

THE ANALYSIS AND DESIGN OF A SYSTEM FOR
TESTING MATERIALS AT INTERMEDIATE
STRAIN RATES

by

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CHAPTER I

INTRODUCTION

A review of experimental work performed to study the influence of rate of strain on material properties reveals there are several experimental methods which have been employed to strain materials at high rates. Many of these methods are simply alterations of the standard tensile test method to provide a means for straining the specimen at higher than standard rates. For example, Austin and Steidel (1) developed an explosive impact tensile tester that utilized gunpowder as a source of energy. Anderson (2) developed what was called a Fast-Acting Tensile Tester that utilized high pressure nitrogen gas as a source of energy to strain simple tension specimen at high rates of strain. However, Clark and Duwez (3) were one of the first to develop an experimental method that better simulated the forming processes used by industry. Clark and Duwez based their research on the concept of using a thin walled tubular specimen. The specimen was strained circumferentially by forcing fluid into the specimen under high pressure. (See Figure 1.) Thin wall theory was then used to compute material properties. An important aspect of their experimental method, as pointed out by Clark and Duwez, is that the phenomenon of propagation of plastic strain is not present. In contrast, a tensile test specimen subjected to high strain rates will demonstrate strain rate variation from point to point along the specimen that is dependent upon time. Consequently, a pure strain condition does not exist.

The tubular specimen appeared to best satisfy the objective of uniaxial, pure strain and was adopted by Chen (4) and Giles (5) who performed the preliminary experimental work leading to the work developed in this thesis.

Giles (5) built a piston-cylinder device which utilized a universal testing machine as the source of energy to drive the piston. (See Figure 1.) Giles' objective was to devise a simpler machine than used by Clark and Duwez (3) which would function in the range of strain rates up to 25 in./in.-sec. (Clark and Duwez were limited to a minimum of approximately 40 in./in.-sec. with their testing machine.) Results of Giles' work indicated that three

improvements needed to be made in the design of his experimental testing device before suitable data could be collected. These improvements were:

1. lower the working pressure of the fluid to eliminate the problem of fluid leakage through the piston seals; (The working pressure for Giles' device approached 10,000 psi.)
2. improve the uniformity of strain rate;
3. increase the wall thickness of the specimen to something above .010 in. in order to reduce the effects of machining tolerances and material defects. This improvement was substantiated by Clark and Duwez when they commented concerning their results that, "The lack of structural uniformity between specimen having such thin walls is the principal reason for the scattered results".

This report presents the analysis, design and testing performed in developing an improved strain rate testing system. This system will incorporate the improvements outlined above as made available by the preliminary investigations of Giles.

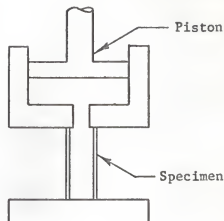


Figure 1. Pressurizing Cylinder and Specimen

CHAPTER II

ANALYSIS AND DESIGN OF TUBULAR SPECIMEN

Analysis of Specimen Strain Rate

For a standard tension specimen the average strain rate introduced into the specimen is a simple function of the rate of pulling the specimen ends apart. However, for the tubular specimen a more complicated expression for the circumferential strain rate results, which is:

$$\dot{\epsilon} = \frac{v_p}{2L_e(1+e)} \left(\frac{r_s}{r_o} \right)^2 \quad [1]$$

where $\dot{\epsilon}$ = engineering strain rate $\frac{de}{dt}$
 L_e = equivalent length of specimen
 r_o = original inside radius of specimen
 r_s = radius of piston (See Figure 1.)
 v_p = velocity of piston.

The derivation of equation [1] is given in Appendix A. Equation [1] is a function of the amount of circumferential strain the specimen has sustained. Figure 2 is a graph of equation [1] which shows, for example, that material capable of being strained as much as 20% before rupture will demonstrate a reduction in strain rate of about 15%. The piston velocity (v_p) is assumed to remain uniform. This means in the plastic region a tubular specimen will inherently demonstrate a reduction in strain rate with increasing strain. However, in the derivation of equation [1] uniform diametrical expansion from end to end of the specimen with no change in specimen length was assumed. Actually, as the specimen expands diametrically it will shorten in length. Also, the diametrical expansion will probably not be uniform from end to end

but will tend to focus at the midpoint of the specimen. Therefore, the actual reduction in strain rate at the specimen midpoint will be something less than that described by equation [1] and will have to be determined by test. Although the diametrical expansion will not necessarily be uniform from end to end this will not interfere with the pureness of circumferential strain at any particular point along the specimen length. This means specimen strain should be measured at the midpoint of the specimen where the strain rate should be maximum and nearly uniform.

Of course, in the elastic range, equation [1] reduces, for all practical purposes, to:

$$\dot{\epsilon} = \frac{v_d}{2L_e} \left(\frac{r_s}{r_o} \right)^2 \quad [2]$$

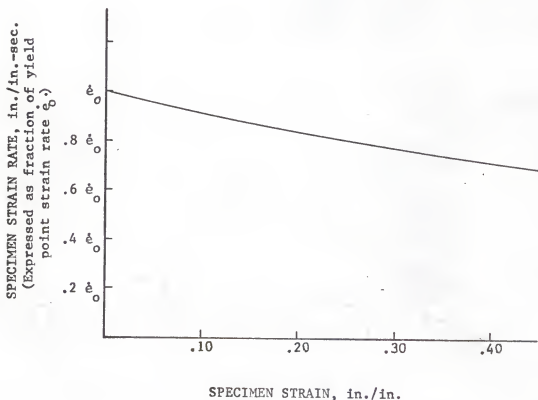


Figure 2. Comparison Between Specimen Strain Rate and Specimen Strain

That is, the amount of strain incurred in the elastic range by the specimen is so small that the strain term can be eliminated from the equation.

Analysis of Specimen Fluid Pressure

The stress induced in a simple tension specimen is directly proportional to the load applied to the specimen ends. However, the tubular specimen is subjected to a state of stress by pressurizing the fluid inside the specimen. In order to properly design a mechanism for pressurizing the fluid it was necessary to develop an expression for the fluid pressure as a function of the specimen circumferential strain. The expression is developed in Appendix B and is:

$$P = \frac{K[\ln(1+e)]^n}{\sigma_o(1+e)} P_o \quad (e_o \leq e \leq e_c) \quad [3]$$

where

- P_o = pressure of the fluid in the specimen at the yield point
- n = strain hardening exponent
- e = engineering strain
- K = strength coefficient
- σ_o = yield point strength
- P = pressure of fluid in specimen
- e_o = yield point strain ($e_o \doteq .01$)
- e_c = critical or necking strain.

Some important observations were made from the graph of equation [3] as shown in Figure 3. (A .2% carbon steel material with a yield point strength of 45,000 psi was used to develop Figure 3.) The maximum load applied to a standard tensile test specimen occurs at the point of unstable strain generally referred to as the necking point. The specimen stress at this point

is defined as the ultimate strength of the material. Dieter (7) explains in Section 9-3 of his text, that the true strain (ϵ) at the point of unstable strain is equivalent to the strain hardening exponent ($\epsilon = n$). For the material mentioned above $n = .20$. Using equation [g] of Appendix B, the engineering strain at the necking point for $\epsilon = n = .20$ is, $e = .22$. Referring to Figure 3, it was readily apparent that the maximum fluid pressure occurred at a strain point considerably less than the theoretical ultimate strength of the material. To be more specific, the results of a maxima-minima analysis performed on equation [3], for $n = .20$, showed that the engineering strain at the maximum pressure was:

$$e = e_1^{\frac{n}{2}} - 1 = e_1^{0.1} - 1 = 0.105 \text{ in./in.} \quad (e_1 = 2.718) \quad [4]$$

Equation [4] is developed in Appendix B.

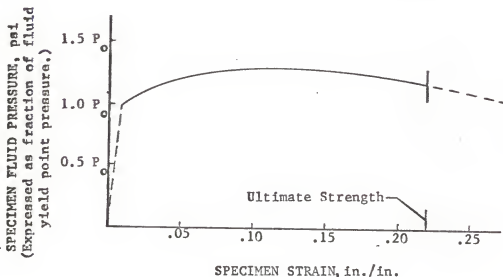


Figure 3. Comparison Between Specimen Fluid Pressure and Specimen Strain

A second observation important to the analysis of the testing system developed later in this report was that the magnitude of the maximum pressure

equaled approximately $1.28 P_o$. (P_o = yield point fluid pressure*.) This is a small increase when compared to the standard tensile test which may require a maximum load approaching twice the value of the yield point load.

Design of Specimen

Low carbon steel, aluminum and copper alloys will be the materials initially tested. Specimen will usually be tested in the annealed condition which would be the condition normally used in an industrial forming process. The yield point strength for these materials in the annealed condition ranges from 25,000 psi to 65,000 psi. Selecting 45,000 psi as a practical yield point stress for computational purposes, the expression for specimen yield point fluid pressure as a function of specimen diameter and wall thickness is:

$$P_o = 9(10^4) \frac{t}{D_o} \quad [5]$$

where P_o = specimen yield point fluid pressure

t = specimen wall thickness

D_o = specimen inside diameter.

Equation [5] is developed in Appendix C. Figure 4 is a graph of equation [5] with specimen wall thickness (t) as the parameter.

The fluid pressure limit for most dynamic seals of one piece construction is less than 10,000 psi. Therefore, an equivalent yield point fluid pressure of 2500 psi or less was considered a suitable design value for the materials mentioned above. This allowed a comfortable margin for the expected increase in material strength properties. From Figure 4 a specimen size of $7/8$ I.D. x .020 wall thickness was selected. This specimen size maintained

*The maximum fluid pressure may be found from Appendix B, equation [1].

the yield point fluid pressure in the vicinity of 2200 psi. Although the maximum fluid pressure will be greater than 2200 psi, it has already been shown that the maximum pressure is only a small fraction above the yield point fluid pressure. The 7/8 I.D. x .020 wall more than satisfied the ten to one ratio of specimen radius to wall thickness considered a minimum for the use of thin wall theory. Also, the objectives of lowering the fluid pressure and increasing the specimen wall thickness as outlined in the introduction of this thesis were obtained. The specimen dimensions and finish are specified in Appendix D.

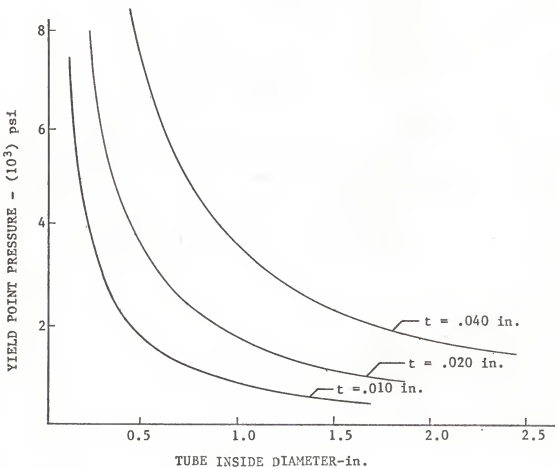


Figure 4. Comparison Between Specimen Yield Point Pressure and Specimen Diameter for Several Wall Thicknesses

CHAPTER III

ANALYSIS AND DESIGN OF TESTING SYSTEM

Proposed Testing System

The need for increasing the specimen diameter and wall thickness above the values tested by Giles (5) resulted in exceeding the load capacity and ram velocity of the 150,000 lb. universal testing machine used by Giles. Therefore, another source of power had to be utilized. Figure 5 is a schematic of the system proposed to replace the test fixture used by Giles.

High pressure gas in reservoir L subjects the large end of the double ended piston to a load. Fluid in reservoir M is allowed to flow out through the metering orifice thus restricting the travel of the double ended piston to a specific velocity. The small end of the piston acts upon a fluid in reservoir S causing it to flow into the specimen and enlarge the specimen. The rate of straining the specimen is therefore a function of the rate at which the fluid flows from reservoir M and is controlled by the size of the metering orifice. Also, the uniformity of piston velocity depends upon how uniform the metering fluid flows through the orifice which in turn depends upon the pressure variation in reservoir M during the test cycle. If reservoir L is made large enough, the pressure in reservoir L will remain

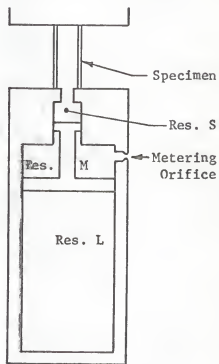


Figure 5. Proposed Test System

constant for all practical purposes. The pressure variation in reservoir M becomes dependent only upon the relative pressure difference between reservoir S and reservoir L and their respective areas A_s and A_l .

Analysis of Proposed Testing System

A third objective presented in the introduction of this report concerned uniformity of strain rate. With respect to the proposed test system shown in Figure 5, uniformity of strain rate is directly proportional to the uniformity of piston travel. Of course, this neglects rate variations which are inherent with the tubular specimen as demonstrated by equation [1]. In order to analyze the problem of uniformity of piston travel, a set of parameters related to the physical operation of the proposed system was assigned as follows:

$$\text{let } \eta = \frac{P_l}{P_o} \quad [6]$$

where P_l = pressure of the gas in reservoir L

P_o = pressure of the fluid inside the specimen at the yield point stress

$$\text{and let } \alpha = \frac{A_l}{A_s} \quad [7]$$

where A_l = area of large end of piston

A_s = area of small end of piston.

Then, the fractional difference in piston velocity experienced at some arbitrary specimen pressure (P) as compared to the piston velocity experienced at the yield point pressure was designated ψ . ψ can be expressed in terms of the system parameters η and α as:

$$\psi = 1 - \sqrt{\frac{\eta\alpha - P/P_o}{\eta\alpha - 1}} \quad [8]$$

This expression is derived in Appendix E, where:

P = pressure inside the specimen for an arbitrary amount of specimen strain.

$n\alpha$ was considered a single variable. Then the absolute value of Ψ was plotted as a function of $n\alpha$ as shown in Figure 6. The variation in specimen fluid pressure (P) is the parameter and is expressed as fractional parts of the yield point fluid pressure (P_o).

A maximum variation in piston velocity of approximately 5% was considered a suitable value during the plastic phase of straining the specimen. This meant that the strain rate variation inherently contributed by the testing system would be about 5%. To obtain a quantitative value for the system parameter $n\alpha$, the low carbon steel selected to construct Figure 3 was used as a basis for further design analysis. From Figure 3 the maximum fluid pressure during plastic straining of the specimen was found to be $1.28 P_o$. From Figure 6 for $\Psi < .05$ and $P = 1.28 P_o$, $n\alpha$ was found to be equal to or greater than 4.0.

Figure 7 is a plot of the relation $n\alpha = 4.0$. Any point in region I will satisfy the criterion of maintaining the strain rate variation to a value less than 5%. Quantitative values for the gas pressure (P_ℓ) and metering fluid flow rate (Q_{mo}), that would satisfy the stress and strain rate conditions experienced at the yield point, were considered before numerical values for n and α were selected. Expressions for P_ℓ and Q_{mo} are developed in Appendix F and can be written as:

$$P_\ell = \frac{2\sigma_o t}{D_o} n \quad [9]$$

and

$$Q_{mo} = \frac{\pi}{2} L_e D_o^2 \dot{e}_o (\alpha - 1) \quad [10]$$

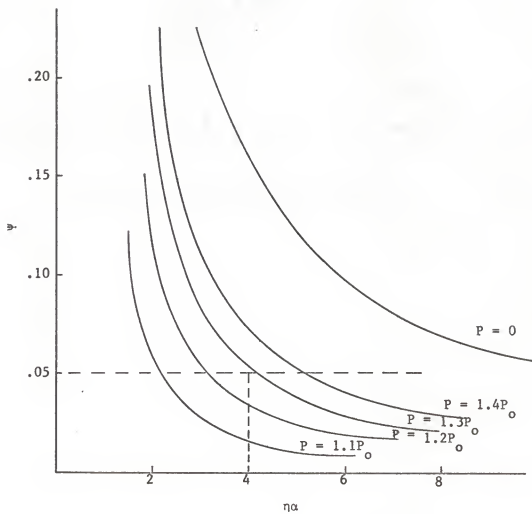


Figure 6. Comparison Between $\eta\alpha$ and Absolute Value of Ψ

where D_o = initial specimen I.D.
 σ_o = yield point stress of specimen
 t = wall thickness of specimen
 P_l = pressure of gas in reservoir L
 L_e = effective length of specimen
 Q_{mo} = metering fluid flow rate
 $\dot{\epsilon}_o$ = specified strain rate.

Equations [9] and [10] are superimposed upon Figure 7 for the low carbon steel material, $\sigma_o = 45,000$ psi. Values for D_o , t and L_e were taken from Appendix D. The maximum strain rate expected to be tested, $\dot{\epsilon}_o = 25$ in./in.-sec., was used in developing Figure 7.

A compromise between the high metering fluid flow rates (Q_{mo}) and high gas pressures (P_l) had to be made. α was selected to have the value of ten. η was selected to have the value of .45 which maintained the design point within region I of Figure 7. For these values $\eta\alpha = 4.5$ and from Figure 6 the design piston velocity variation in the plastic range became approximately 4%.

Although the plastic strain region was of primary importance when studying properties affecting the formability of materials, the elastic region was also of interest. From Figure 6 for $\eta\alpha = 4.5$ and the initial elastic range pressure of zero ($P = 0$), the piston velocity variation was approximately 13.8%. This meant the initial piston velocity was 13.8% greater than the yield point velocity; or that the initial specimen strain rate would be 13.8% greater than the yield point strain rate.

Design Configuration

The complete design configuration is shown in Figure 8. Briefly, the system is composed of the main cylinder, double ended piston, specimen,

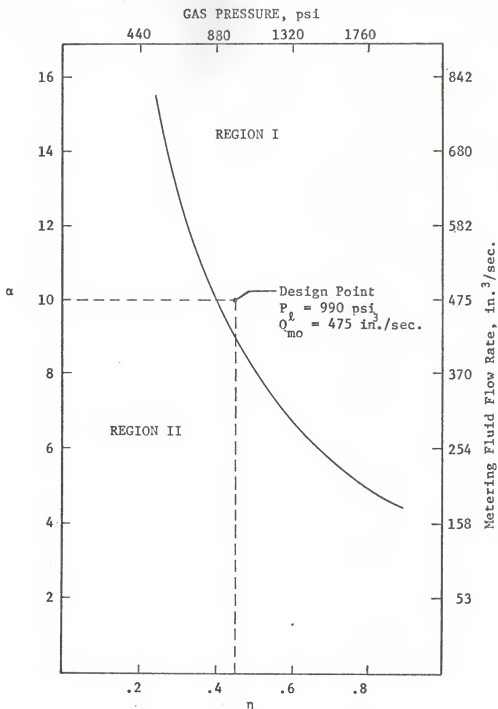


Figure 7. Comparison Between η and α and Between Gas Pressure and Metering Fluid Flow Rate

specimen retainer, valve and manifold assembly, metering orifice, release mechanism, fluid reservoir and gas reservoir. Not shown in the figure are the gas fill bottle used to fill the gas reservoir and fluid reservoir pressurization bottle used to pressurize the metering fluid. This causes the fluid to flow back into the main cylinder and move the piston back into its test position. Both of these bottles were commercial gas bottles. A pressure regulator was used with the bottle for pressurizing the metering fluid since pressures less than 40 psi were needed.

The ends of the double ended piston were selected to be 1-1/4 in. in diameter and 4 in. in diameter. These values satisfied equation [7] for $\alpha = 10$. The piston stroke, piston velocity and piston displacement required to rupture the specimen were considered in selecting the diameter of the small end of the piston. For a 1-1/4 in. diameter small end, the double ended piston velocity would approach 40 in./sec. and require a stroke of approximately .60 in. to rupture a specimen experiencing a strain of 30% at a strain rate of 25 in./in.-sec. The specimen fluid retaining plug (see Figure 8) was designed so as to provide for a means of accelerating the piston before loading the specimen. The plug is free to be positioned at any point inside the specimen. During the transient period of piston travel, the plug travels to the end of the specimen. When the plug reaches the end of the specimen, it strikes the specimen retainer thus terminating its travel and causing the fluid to pressurize. An analysis of the piston transient motion is made in Appendix G. A means for decelerating the piston was provided by shaping the piston to act as a valve during its stroke and close off the port through which the metering fluid flows in leaving the main cylinder. The overall piston stroke was selected to be approximately 2 in. This proved adequate for accelerating the piston, rupturing the specimen and decelerating the piston. Pressure

variation in the gas reservoir directly affects the uniformity of piston velocity. Therefore, the volume of the gas reservoir was made large enough to maintain less than a 1/4% reduction in gas pressure during the working portion of the piston stroke. This amounted to a volume of about 2300 in.³.

The diameter of the gas valve opening was made large enough to maintain less than a 1% drop in the pressure of the gas flowing through the valve. This was computed for the maximum strain rate of 25 in./in.-sec. and yielded a minimum valve diameter of 1-1/2 in.

Metering fluid orifice sizes were determined from Appendix H for strain rates in powers of ten from .001 in./in.-sec. to 100 in./in.-sec. These are shown in Table 1.

Stress Analysis

Since the system was to be operated with moderately high pressure gas, the static design stresses were maintained to a conservative value of 1/4 the yield point strength of the material for areas critical to safety. In order to verify the design strength, the system was tested to a proof pressure of twice the maximum operating pressure or 3000 psi. The maximum permissible operating pressure was specified to be 1500 psi. Fatigue was not considered a factor in the design since the expected life of the machine would include less than 10³ cycles. Stress calculations of several of the components are included in Appendix I.

Materials

Metal materials were limited to machinable stock that was locally available. C1018 steel in the cold rolled condition was used almost exclusively. Surface finishes were specified to be 125 RMS for general machine operations, 16-32 RMS for static seal surfaces and 8-16 RMS for dynamic

TABLE 1.

Metering Orifice Sizes to be Used
for Various Specimen Strain Rates

Strain Rate, in./in.-sec.	Orifice Area, in. ²	Orifice Diameter, in.
100.0	.63	.895
25.0	.157	.447
10.0	.063	.283
1.0	.0063	.089
0.1	.00063	.028
0.01	.000063	.009
0.001	.0000063	.003

Note: Orifice sizes were computed for an effective specimen length of
 $L_e = 2$ in.

seal surfaces. For a complete list of parts and materials, see Figure 8.

Seals

Positive sealing of the working fluids had been a problem with the preliminary testing done by Giles (5). Therefore, U-cup seals with a positive seal specification of 10,000 psi were used wherever possible in the new design. O-rings were used for the piston seals. The seal material was specified to be a urethane elastomer compound. The hardness was specified to be 92 Shore A to help maintain leakage to an insignificant value.

Fluids

Due to relatively high pressure gas being required to properly operate the test system, commercial nitrogen gas was specified to eliminate the hazard of explosion. The metering fluid and specimen pressurizing fluid were specified to be No. 5 wt. hydraulic fluid.

Functional Tests

To verify that the testing system met the design objectives, two series of functional tests were performed as follows:

1. With the specimen removed, a linear potentiometer was attached to the piston. Time-displacement curves of the piston travel were recorded to verify that the piston reached a constant velocity within an acceptable piston displacement and remained uniform for a sufficient length of the stroke.
2. With a specimen installed, time-displacement curves were recorded to determine that the uniformity of piston

velocity remained near the prescribed 5% variation during the yield portion of straining the specimen.

Figure 9 is a time-displacement plot of the data taken for part 1. above. The piston velocity becomes uniform within approximately .060 inches of travel. This was near the value of .045 inches computed (see Appendix G). The piston velocity remains uniform for approximately 1.32 in. of travel which was more than sufficient since computation shows that approximately .60 in. of piston travel is needed to rupture a specimen experiencing 30% strain. Figure 10 is a time-displacement plot of the data taken for part 2. The maximum variation in piston velocity was computed from the maximum change in slope occurring within the specimen yield portion of the piston stroke. A 5.6% velocity variation was computed from the figure which was very near the desired 5% variation and approximately 1.6% greater than the design value. The equivalent minimum strain rate computed from the figures was approximately 29.6 in./in.-sec. This demonstrates that the testing machine will function up to the desired maximum strain rate 25 in./in.-sec. to be tested and meet acceptable values for linearity of piston travel.

The difference between the initial piston velocity (before specimen is pressurized) and the piston velocity experienced during initial plastic straining of the specimen was computed from Figure 10 to be 9.2%. This is essentially the velocity variation occurring during elastic straining of the specimen.

Proposed Instrumentation

A means for measuring specimen fluid pressure and strain is necessary in order to provide the information to make a stress-strain analysis of the materials as a function of strain rate. Provisions were made in the design

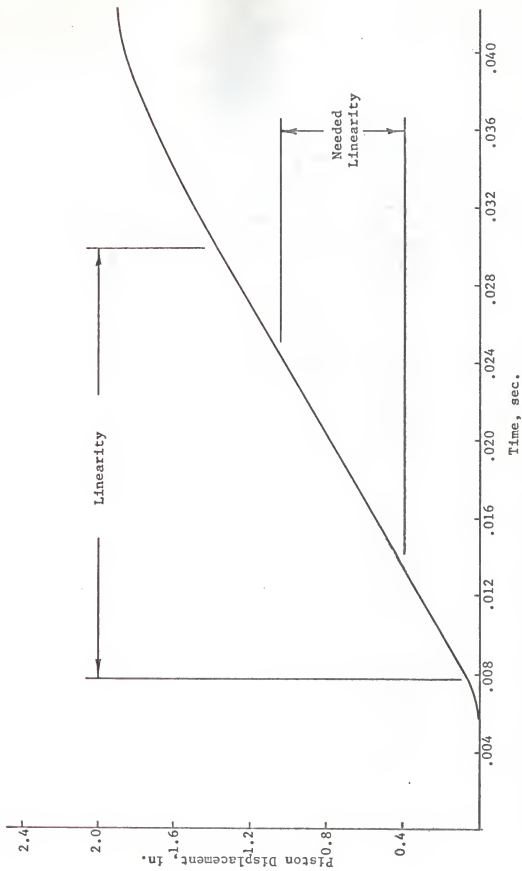


Figure 9. Piston Time-Displacement Diagram, Without Specimen

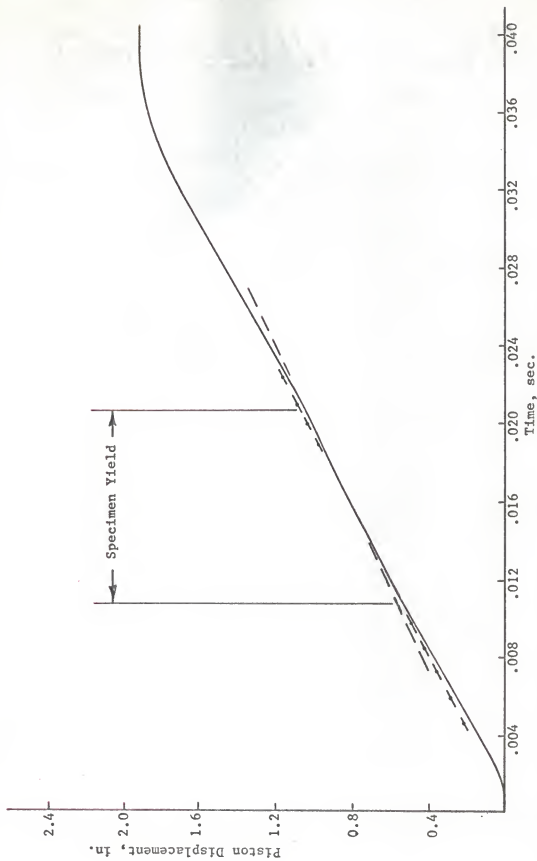


Figure 10. Piston Time-Displacement Diagram, With Specimen

of the specimen fluid retainer plug for incorporating a strain gage dynamometer. (See Figures 8 and 11.) Specimen circumferential strain will be sensed by three high resolution linear potentiometers equally spaced around the circumference of the specimen and will measure the diametrical expansion of the specimen which can be converted to circumferential strain. (See Figure 11.) The potentiometers can be mounted in a metal cylinder which also acts as a shield against the specimen fluid spray upon rupture of the specimen.

Simple computations show that for a strain rate of 25 in./in.-sec., the yield point will be reached in the specimen in approximately $.6 \times 10^{-3}$ sec. and if the specimen ruptures after 20% strain having occurred, the total elapsed time will be 8×10^{-3} sec. To provide for proper recording of these events, a Honeywell Model 1508 Visicorder will be used which has a frequency response of 3000 cps. The strip chart recording technique of this machine eliminates the need for having a precision triggering mechanism and allows for both pressure and strain to be recorded simultaneously upon the same chart. Figure 11 is the complete instrumentation schematic and list of parts.

Operating Instruction

A complete list of operating instructions are given in Appendix J which should be used for proper and safe operation of the testing machine.

Discussion

The fact that the ultimate strength in a tubular specimen does not necessarily coincide with the maximum fluid pressure, as substantiated by equation [2], has been overlooked by some experimenters in the past. The ultimate strength must be associated with the actual fluid pressure at the point of specimen rupture. The amount of error introduced into the data, by associating

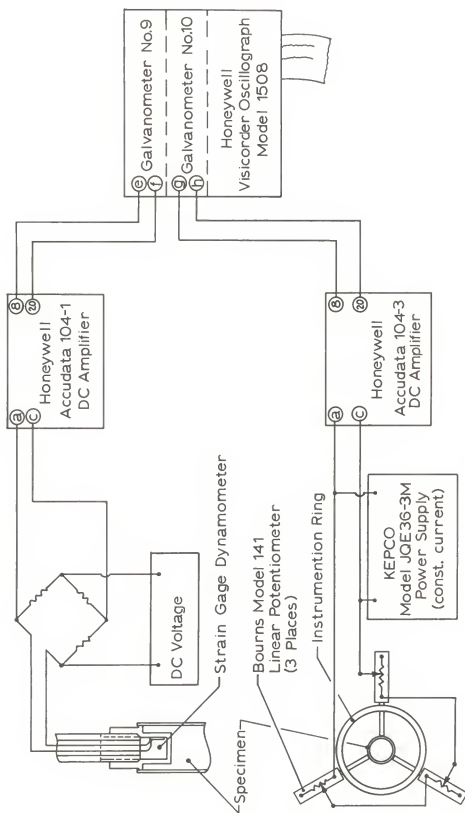


Figure 11. Electrical Wiring Diagram

the ultimate strength with the maximum fluid pressure, is dependent upon how close to the maximum pressure point the specimen happened to fail. The tendency for the tubular specimen to demonstrate a rather flat load curve, as shown in Figure 3, may partly explain why the yield point strength and ultimate strength tend to become inseparable at the higher strain rates, as mentioned by Clark and Duwez (3).

Strain rate being dependent upon the amount of specimen strain incurred, as indicated by equation [1], may prove to be a limitation with the tubular specimen configuration. However, since it is impractical not to mount the specimen with the ends restrained, the inherent decrease in strain rate expected with the tubular specimen may be an advantage. The restrained ends of the specimen will cause the circumferential strain to focus toward the center section of the specimen. Since this tends to reduce the effective length of the specimen (L_e), an increase in strain rate for the center section can be expected as viewed from equation [1].

To reduce the complexity of determining material properties, it is important that the specimen experience only unidirectional strain. However, with the ends of the tubular specimen restrained, the specimen will assume a shape similar to that shown in Figure 12. (The specimen is truncated at the center section with only the lower half shown.) From the loading superimposed upon the figure, it is apparent that an increasing longitudinal state of stress develops in the specimen at the center section with increased circumferential strain. The magnitude of the longitudinal stress (σ_L) is proportional to $P(A_c - A_e)$, where A_c is dependent upon the amount of specimen strain at the center section and P is the fluid pressure. To compensate for this effect, the specimen ends should be mounted so they will be exposed to the fluid pressure. This creates a longitudinal compression load that counteracts the

developing longitudinal tensile load. For a 7/8 I.D. x .020 wall specimen experiencing 20% strain, the net longitudinal stress will be less than 12,000 psi tension. This is well under the yield point strength of most materials and is not expected to have a significant effect upon material properties data.

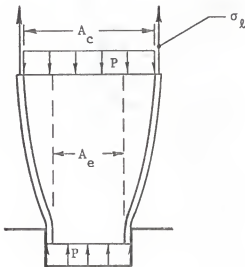


Figure 12. Specimen Load Diagram

The design of the testing system, presented in this report, centers around the specimen proportional limit. This presented a common working point from which to view either the elastic strain region or plastic strain region of the specimen. The static design yield strength was selected to be 45,000 psi. However, provisions were made in the design to accommodate the expected increase in material strength properties as a function of increased strain rate. For example, the specimen fluid pressure required to induce a 45,000 psi yield stress in the specimen was selected to be 2200 psi. Even if the dynamic strength properties double in value, the resulting fluid pressure for both the specimen yield point strength and ultimate strength will remain

well under the 10,000 psi design limit for the seals.

From equation [3] it is obvious that materials having different K/σ_0 and n values from the low carbon steel plotted in Figure 3 will produce different maximum fluid pressures than that shown in Figure 3. This will result in a strain rate variation different from the 4% predicted for the low carbon steel material. To account for these differences, the gas pressure (P_g) of the system can be adjusted. For example, if the maximum fluid pressure becomes $1.4 P_0$, rather than the $1.28 P_0$ discussed for the low carbon steel, then from Figure 6 a value of $\eta\alpha = 5.5$ must be chosen to maintain the piston velocity variation (strain rate variation) within 5% of the desired velocity (strain rate). Since the system parameter $\alpha = 10$ was physically set by the piston area ratio, then η is the only remaining variable and for $\eta\alpha = 5.5$, $\alpha = 10$, $\eta = .55$. From Figure 7, the gas pressure (P_g) must be increased to a value of 1210 psi for $\eta = .55$.

The formability of a material depends almost entirely upon the plastic properties of the material with particular attention given to the dynamic ultimate strength. Therefore, in designing the system presented in this thesis, an emphasis was made to control the specimen variables during plastic straining of the specimen. The instrumentation proposed will be used to determine the stress-strain curve in the plastic region. Strain (e) will be measured and stress (σ) will be calculated by use of thin wall theory $\sigma = PD/t$. (P is the specimen fluid pressure associated with an arbitrary specimen diameter D , and t is the original wall thickness.)

Wire resistance strain gages which would be suitable for this experimental work are usually limited to a maximum elongation of 15%; therefore, high resolution, linear potentiometers were proposed to sense specimen circumferential strain. This assures good data can be gathered for the specimen

ultimate strength. If it becomes desirable to obtain more accurate data in the elastic region, then wire resistance strain gages can easily be integrated into the instrumentation circuit in place of the linear potentiometer.

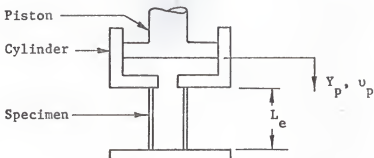
In conclusion, it is pertinent to mention that the testing system presented in this thesis will probably be suitable for testing materials at strain rates considerably higher than 25 in./in.-sec. Results of the functional tests presented in Figure 9 indicated that the piston responded fast enough to provide more than twice the amount of linear motion needed to strain specimen at 25 in./in.-sec. It is likely that strain rates greater than 100 in./in.-sec. can be tested without becoming limited by piston response. However, experience gained from performing the functional tests indicated that a minor alteration should be made to the release plunger to provide more damping of its motion if strain rates exceeding 25 in./in.-sec. by any significant amount are to be tested.

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APPENDIX A

Determination of Strain Rate



Assumptions:

1. The specimen and cylinder are filled with an incompressible fluid.
2. The piston travels at a uniform velocity, v_p .
3. The specimen expands radially and uniformly from end to end with no change in length.

Let engineering strain be defined

$$e = \frac{\Delta C}{C_0}$$

Then $C = C_0 + \Delta C = C_0 + C_0 e = C_0 (1 + e)$

where C = circumference of specimen
 C_0 = original circumference of specimen
 ΔC = incremental change in circumference.

$$C_0 = \pi D_0 \text{ and } C = \pi D \text{ or } D = \frac{C}{\pi}$$

so $D = \frac{C_0 (1+e)}{\pi} = D_0 (1+e) = 2r_0 (1+e) = 2r$.

Then the specimen volume is

$$V = \pi r^2 L_e = \pi [r_0 (1+e)]^2 L_e$$

The change in specimen volume is

$$\frac{dV}{dt} = 2\pi r_0^2 L_e (1+e) \frac{de}{dt}$$

This must be equal to the change in cylinder volume,

$$\frac{dV}{dt} = \pi r_s^2 v_p = 2\pi r_o^2 L_e (1+e) \frac{de}{dt}$$

where r_s = piston radius
 v_p = piston velocity $\frac{dy}{dt}$.

The expression for specimen strain rate becomes

$$\dot{e} = \frac{de}{dt} = \left(\frac{r_s}{r_o} \right)^2 \frac{v_p}{2L_e (1+e)}. \quad [a]$$

For the elastic region ($e \neq 0$) the specimen strain rate reduces to

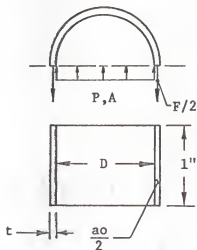
$$\dot{e} = \left(\frac{r_s}{r_o} \right)^2 \frac{v_p}{2L_e}. \quad [b]$$

APPENDIX B

Part 1. Determination of Fluid Pressure

Assumptions:

1. The theory of thin walled cylinders applies.
2. The material is homogenous and isotropic.
3. The pressure is uniform throughout volume.



From a summation of forces

$$\frac{F}{2} + \frac{F}{2} = PA = \sigma \left(\frac{a_0}{2} + \frac{a_0}{2} \right)$$

or
$$P = \left(\frac{a_0}{A} \right) \sigma \quad [a]$$

where P = fluid pressure in specimen
 a_0 = original crosssection of specimen wall
 A_0 = crosssectional area of fluid
 σ = engineering circumferential stress.

The fluid pressure at the specimen yield point is

$$P_0 = \left(\frac{a_0}{A_0} \right) \sigma_0 \quad [b]$$

where A_0 = original crosssectional area of fluid
 σ_0 = yield point strength of material.

From Appendix I, the specimen diameter was shown to be

$$D = D_0 (1+e) \quad [c]$$

Then for a unit length of the specimen

$$A = A_0(1+e) \quad [d]$$

where e = engineering strain.

Combining equations [a], [b] and [d],

$$P = \frac{P_0}{\sigma_0} \frac{\sigma}{1+e} \quad [e]$$

From Dieter (7), Chapter 9, expressions for true stress and strain in the plastic strain region are:

$$\Sigma = K\varepsilon^n \quad [f]$$

$$\varepsilon = \ln(e+1) \quad [g]$$

$$\Sigma = \sigma(e+1) \quad [h]$$

where Σ = true stress
 K = material strength coefficient
 ε = true strain
 n = strain hardening coefficient.

Combining equations [f], [g] and [h],

$$\sigma = \frac{K[\ln(e+1)]^n}{e+1} \quad [i]$$

Combining equations [e] and [i],

$$P = \frac{K[\ln(1+e)]^n P_0}{\sigma_0(1+e)^2} \quad [j]$$

Part 2. Determination of Maximum Fluid Pressure

To determine the maximum fluid pressure which occurs during plastic yielding, the principle of maxima-minima was applied to equation [j].

$$\frac{dP}{de} = \frac{P_o K n (1+e)^2 [\ln(1+e)]^{n-1}}{\sigma_o (1+e)^5} = \frac{2P_o K (1+e) [\ln(1+e)]^n}{\sigma_o (1+e)^4} = 0$$

or
$$n[\ln(1+e)]^{n-1} = 2[\ln(1+e)]^n.$$

This reduces to

$$e = e_1^{n/2} - 1 \quad [k]$$

where $e_1 = 2.718.$

Combining equations [k] and [j],

$$P_m = \frac{K}{\sigma_o} \left(\frac{n}{2e_1} \right)^n P_o = \frac{K}{\sigma_o} \left(\frac{n}{5.44} \right)^n P_o \quad [1]$$

The fact that equation [1] represents a maximum is substantiated by Figure 3.

APPENDIX C

Determination of Specimen Fluid Pressure at the Yield Point

From Appendix B, the specimen fluid pressure at the yield point was determined to be

$$P_o = \frac{a_o}{A_o} \sigma_o \quad [a]$$

where a_o = crosssectional area of specimen wall
 A_o = crosssectional area of fluid
 σ_o = yield strength of specimen.

For a unit length section of the specimen,

$$\frac{a_o}{2} = (1)t$$

$$A_o = (1)D_o.$$

Therefore,

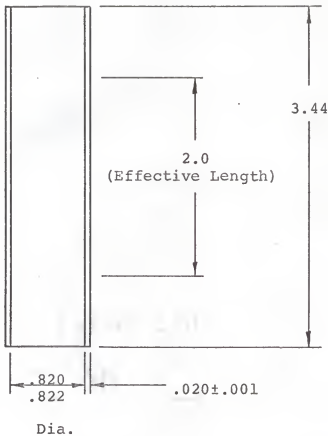
$$P_o = 2\sigma_o \left(\frac{t}{D_o} \right). \quad [b]$$

For material with a yield strength $\sigma_o = 45,000$ psi

$$P_o = 9(10^4) \left(\frac{t}{D_o} \right). \quad [c]$$

APPENDIX D

Specimen Configuration

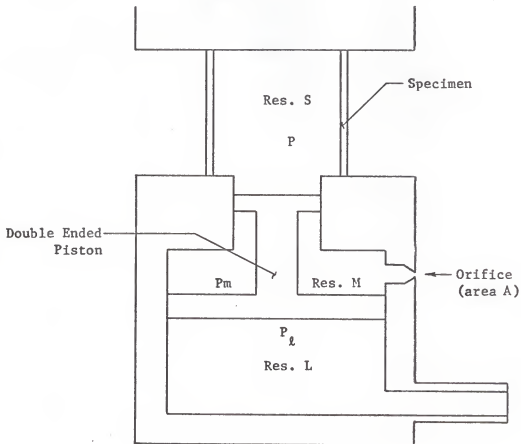


NOTES:

1. "Effective length" represents the actual length of specimen expected to be deformed.
2. Wall thickness variation from point to point on the specimen shall be less than .0005 in.
3. Surface finish shall be 8 to 16 RMS.
4. Dimensions are in inches.

APPENDIX E

Determination of Uniformity of Piston Velocity



Assumptions:

1. Reservoir L is filled with a constant pressure gas.
2. Reservoir M is filled with an incompressible fluid which is metered through the orifice.
3. Reservoir S is filled with an incompressible fluid which expands the specimen with piston travel.
4. The piston velocity is constant or the system is in a steady state condition.
5. Seal friction forces are negligible in comparison to the piston actuating force, $P_\ell A_\ell$.

The following parameters are defined:

$$\eta = \frac{P_l}{P_o} \quad [a]$$

$$\alpha = \frac{A_l}{A_s} \quad [b]$$

where P_l = gas pressure in reservoir L
 P_s = fluid pressure in reservoir S at yield point
 A_o = area of large end of piston
 A_s = area of small end of piston.

Summing static forces on the piston,

$$P_l A_l = P_m (A_l - A_s) + P A_s$$

$$\text{or } P_m = \frac{P_l A_l - P A_s}{A_l - A_s} \quad [c]$$

Combining equations [a], [b] and [c],

$$P_m = \frac{\eta \alpha P_o - P}{\alpha - 1} \quad [d]$$

From Blackburn (7), equation 3.47, incompressible flow through an orifice can be expressed

$$Q_m = C_v A \sqrt{\frac{2}{\rho} P_m}$$

where A = orifice area
 C_v = orifice discharge coefficient
 ρ = fluid density.

$$\text{But, } Q_m = A_m v_p$$

where A_m = metering fluid piston area
 v_p = double ended piston velocity,

$$\text{so } v_p = \frac{Q_m}{A_m} = C_v \left(\frac{A}{A_m} \right) \sqrt{\frac{2}{\rho} P_m} \quad [e]$$

Combining equations [d] and [e],

$$v_p = C_v \left(\frac{A}{A_m} \right) \sqrt{\frac{2(\eta\alpha P_o - P)}{\rho(\alpha - 1)}} \quad [f]$$

Then the piston velocity at the yield point pressure is

$$v_{po} = C_v \left(\frac{A}{A_m} \right) \sqrt{\frac{2(\eta\alpha P_o - P_o)}{\rho(\alpha - 1)}} \quad [g]$$

where subscript "o" signifies value at specimen yield point.

The fractional difference in piston velocity experienced at some arbitrary specimen pressure (P) as compared to the piston velocity experienced at the yield point pressure (P_o) is defined:

$$\psi = \frac{v_p - v_{po}}{v_{po}} = 1 - \frac{v_p}{v_{po}} \quad [h]$$

Combining equations [f], [g] and [h],

$$\psi = 1 - \sqrt{\frac{\eta\alpha - P/P_o}{\eta\alpha - 1}} \quad [i]$$

APPENDIX F

Part 1. Determination of Gas Pressure. (P_ℓ)

From equation [a], Appendix E, $P_\ell = \eta P_o$.

From equation [b], Appendix C, $P_o = 2\sigma_o \left(\frac{t}{D_o} \right)$.

Combining these equations,

$$P_\ell = \frac{2\sigma_o t \eta}{D_o}.$$

Part 2. Determination of Metering Fluid Flow Rate (Q_{mo}) at the Yield Point Fluid Pressure

From Appendix A, equation [b],

$$\dot{e} = \left(\frac{r_s}{r_o} \right)^2 \frac{v_p}{2L_e}.$$

From Appendix E, figure, $A_m = A_\ell - A_s$

where A_m = metering fluid piston area
 A_ℓ = area of large end of piston
 A_s = area of small end of piston.

If $A_\ell = \alpha A_s$, then $A_m = A_s (\alpha - 1) = \pi r_s^2 (\alpha - 1)$.

Then the metering fluid flow rate may be defined:

$$Q_{mo} = A_m v_{p_o}$$

$$Q_{mo} = \pi r_s^2 (\alpha - 1) \left[2L_e \left(\frac{r_o}{r_s} \right)^2 \dot{e}_o \right]$$

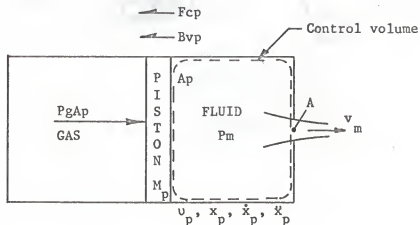
$$Q_{mo} = 2\pi L_e r_o^2 \dot{e}_o (\alpha - 1)$$

$$Q_{mo} = \frac{\pi}{2} L_e D_o^2 \dot{e}_o (\alpha - 1).$$

Subscript "o" signifies value at specimen yield point.

APPENDIX G

Determination of Piston Response



Assumptions:

1. The gas pressure (P_g) remains constant.
2. The fluid is incompressible. Density (ρ) is constant and uniform throughout the control volume.
3. The fluid pressure (P) is uniform throughout the control volume.
4. The piston and cylinder containing the fluid are rigid.

Using a control volume analysis and the principle of conservation of mass

$$\rho A v_m = \rho A v_{in} - \frac{\partial}{\partial t} (\rho V) = 0 - \frac{\partial \rho}{\partial t} V - \rho \frac{\partial V}{\partial t}, \quad [a]$$

where A = metering orifice area
 v = fluid velocity through orifice
 V^m = instantaneous fluid volume.

But if ρ is constant, then $\frac{\partial \rho}{\partial t} = 0$ and [a] becomes $\rho A v_m = -\rho \frac{\partial V}{\partial t}$

$$\text{or} \quad Q_m = A v_m = -\frac{\partial V}{\partial t}. \quad [b]$$

$$\text{However, } V = V_0 - A_p x_p \quad [c]$$

where V_0 = initial fluid volume
 A_p = piston area
 x_p = piston displacement.

Combining equations [b] and [c],

$$Q_m = \frac{\partial V}{\partial t} = A_p \frac{\partial x_p}{\partial t} = A_p \dot{x}_p \quad [d]$$

From Appendix E,

$$Q_m = C_v A \sqrt{\frac{2}{\rho} P_m} \quad [e]$$

Combining equations [d] and [e],

$$C_v A \sqrt{\frac{2}{\rho} P_m} = A_p \dot{x}_p \quad [f]$$

From a summation of forces on the piston,

$$P_g A_p = P_m A_p + M_p \ddot{x}_p + B_{vp} \dot{x}_p + F_{cp} \frac{|\dot{x}_p|}{\dot{x}_p} \quad [g]$$

Assuming the damping force ($B_{vp} \dot{x}_p$) and friction force $F_{cp} \frac{|\dot{x}_p|}{\dot{x}_p}$ are negligible in comparison to the gas force ($P_g A_p$) and inertia force $M_p \ddot{x}_p$, then

$$P_g A_p = P_m A_p + M_p \ddot{x}_p \quad [g]$$

Combining equations [f] and [g],

$$P_g A_p = \frac{\rho}{2} \frac{A_p^3}{(C_v A)^2} \dot{x}_p^2 + M_p \ddot{x}_p$$

or in integral form,

$$\int \frac{d\dot{x}_p}{\frac{2(C_v A)^2 P_g}{\rho A_p^2} - \dot{x}_p^2} = \frac{\rho A_p^3}{2(C_v A)^2 M_p} \int dt \quad [h]$$

For the initial condition $t = 0$, $\dot{x}_p = 0$, equation [h] reduces to the form

$$\dot{x}_p = \sqrt{\frac{2(C_v A)^2 P_g}{\rho A_p^2}} \tanh \sqrt{\frac{\rho A_p^4 P_g}{2(C_v A)^2 M_p^2}} t \quad [i]$$

In integral form, [i] becomes

$$\int dx_p = \sqrt{\frac{2(C_v A)^2 P_g}{\rho A_p^2}} \int \tanh \sqrt{\frac{\rho A_p^4 P_g}{2(C_v A)^2 M_p^2}} t dt,$$

and for the initial condition $t = 0$, $x_p = 0$,

$$x_p = \frac{2(C_v A)^2 M_p}{\rho A_p^3} \ln \cosh \sqrt{\frac{\rho A_p^4 P_g}{2(C_v A)^2 M_p^2}} t . \quad [j]$$

For the proposed test system, operating at a strain rate of 25 in./in.-sec., parameter values were assumed:

Orifice discharge coefficient, $C_v = .65$
 Orifice area, $A = .63 \text{ in.}^2$
 Equivalent piston and fluid weight, $W = 10 \text{ lbs.}$
 Fluid density, $\rho = 7.95 \times 10^{-5} \text{ slugs/in.}^3$
 Piston area, $A_p = 12 \text{ in.}^2$
 Pressure of the gas, $P_g = 1000 \text{ lbs./in.}^2$.

Equations [i] and [j] reduce respectively to:

$$\dot{x}_p = 173 \tanh 403t$$

$$x_p = .064 \ln \cosh 403t.$$

The time required for the piston to reach 98% terminal velocity is then,

$$(.98)(173) = 173 \tanh 403t$$

or $403t = \tanh^{-1} .98 = 2.30.$

$$t = \frac{2.30}{403} = .0057 \text{ sec.},$$

and the distance the piston travels in reaching 98% terminal velocity is

$$x_p = .064 \ln \cosh 403 (.0057) = .045 \text{ in.}$$

APPENDIX H

Determination of Metering Fluid Orifice Area

Metering fluid flow rate at the specimen yield point can be expressed as

$$Q_{mo} = A_m v_{po}$$

where A_m = metering fluid piston area
 v_{po} = piston velocity at yield point.

From Appendix E, equation [g], piston velocity is

$$v_{po} = C_v \left(\frac{A}{A_m} \right) \sqrt{\frac{2(\eta \alpha P_o - P_o)}{\rho(\alpha - 1)}}$$

Then $Q_{mo} = C_v A \sqrt{\frac{2P_o(\eta \alpha - 1)}{\rho(\alpha - 1)}}$,

and $A = \frac{Q_{mo}}{C_v \sqrt{\frac{2P_o(\eta \alpha - 1)}{\rho(\alpha - 1)}}}$

From Appendix F, $Q_{mo} = \frac{\pi}{2} L_e D_o^2 (\alpha - 1) \dot{\epsilon}_o$.

Then $A = \frac{\pi L_e D_o^2 (\alpha - 1)^{3/2} \dot{\epsilon}_o}{2 C_v \sqrt{\frac{2P_o}{\rho} (\eta \alpha - 1)}}$

where A = metering orifice area
 L_e = effective length of specimen
 D_o = initial specimen I.D.
 C_v = orifice discharge coefficient
 P_o = specimen yield point fluid pressure
 ρ = fluid density
 $\dot{\epsilon}_o$ = specimen yield point strain rate.

APPENDIX I

Stress Analysis

NOTE: Only one set of parts was to be fabricated for the testing system. Cost of fabrication was of more importance than cost for material; therefore, only the machine operations necessary for proper assembly were made leaving excess material in some cases. Under these circumstances, most of the parts were more than adequately sized for strength. Therefore, the following stress analysis concerns only those sections of the parts which were critical to safety or proper operation of the machine.

Nomenclature:

A	area
A ^e	equivalent area
A ^e	piston area (small end)
D ^s	diameter
D	equivalent diameter of part
E ^e	modulus of elasticity (30 x 10 ⁶ , lb./in. ²)
F	external load
F ^e	initial required tightening load
ΔF _b	increase in bolt load
F _b	maximum bolt load
K ^m	equivalent spring constant for part
K _b ^p	bolt spring constant
L _b	length
L	effective length
N ^e	factor of safety (2.0 for parts critical to safety, 1.5 otherwise*)
n	number of bolts
P	internal proof pressure (3000 psi)
Q ^p	part separation factor (1.25)
r _i	internal radius
S _i ^t	yield point stress (48,000 psi)
S _s ^y	shear strength yield point (.6 S _y)
t _s	wall thickness

*These factors are applied to the proof pressure, 3000 psi. Equivalent factors for max. operating pressure are 4 and 3, respectively.

Part 1. Analysis of Manifold Bolts
(Reference pages 155-167 of Faires (8).)

Assume that ten 1/2-13NCx7-1/2 bolts will be used to fasten manifold to main cylinder.

$$K_b = \frac{AE}{L_e} = \frac{(.142 \text{ in.}^2)(30 \times 10^6 \text{ psi})}{6 \text{ in.}} = .71 \times 10^6 \text{ lb./in.}$$

Equivalent spring constant for parts.

$$D_e = (\text{Head Width}) + \frac{L_e}{2} = 3/4 \text{ in.} + \frac{6}{2} \text{ in.} = 3-3/4 \text{ in.}$$

$$A_e = \frac{\pi}{4} [D_e^2 - (1/2)^2] = \frac{\pi}{4} [(3-3/4)^2 - (1/2)^2] = 10.4 \text{ in.}^2$$

$$K_p = \frac{A_e E}{L_e} = \frac{(10.4 \text{ in.}^2)(30 \times 10^6 \text{ psi})}{6 \text{ in.}} = 52 \times 10^6 \text{ lb./in.}$$

Tightening load.

$$F_e = \frac{P A}{N} = \frac{(3000 \text{ psi})(16.1 \text{ in.}^2)}{10} = 4820 \text{ lb./Bolt.}$$

$$F_i = Q F_e \frac{K_p}{K_b + K_p} = 1.25(4820 \text{ lb.}) \frac{52 \times 10^6 \text{ lb./in.}}{.71 \times 10^6 \text{ lb./in.} + 52 \times 10^6 \text{ lb./in.}}$$

$$F_i = 5940 \text{ lbs.}$$

Increase in bolt load with pressure.

$$\Delta F_b = F_e \frac{K_b}{K_b + K_p} = 4820 \text{ lbs.} \frac{.71 \times 10^6 \text{ lb./in.}}{.71 \times 10^6 \text{ lb./in.} + 52.0 \times 10^6 \text{ lb./in.}}$$

$$\Delta F_b = 65 \text{ lbs.}$$

Maximum bolt load.

$$F_{\max} = F_i + \Delta F_b = 5940 \text{ lbs.} + 65 \text{ lbs.} = 6005 \text{ lbs.}$$

Bolt stress.

$$S_p = \frac{NF_{\max}}{A} = \frac{2(6005 \text{ lbs.})}{.142 \text{ in.}^2} = 84,400 \text{ lb./in.}^2.$$

Use ten SAE Grade 5 bolts ($S_p = 85,000 \text{ lb./in.}^2$).

Part 2. Analysis of Main Cylinder
(Reference pages 255-256 of Faires (8).)

Wall thickness of cylinder.

$$t = r_i \left[\left(\frac{1}{1 - N\sqrt{3} P_i / S_y} \right)^{1/2} - 1 \right]$$

$$t = 2in \left[\left(\frac{1}{1 - 2\sqrt{3} \frac{(3000 \text{ psi})}{48,000 \text{ psi}}} \right)^{1/2} - 1 \right]$$

$$t = .26 \text{ in.}$$

Since 1/2 in. diameter bolts are to be used, the wall thickness will be made 1.0 in. thick for fabrication purposes.

Part 3. Analysis of Valve Stem Housing Wall Thickness

Wall thickness of housing.

$$t = r_i \left[\left(\frac{1}{1 - N\sqrt{3} P_i / S_y} \right)^{1/2} - 1 \right]$$

$$t = .62 \text{ in.} \left[\left(\frac{1}{1 - 2\sqrt{3} \frac{(3000 \text{ psi})}{48,000 \text{ psi}}} \right)^{1/2} - 1 \right]$$

$$t = .082 \text{ in.}$$

To allow for threading housing, the wall thickness will be made .25 in. thick.

Part 4. Analysis of Release Plunger for Column Strength
(Reference pages 211-213 of Faires (8).)

Slenderness ratio. (Assume diameter $D = .375 \text{ in.}$)

$$\frac{L_e}{D/4} = \frac{5.5 \text{ in.}}{.375 \text{ in.}/4} = 58.7.$$

Critical load is defined,

$$NF_e = S_y A \left[1 - \frac{S_y \left(\frac{4L_e}{D} \right)^2}{4\pi^2 E} \right]$$

Solving for column diameter,

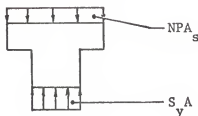
$$D = \frac{4}{\pi} \left(\frac{NF_e}{S_y} + \frac{S_y L_e^2}{\pi E} \right)^{1/2}$$

$$D = \left[1.27 \left(\frac{1.5 \times 3000 \text{ psi}}{48,000 \text{ psi}} + \frac{48,000 \text{ psi} (5.5 \text{ in.})^2}{\pi (30 \times 10^6 \text{ lb./in.}^2)} \right) \right]^{1/2}$$

$$D = .372 \text{ in.}$$

Make standard diameter $D = .375 \text{ in.}$

Part 5. Analysis of Piston Neck Area



Assume that specimen fluid pressure never exceeds $P = 10,000 \text{ psi.}$

(Neglect metering fluid pressure.)

From a summation of forces,

$$S_y A = NPA_s$$

$$A = A_s \left(\frac{NP}{S_y} \right)$$

$$\frac{\pi D^2}{4} = \frac{\pi D_s^2}{4} \left(\frac{NP}{S_y} \right)$$

$$D = D_s \left(\frac{NP}{S_y} \right)^{1/2}$$

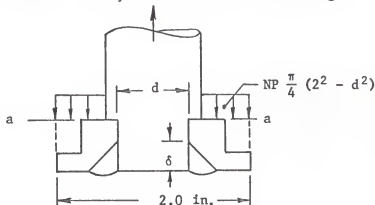
Diameter of small end of piston has been defined, $D_s = 1.25$ in.

Neck diameter is

$$D = 1.25 \left[\frac{1.5 (10,000 \text{ psi})}{48,000 \text{ psi}} \right]^{1/2} = .694 \text{ in.}$$

Make standard diameter, $D = .750$ in.

Part 6. Analysis of Main Gas Valve Plug.



From summation of forces at section a-a,

$$NP \frac{\pi}{4} (2^2 - d^2) - S_y \left(\frac{\pi}{4} \right) d^2 = 0$$

$$d = \left(\frac{4 \text{ in.}^2}{1 + S_y/NP} \right)^{1/2} = \left(\frac{4 \text{ in.}^2}{1 + \frac{48,000 \text{ psi}}{2(3000 \text{ psi})}} \right)^{1/2}$$

$$d = .667 \text{ in.}$$

Make standard diameter, $d = .750$ in.

From summation of forces at weld joint,

$$NP \frac{\pi}{4} [(2)^2 - d^2] = S_s \pi d \delta$$

$$\delta = \frac{NP}{4S_s d} (4 \text{ in.}^2 - d^2) = \frac{2(3000 \text{ psi}) [4 \text{ in.}^2 - (.75 \text{ in.})^2]}{4(28,700 \text{ psi})(.75 \text{ in.})}$$

$$\delta = .237 \text{ in.}$$

Make

$$\delta = .250 \text{ in.}$$

APPENDIX J

Operating Instructions for Strain Rate Testing System

CAUTION

This testing system utilizes fluid subjected to relatively high pressures. Therefore, care should be taken to assure that operation of the system is in strict accordance with the following procedure. Figures 8 and 11 should be used in conjunction with this procedure.

I. To Fill Main Gas Reservoir:

1. Close the main gas valve and gas fill valve.
2. Connect 2200 psi nitrogen bottle to gas fill valve.
3. Open nitrogen bottle valve.
4. Open gas fill valve and fill main gas reservoir to desired operating pressure.
5. Close gas fill valve.
6. Close nitrogen bottle valve when test system is not to be used for extended periods.

II. To Install (or change) Orifice:

1. Push release plunger into place to keep fluid from flowing out of fluid reservoir.
2. Loosen and remove (one turn at a time) the four nuts holding the fluid reservoir in place.
3. Pull the fluid reservoir loose from the main cylinder.
4. Pour the fluid out of the fluid reservoir.
5. Remove orifice.
6. Position release plunger back into reservoir to allow fluid to flow through reservoir connection.
7. Insert desired orifice.
8. Install O-ring on orifice
9. Install fluid reservoir (tighten nuts one turn at a time).

III. To Fill System with Metering Fluid:

1. Fill fluid reservoir to within 7 inches of top with No. 5 wt. hydraulic fluid.
2. Replace fluid reservoir lid and tighten lightly with wrench.
3. Loosen fluid bleed nut on top of piston (one turn).
4. Pressurize fluid reservoir to 10 psi and hold pressure until all air has bubbled through bleed nut.
5. Re-tighten fluid bleed nut lightly with wrench.

IV. To Reset Piston for Test:

1. Remove gas bleed nut on end of main gas valve stem.
2. Pressurize fluid reservoir to 40 psi and hold until piston moves to bottom of stroke.

V. To Set Release Mechanism:

1. Loosen fluid bleed nut on top of piston (one turn).
2. Push release plunger into place (CAUTION - the release plunger must be pushed in far enough that it will not interfere with positioning the release pin and proper seating of the release ring. See Detail A of Figure 8.).
3. Tighten fluid bleed nut.
4. Place release ring on release pin.
5. Insert release pin into release nut.
6. Install release nut on release nipple (hand tight).

VI. To Install Specimen:

1. Place end plug seal inside specimen, one inch from top of specimen.
2. Set specimen in place.
3. Fill specimen with fluid to top of seal.
4. Insert specimen end plug into end of specimen. (Push into place so that seal will move into place on the end plug.)
5. Move potentiometer wipers on instrumentation ring back to provide room for specimen.
6. Place instrumentation ring and seal into specimen retainer.
7. Install specimen retainer.
8. Set potentiometer wipers against specimen.

VII. To Perform Test:

1. Replace gas bleed nut on end of main gas valve stem and tighten.
2. Open main gas valve.
3. Adjust main gas reservoir pressure to proper test pressure.
4. Ready instrumentation for test.
5. Initiate test with release mechanism. (Use small wrench to turn release nut.)
6. Put instrumentation in stand-by condition.
7. Shut off main gas valve.
8. Loosen gas bleed nut on end of main gas valve stem.
9. Remove specimen.

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THE ANALYSIS AND DESIGN OF A SYSTEM FOR
TESTING MATERIALS AT INTERMEDIATE
STRAIN RATES

by

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Considerable interest has developed in recent years for studying the effect of rate of strain on material properties. Several investigators have performed research in the area of low strain rates or very high rates such as developed with explosive forming methods. However, for intermediate strain rates (up to 25 in./in.-sec.) there seems to be little data available which are conclusive in a quantitative sense. Since many of the metal forming operations used by modern industry function in the range of intermediate strain rates, it is important that sufficient data be made available to thoroughly analyze material properties for these rates.

A program was initiated at Kansas State University to develop a technique for testing materials at intermediate strain rates. A review of previous investigations was made and an experimental test fixture was proposed to be used for initial material testing. The fixture was constructed and initial testing conducted to establish suitable testing methods and to determine the difficulties that had to be overcome before valid experimental data could be gathered.

This thesis presents the subsequent work performed in support of the test program. An analysis of the unique specimen configuration proposed to be used in this experimental program was made. Expressions for the specimen strain rate and specimen loading were developed in terms of the testing system parameters. A complete strain rate testing system that will be suitable for collecting valid material properties data was proposed. The analysis and design of the testing system was presented with emphasis on satisfying objectives of the magnitude and uniformity of strain rate. Results of functional tests performed on the system to verify that the system conformed to the design objectives were presented.