Division

UNCLASSIFIED

AD

The work in this report presents both static and dynamic considerations as applied to the design of a thick walled extrusion device. A design concept is derived and checked.

UNCLASSIFIED

I Pfanner, G
II Wagner, J. H.
III Republic Aviation Corp.
IV Contract AF33 (657)11265
V ASD Project 8-111
VI Manufacturing Technology Division

UNCLASSIFIED

AD

The work in this report presents both static and dynamic considerations as applied to the design of a thick walled extrusion device. A design concept is derived and checked.

UNCLASSIFIED

I Pfanner, G
II Wagner, J. H.
III Republic Aviation Corp.
IV Contract AF33 (657)11265
V ASD Project 8-111
VI Manufacturing Technology Division

UNCLASSIFIED