THE DESIGN AND CONSTRUCTION OF A 400,000-LB-CAPACITY TENSILE-TESTING DEVICE

By

Darrel D. Lemon

A thesis submitted to the Faculty and the Board of Trustees of the Colorado School of Mines in partial fulfillment of the requirements for the degree of Master of Science in Metallurgical Engineering.

Signed: Darrel D. Lemon
Darrel D. Lemon

Golden, Colorado

Date: April 27, 1970

Approved: Frank R. Campbell
F. R. Campbell
(Thesis Advisor)

Paul G. Herold
P. G. Herold
(Head, Department of
Metallurgical Engineering)

Golden, Colorado

Date: April 27, 1970

ABSTRACT

Tensile-testing machines of very high capacity are required to investigate the effects of size on the brittle fracture of steels. Capacities to 30,000,000 pounds are currently desired. No existing device has this capacity. A conventional testing machine of this capacity would be extremely expensive (approximately $3,000,000).

The purpose of this investigation was to attempt to prove that a tensile-testing device could be built at a reasonable cost which would satisfactorily test large specimens. In order to test this hypothesis it was necessary to design, construct, and test a large-scale device. A device of 400,000-lb capacity was decided upon. Satisfactory proof of the device's ability required that the concentricity of loading be at least as good as that produced by conventional tensile-testing machines. Another requirement was that there be no practical size limitation to the device. For instance, a device based upon the prototype of 40,000,000-lb capacity must be possible and must also give satisfactory performance. Low bulk and

versatility are desirable but not necessary features of such a device.

All of the requirements for the device were not only attained but exceeded by the 400,000-lb-capacity prototype. The concentricity exceeded that of standard ball seat fixtures by as much as one order of magnitude. The cost of having the prototype built would be \$0.50 for each 100 lb of capacity compared to approximately \$30.00 for each 100 lb of capacity for conventional tensile-testing machines. This is a cost ratio of approximately 60:1.

There is no reason to believe that there is any size limitation to the device. The hardenability of the hardened steel pins in the device does depend upon size. But the pins are designed primarily on the surface stress, and as long as the surface hardness can be obtained within approximately 10 R_c points of the theoretical, no changes in the design would be necessary.

TABLE OF CONTENTS

Figures and Tables

ACKNOWLEDGMENTS

The author wishes to thank Mr. F. R. Campbell, Dr. J. D. Lubahn, and Dr. W. D. Copeland for their guidance and encouragement during the course of this investigation.

The author is also indebted to Mr. G. M. Neunzert for his assistance and Dr. N. C. Schieltz for his photographic work.

The author wishes to acknowledge the financial support provided by the Aluminum Company of America.

INTRODUCTION

Many large steel structures being designed today have large cross-sectional areas. It is known that large structures are more susceptible to catastrophic failure by brittle fracture because of size effects. However, a quantitative relationship between large sizes and brittle fracture is not known. One reason for this lack of knowledge is the lack of mechanical testing capabilities to determine the strengths of various large-cross-section specimens.

The purpose of the present work is to design, build, and test a 400,000-lb capacity tensile-testing device which will give results comparable to results of a commercial tensile-testing machine. If successful, the basic design of this "prototype" will be used to build much larger devices. These larger devices will be used to test specimens having comparable cross-sections to those found in actual structures.

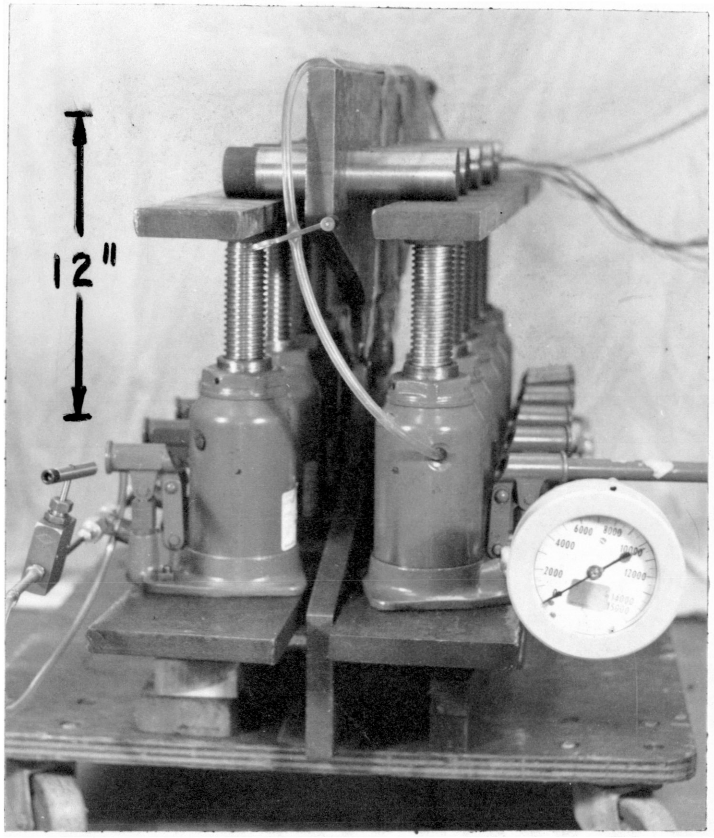
The design of the tensile-testing device is based primarily on equilibrium of force and moment. A plate specimen is surrounded by hydraulic jacks of identical capacity (equivalent force on each ram at the same hydraulic pressure). The jacks are equally spaced in all principal directions. The high-pressure reservoirs of all jacks are connected to insure equal force on each ram at a given hydraulic pressure. The jacks transmit load to the plate via pins inserted through accurately located holes in the plate. The finished tensile-testing device is shown in Figure 1. A combination of electrical strain gages and strain recorder is used to determine whether or not the loading is concentric.

Although testing to failure should be possible with the device, such a test is not deemed necessary to proving the device. If the loading can be shown to be sufficiently concentric up to the capacity of the device, it would certainly be concentric at a load between 0 and 400,000 lb. The present study terminated when this was satisfactorily proven.

After one specimen is tested, the specimen is cut from the plate, and another specimen is welded to the plate in the location of the previous specimen. A tensile test is then made on the new specimen. This procedure is continued for the number of specimens desired to be tested.

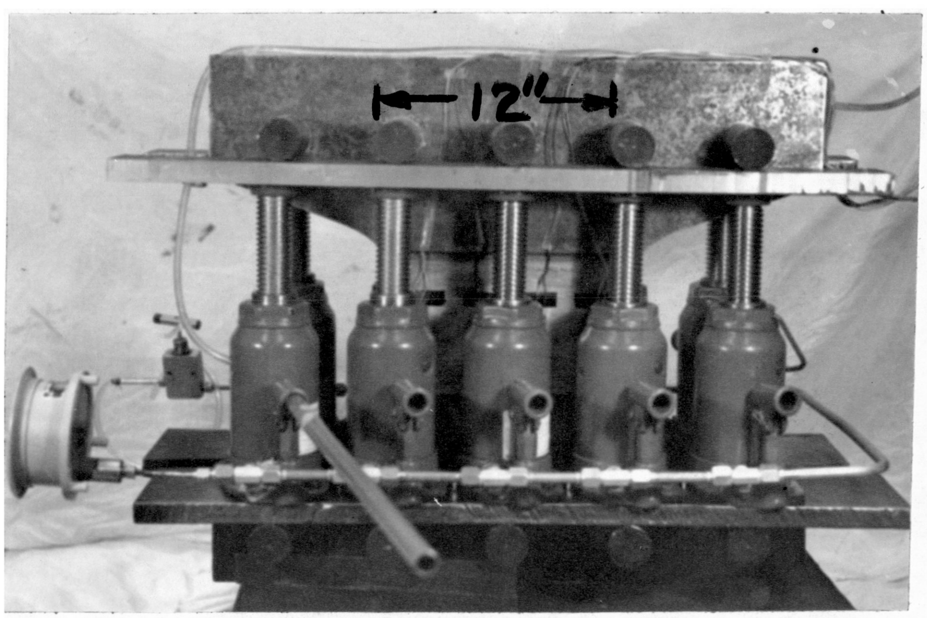
Several alternatives to the above method are possible, depending on the versatility required. A

Left Side



Right Side

Side View



Front View

Figure 1. Tensile-Testing Device

different plate of the material to be tested could be used for each test. The same plate could be slotted with the specimen held in place by pins during the test. Other designs could be feasible.

This investigator could find very little technical literature on this specific design problem.

DESIGN AND FABRICATION OF EQUIPMENT

Much of the important materials used came from local sources. This use of available materials restricted the design of the device to a certain extent.

The lack of machining experience on the part of this investigator was more of a handicap than was the restriction on materials. With the exceptions of flame-cutting the plate and drilling it, he performed all necessary machining and fabrication.

Professor F. R. Campbell assisted and advised on the installation of electrical resistance strain gages.

The design of the tensile-testing device consisted basically of three parts: design of the specimen, design of the hydraulic system, and design of the remainder of the device. The latter will be discussed first, requiring the major design effort. Much of the design was based on the textbook, Mechanics of Materials, by Higdon, Ohlsen, and Stiles[1].

Design of Testing Device - Less Hydraulics

It was decided to use a 1-inch-thick plate of AISI 1022 which was locally available and whose material properties are given in Table 1. This plate was originally 36 inches square but was flame cut to the shape shown in Figure 2. The vertical 6-inch center portion of the plate is the actual specimen, the remainder of the plate being part of the testing device. The combination of this plate containing the specimen as the center portion will hereafter be referred to as the plate-specimen. The specimen thickness was cut down to approximately 0.475 inch from the original 1-inch thickness.

It is desirable to place the hydraulic jacks as close as possible to the specimen to decrease the possibility of bending the specimen. The size of the base of the jacks is the physical limitation on this placement. The final placement is felt to have accomplished the objective within the physical limitations. The jacks used had a rated capacity of 40,000 lb each.

An important factor in the design was size in combination with weight. The size was limited so that

TABLE 1. Material Properties

1. Plate

Heat No.: U.S. Steel Corp. 49984 10-3

Nominal Composition: Reported by U.S. Steel Corp.
0.22% C max.
0.30/0.60% Mn
0.04% S max.
0.04% P max.

Heat Treatment: Hot rolled, annealed.

Tensile Properties: Plate specimen 1.00 x 0.5 in. thick —— room temperature test.

(Average of three tests)

0.2% Y. S.	33,600	psi
U T S	61,700	psi

Bearing Strength: 90,000 psi (Ref. 1, p.444)

2. Pins (All information from Clark and Varney[8])

Nominal Composition:
0.40% C
0.75% Mn
0.30% Si
1.83% Ni
0.80% Cr
0.25% Mo

Heat Treatment:

1650°F	Normalized and air cooled
1575°F	Austenitized and violently oil quenched
400°F	Temper

Tensile Properties:
0.2% Y. S. - 215,000 psi
U. T. S. - 270,000 psi

Bearing Strength: 320,000 psi (Ref. 1, p.444)

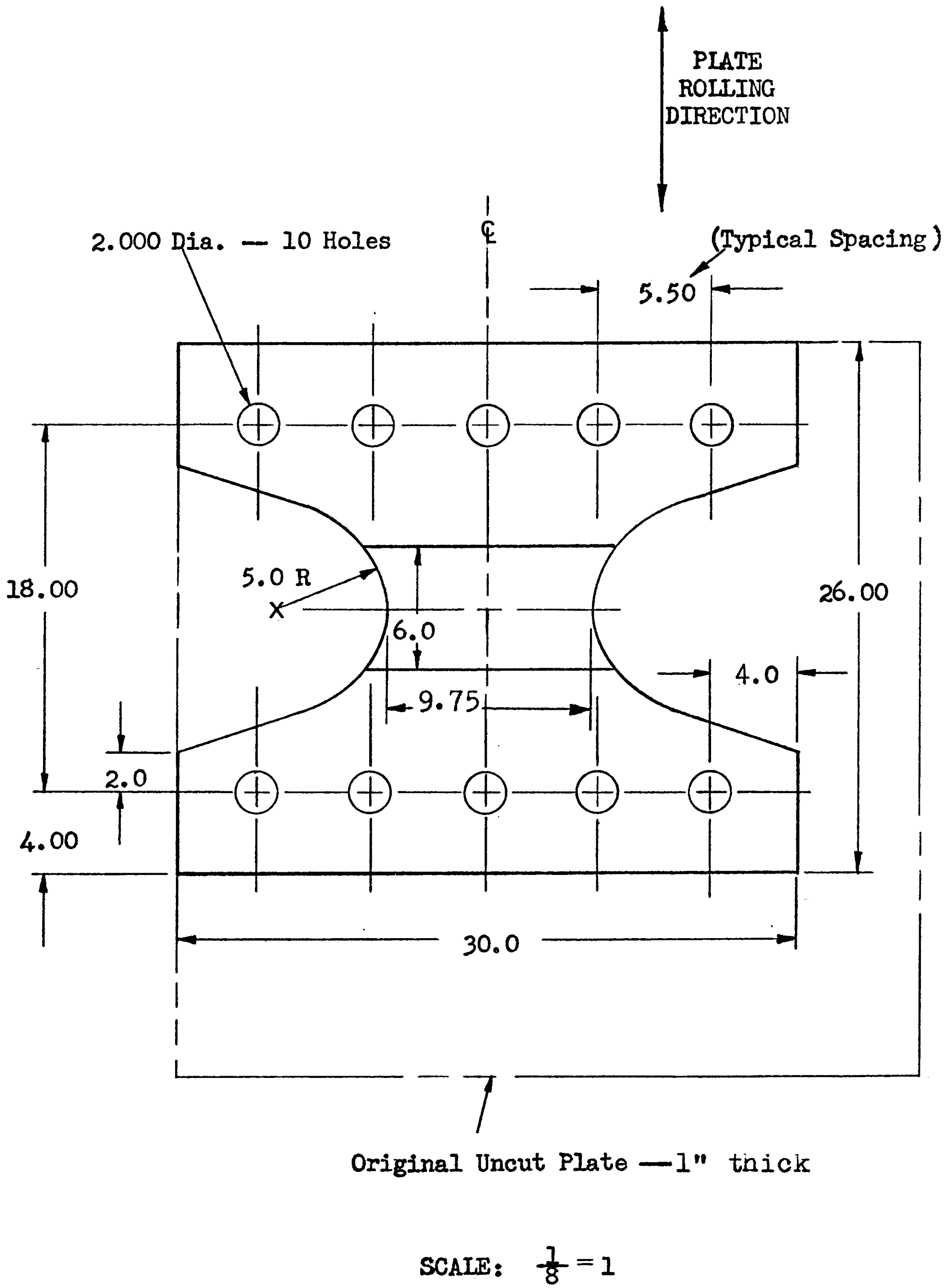


Figure 2. Specimen as Cut and Drilled from Original Plate.

the full-scale model would not be unmanageable due to size. High-strength AISI 4340 pins were chosen to transmit the load from the hydraulic jacks to the plate and then to the specimen. These pins reduced the weight of the device greatly and facilitated handling of the plate, since the pins are removable.

Design of Pins

All pins had the same cross-sectional area. The pins extended approximately 2.5 inches from the upper holes and approximately 7 inches from the lower holes in the plate. These maximum distances were used to calculate the moment in the flexure and deflection formulae. These lengths were the minimum lengths required to carry the load of the jacks.

The calculations for the pin design are shown in Appendix I. Bending stress was found to be the critical factor in the design of the pins. 1.9 inch diameter pins were required to transmit the load to the specimen. An allowable yield stress of 150,000 psi was used to determine the size of pin required. This stress level is easily attainable with AISI 4340 steel (see Table 1 for properties) in a 2-in.-diameter round.

The remainder of the plate-specimen was designed using the 1.9-inch-diameter pin.

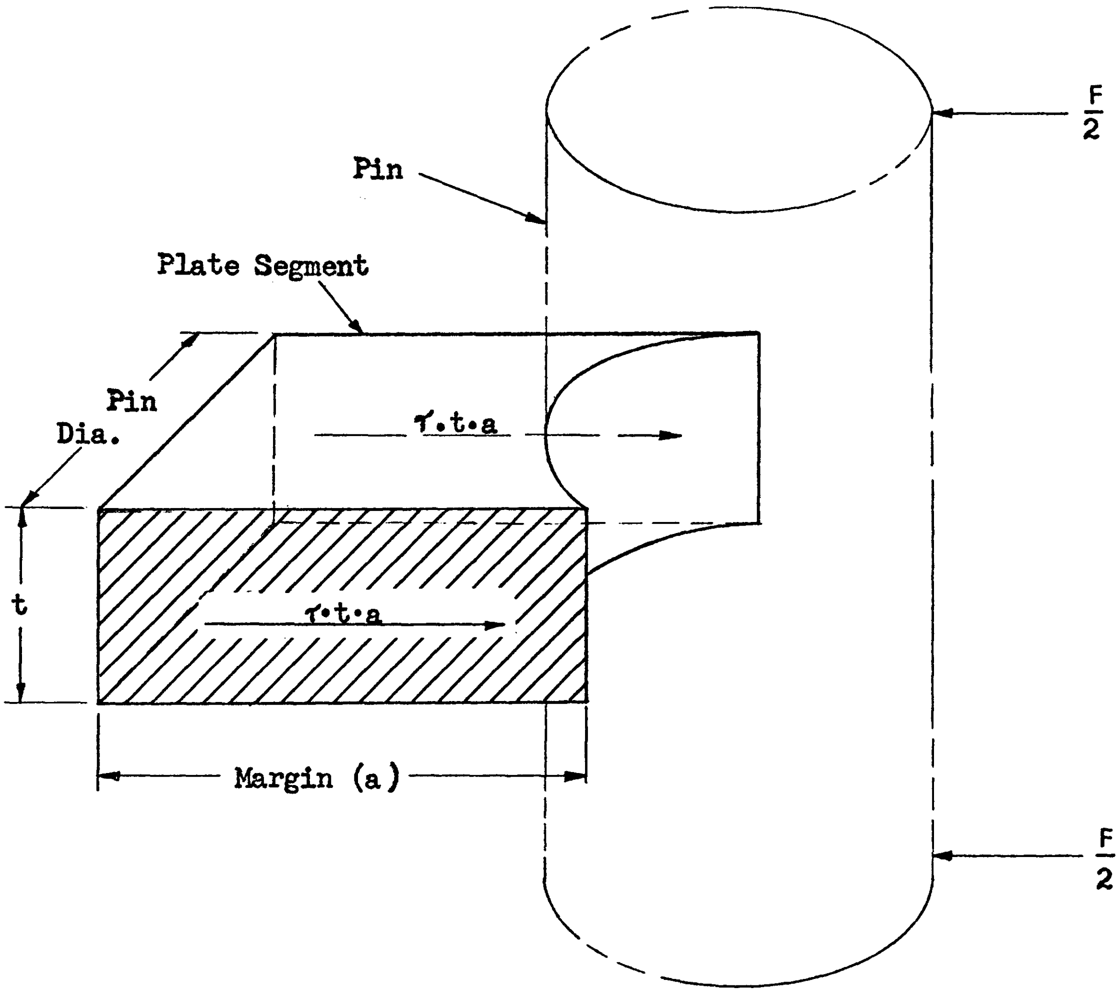
The pins were checked for deflection at the ends (see Appendix I). If the deflections were found to be excessive at maximum load, more rigidity would have to be incorporated in the device. Calculations for deflection were based on point loading at the ends of the pins. This is a conservative assumption. The deflection between pins at maximum load was calculated to be 0.249 in. This amount of deflection can be tolerated without having the jacks begin to move away from the plate-specimen.

The contact stresses between the pins and the bearing plates were calculated to be 274,000 psi. This value is below the bearing strength of the pins, but considerably in excess of that for bearing plates. Therefore, some plastic deformation of the bearing plates was expected. Any deformation of the plates rapidly lowers the bearing stress because the load begins to be distributed over a finite area rather than a line.

The required margin was calculated to be 1.33 in. minimum (Appendix I).

Location of Pins

The prime requirement for pin location is that the pins be located so that loading of the specimen will be as uniform as possible. This geometric requirement was



Approximate Scale: 1 = 1

Figure 3. Free Body Diagram for Calculation of Minimum Margin.

met by assuring the following points:

1. Equal distance must be maintained between adjacent upper pins and equal distance between adjacent lower pins.
2. Constant vertical distance is required between each pair of upper and lower pins.
3. Each upper pin must be vertically above its corresponding lower pin.
4. The longitudinal axes of upper pins must all lie in one horizontal plane.
5. The longitudinal axes of lower pins must all lie in one horizontal plane.
6. The upper pins must extend completely across the head of each jack. The lower pins must extend completely across the base of each jack.

In order to assure that strain in the specimen is not affected by the strain concentrations due to the pins, Saint-Venant's principle was used. The use of Saint-Venant's principle is quite qualitative. It states that the effect of a localized stress disturbance will die out in a distance given by the product of the critical dimension by some small whole number. The critical dimension is obviously the pin diameter. However, the small whole number is not known. In most engineering situations, this number is one, and to a much lesser degree it is two or more.

Since a higher safety factor is required when the calculated design value is uncertain, a "safety factor" of three was used in the determination of pin placement.

The most likely whole number to use is one. The critical dimension — the hole diameter — was therefore multiplied by three. The nearest pin from any strain gage location is 6 inches.

The final criterion for pin location was that the pin could be no closer than one pin diameter from any free surface. This criterion is quite arbitrary and is based more on engineering judgment than on an engineering standard. Compliance with all design requirements resulted in pin locations as shown in Figure 2.

Design of Hydraulic System

The hydraulic system was designed around high-pressure tees which were available. The tees were manufactured by Parker-Hannifin, Cleveland, Ohio, and were described as:

"8BTX Male Branch Tee, stainless steel, 37° flare."

The hydraulic system pressure at full load was calculated by measuring the ID of the cylinder of one of the jacks:

$$P = \frac{40{,}000\#}{\left(\frac{\pi}{4}\right)(2.36\text{in.})^2} = \underline{9140 \text{ psi}}$$

A correction was made for the increased size of the cylinder at full load and a value of 9090 psi was obtained which is not a significant difference and was therefore ignored. This value was checked later with actual gage measurements of 9150 $\pm$ 50 psi, which is excellent agreement. The value of pressure used in the design of hydraulic components was 10,000 psi. Standard manuals[2,3] were used in the design of all hydraulic components. The design proceeded from a basis of the available tees and the 10,000 psi maximum pressure. The tube OD was limited to 0.500 in. since the tees were of a compatable size. The required wall thickness of type 304 hydraulic grade stainless steel was 0.065 in., which included a safety factor of 2.0. However, 0.070-in. tubing was used since this larger size was standard.

In order to utilize an available high-pressure gage (0-15,000 psi) and valve in the system, two tube reducers were used in the system. The tube reducers were also manufactured by Parker-Hannifin of Cleveland, Ohio, and were described as:

"8-4 TRBTX-S Tube reducers,
1/2 in. to 1/4 in., steel, 37° flare."

The ten hydraulic jacks, each having a 20-ton capacity, were standard manually-operated jacks. Figure 4 shows a typical jack with inset showing detail of the release valve. These jacks were built in Germany and therefore had metric threads. There were therefore no commercial fittings to connect the tees to the high-pressure reservoirs. These were designed as shown in Figure 5 with

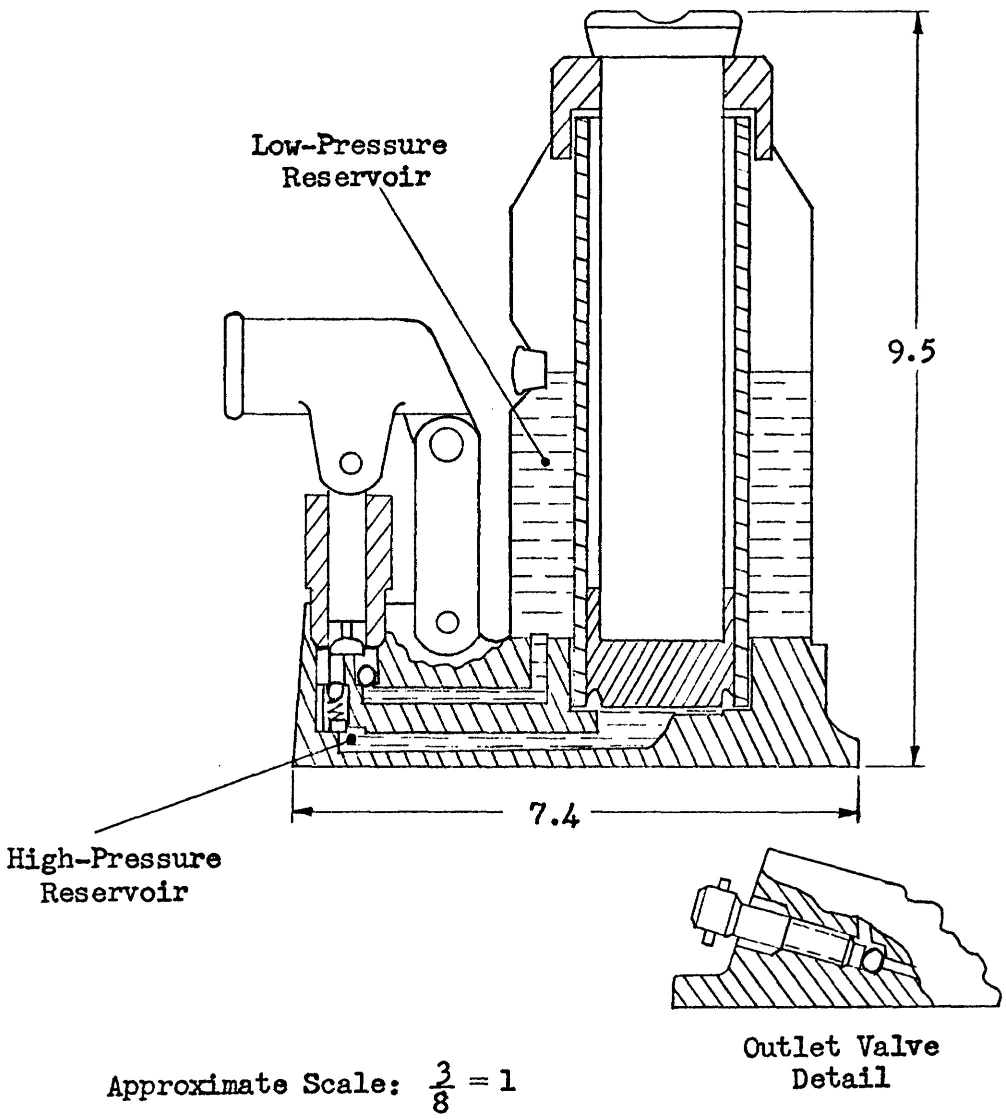
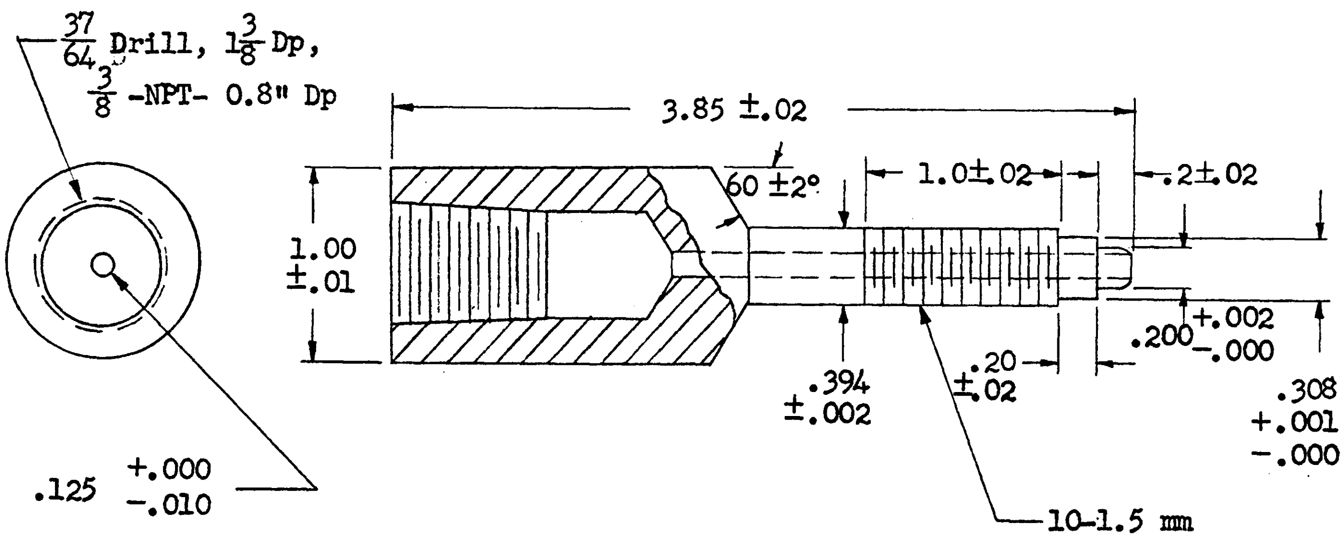


Figure 4. Typical Hydraulic Jack with Outlet Valve Detail.



Scale: 1 = 1

Figure 5. Fitting Used to Connect Tee to High-pressure Reservoir.

installation detail shown in Figure 6.

An O-ring with a back-up ring was used to seal off high-pressure fluid from escaping to the atmosphere or to the reservoir of the jack. The O-rings and back-up rings were manufactured by Parker Seal Company of Culver City, California, and Cleveland, Ohio, and were described as:

"O-ring, 2-008, N219-7, Buna."

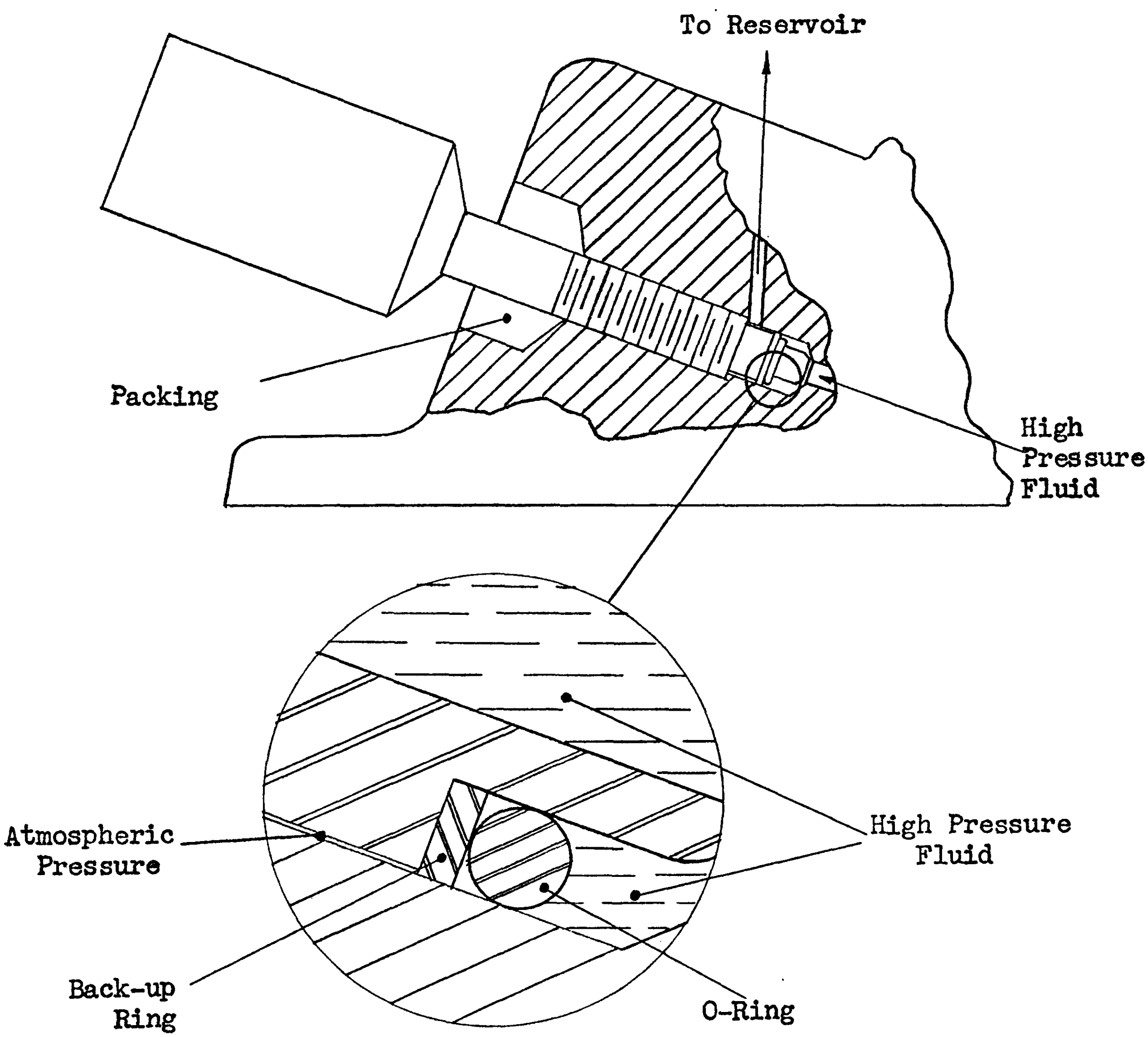
and "Back-up ring, 8-008, N300-9, Buna."

Design of Specimen

It was decided for convenience to use the center of the plate as the specimen since concentric loading could be checked on any specimen. However, the full width of the plate could not be used since there would be more stress — and possibly failure — on a plane through the pin holes.

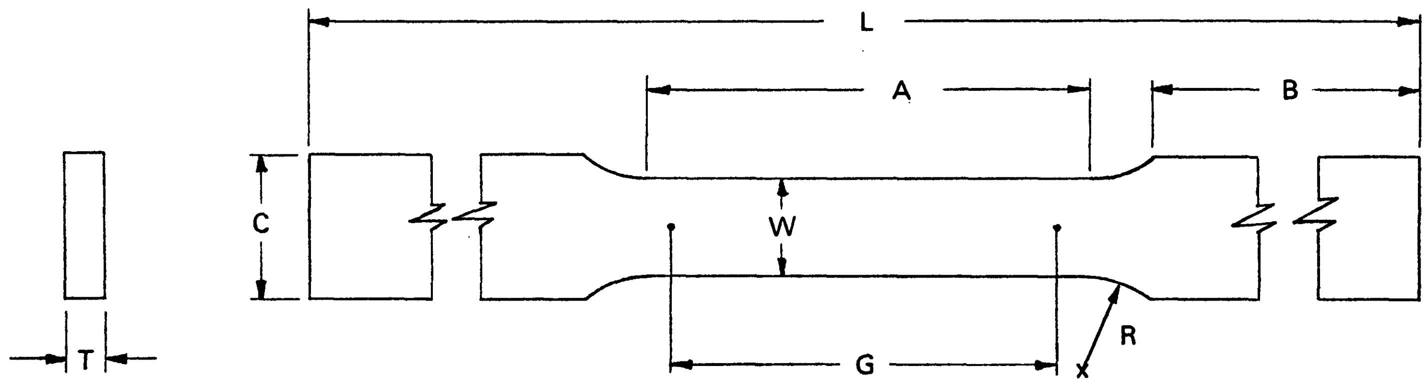
An accepted standard(4) for plate specimens is given in Figure 7. If one were to make a geometrically similar specimen using the width of plate required in this device, the total plate length would have to be approximately 24 feet. This specimen would have a gage length of 6 feet and a width of 1.25 feet. Presumably such a specimen would have no significant stress concentrations. Such a specimen is not feasible for

Scale: 1=1



Scale: 10 = 1

Figure 6. Detail of a Typical Fitting Installation.



G = 2.000 ±.005 in.
W = 0.500 ±.010 in.
T = thickness of specimen
R = $\frac{1}{2}$ in. minimum

L = 8 in. minimum
A = $2\frac{1}{4}$ in. minimum
B = Length of grip section, 2 in. minimum
C = Width of grip section, approximately $\frac{3}{4}$ in.

Scale: $\frac{1}{2} = 1$

Figure 7. ASTM Standard Plate-type Specimen.

two important reasons. Of major importance, as has been stated, is the requirement to limit the size of this smaller device so that a 30,000,000-lb capacity device will not be unwieldy. A geometrically similar specimen for the smaller device would already be unwieldy. The second limitation is seen in the possible buckling of the hydraulic ram of the jack. The maximum L/D ratio to prevent buckling[5] in the jack ram is given by:

$$L/D = \sqrt{\frac{C \quad \pi^2 E}{8 \quad Sy}}$$

L/D = maximum safe $(\frac{\text{length}}{\text{diameter}})$ ratio

C = constant (1 for column ends free to rotate only)

π = 3.1416

E = Modulus of elasticity (30×10^6 psi)

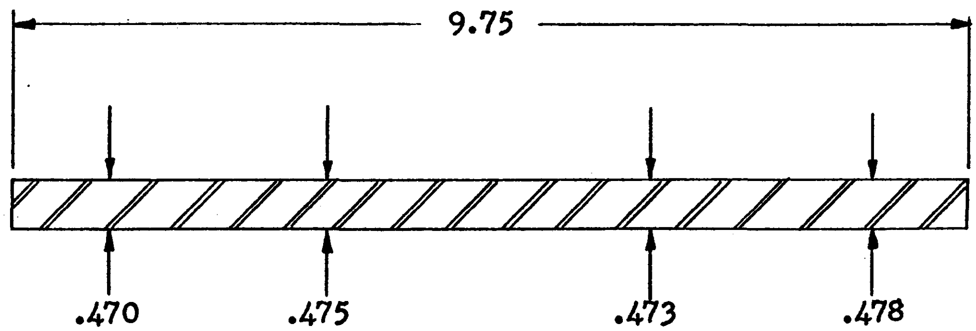
Sy = stress on column (22,600 psi)

L/D = <u>40.4</u>

The use of a 24-foot plate would give an L/D ratio of approximately 184, which far exceeds the maximum safe value. The ratio of 40.4 limits the distance between pins to approximately 60 in. plus jack height of 10 in. or about 70 in. This limitation does not allow a geometrically similar specimen.

Since five 2-in. pins remove 10 in. of width from the plate, the maximum specimen width would be 20 in. At maximum load the average microstrain would be only 667 or one-third of the 0.2 percent yield strength. It was felt that concentric loading up to only one-third the yield strength would not sufficiently prove that loading was truly concentric. Also, as in most tensile tests, there is some bending in the specimen so that the strain is not entirely tensile until perhaps several ten thousandths of strain, which further decreases the accuracy of concentricity measurements. It was decided to make the specimen of a cross-sectional area small enough so that the ultimate tensile strength could easily be attained just before reaching the 400,000-lb capacity of the device. For 65,000-psi UTS material, the maximum cross-sectional area is 6.15 sq in. A 5-sq-in. cross-section was decided upon (see Figure 8). An approximate+ theoretical stress concentration factor, K_t, was obtained[6]. The value from Neuber's nomograph was 1.50. The value from the photoelastic method was 1.53. These values are essentially the same since both values were obtained from graphs which are no more precise than ± 0.03.

+The value of K_t obtained was approximate because the design of the fillets was not a standard design. However, the fillet design was near enough to standard to give a maximum error no greater than 0.1 or 7 percent.



Average Thickness = 0.474 in.
Area = (0.474 in.) (9.75 in.) = <u>4.62 in.^{2}</u>

Scale: $\frac{1}{2} = 1$

Figure 8. Cross-section of Specimen with Actual Dimensions.

Since it was more convenient to actuate only one hydraulic jack, a large hydraulic reservoir was used to supply the oil to that jack. The jack was operated by hand.

The completed tensile-testing device is shown in Figure 1. One-inch plates were placed between the jacks and the pins for three reasons — to lower contact stresses[++], to keep the jacks from slipping to one side on the pins, and to hold the jacks in fixed position with respect to each other by bolting them to the bottom plates.

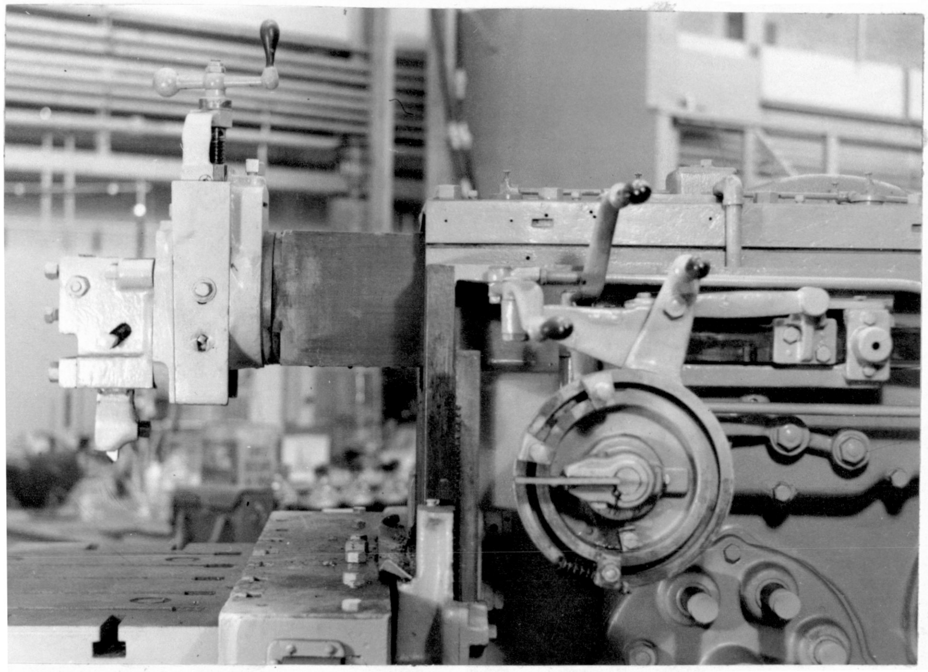
[++]Maximum principal contact stress was calculated as 274,000 psi compression. This stress considerably exceeds the design bearing stress of 90,000 psi and would damage the jack heads.

EXPERIMENTAL PROCEDURE

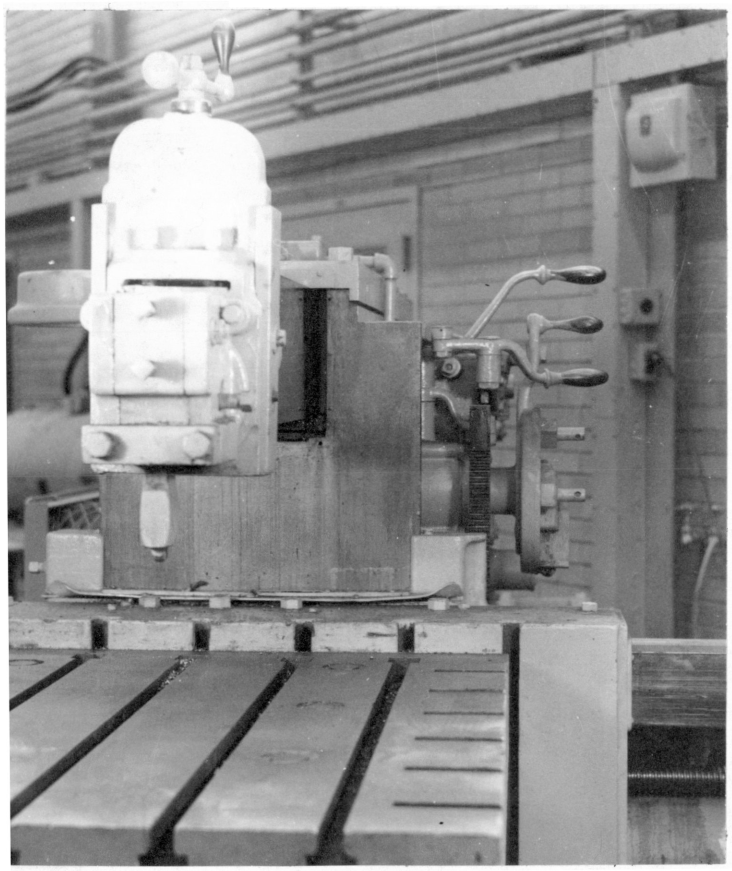
The experimental procedure was straight forward after the design was completed. The parts were fabricated, heat treated, assembled, and tested as explained below. The results of the concentricity test are shown in the last section.

Equipment Manufacture

The plate specimen as shown in Figure 2 was flame-cut by a local firm from a 1-in.-thick mild steel plate in such a manner that the tensile axis coincided with the rolling direction. No warpage from the flame cutting was noted. Then the ten pin holes were drilled. The material properties[7] are shown in Table 1. The center of the reduced-size plate (in the region tnat was to be used for the specimen) was then reduced in thickness to about 0.5 in. by a 42-in. draw shaper. Since the cutting tool on the shaper was improperly sharpened, it was necessary to finish the specimen surface with a portable belt sander. The final specimen thickness, measured at four locations, averaged 0.474 $\pm$0.004 in. The final width at the narrowest location



Side View



Front View

Figure 9. 42-inch Draw Shaper

was 9.75 in.

The plate specimen was not heat-treated after it was machined to proper size. It was used in the hot-rolled condition in which it was received. The same is true of the bearing plates.

The pins were heat treated so that their high potential yield strengths might be utilized. The pins were austenitized and rapidly oil quenched to approximately R_c 51 surface hardness. They were then tempered at 450°F to a surface hardness of 49 R_c to give a ratio of 0.9 of yield strength to tensile strength.

Equipment Assembly

Strain gages were installed by careful preparation of the specimen surface with a final acetone cleaning. Extreme cleanliness is an absolute necessity to insure strain-gage adhesion. A commercial cellulose nitrate cement, SR-4, was used to mount the strain gages. Color-coded leads were then soldered to the strain gages. The strain gages were then checked for short circuits and covered with sponge rubber so that they would not be damaged during assembly of the tensile-testing device.

The ten fittings with O-rings and back-up rings attached were screwed into the former locations of the release valves. The ten tees were then screwed into the fittings after coating the pipe threads with a commercial pipe "dope" in the form of tape. After the proper location+ of the jacks on their bearing plate (a 7x36x1 in. thick plate) was determined, the plates and jacks were marked, drilled, and bolted to form two assemblies of five jacks each. The distances between the tees were measured, and the hydraulic tubing was cut. After fittings were placed on each length of tubing, the tubing was flared, and the tubing with fittings was connected to the tees, again using pipe dope.

In a similar manner a high-pressure valve was placed on the end tee of one set of jacks, and a high-pressure gage was placed on the end tee of the other set of jacks. However, since these two pieces of equipment required a 1/4-in. tube, a tube reducer had to be used on one outlet of each tee.

The finished plate specimen was placed on a small wooden platform on wheels, which made the device completely portable. The plate specimen was held upright

---

+Each jack was placed vertically in line with a pair of pins (one upper and one lower) and as close to the plate specimen as possible. The combination of the latter and the angle at which the fitting extends from the jack required the jacks to be placed at about a 15° angle to the plate specimen.

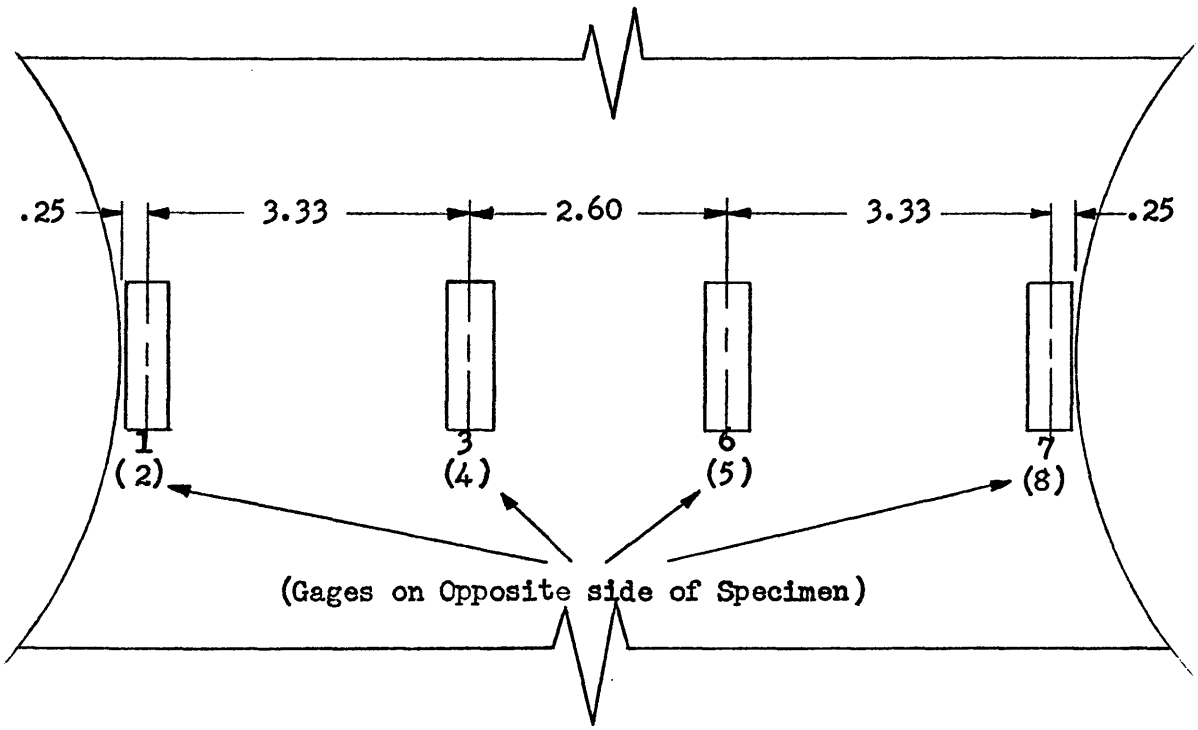
by placing blocks under the lower row of pins on both sides of the device after all five lower pins were inserted. The two sets of hydraulic jacks were then placed on the lower pins. A hydraulic line was then installed, connecting the two sets of jacks on the end opposite to the high-pressure gage and high-pressure valve. Upper pins were inserted and upper bearing plates were installed by placing them between the pins and the top of the jacks after extending the jack extension to within about one inch of the top pins.

Testing

The testing of the specimen can be broken into two parts — the measurement of strain and the actual loading of the specimen.

Strain Measurement

Electrical precision resistance strain gages were used on the specimen to measure load strains. Originally it was planned to have three strain gages on each side — one near each edge and one in the center of the specimen. However, when an eight-channel switch-and-balance strain indicator became available, it was decided to make maximum use of the increased capacity. Four strain gages were used on either side of the specimen, equally spaced as shown in Figure 10.



Scale: $\frac{1}{2} = 1$

Figure 10. Location of Electrical Strain Gages on Specimen.

The strain-gage specifications are as follows:

Gage length:	0.8 in.
Gage factor:	2.07 ± 1/2%
Resistance :	120 ± 0.2 ohm
Type :	80

These were not special gages. They were inexpensive paper-backed gages which were readily available. The only shortcoming of these gages was their gage length, and it is felt that this shortcoming — discussed later — is not serious. The most important requirement for the gages, assuming that they accurately measure strain, is that they have a sufficient strain range to measure the strain in the specimen. The strain gages used had a guaranteed strain range of 0-2000 microstrain. However, several gages exceeded the maximum value and continued to function properly, as can be seen by the continued straight lines of Figure 13 beyond 2000 microstrain.

The Strainsert strain indicator is a null-balance strain indicator and was used with a dummy gage for temperature compensation. According to Dally and Riley[9], for a static-strain application, no other system should be considered. The Strainsert is based on a null-balance Wheatstone bridge, as illustrated in Figure 11. Briefly, the bridge works as follows. Initially the bridge is balanced with $R_1 R_3 = R_2 R_4$ and $R_5 = R_6$ and galvanometer

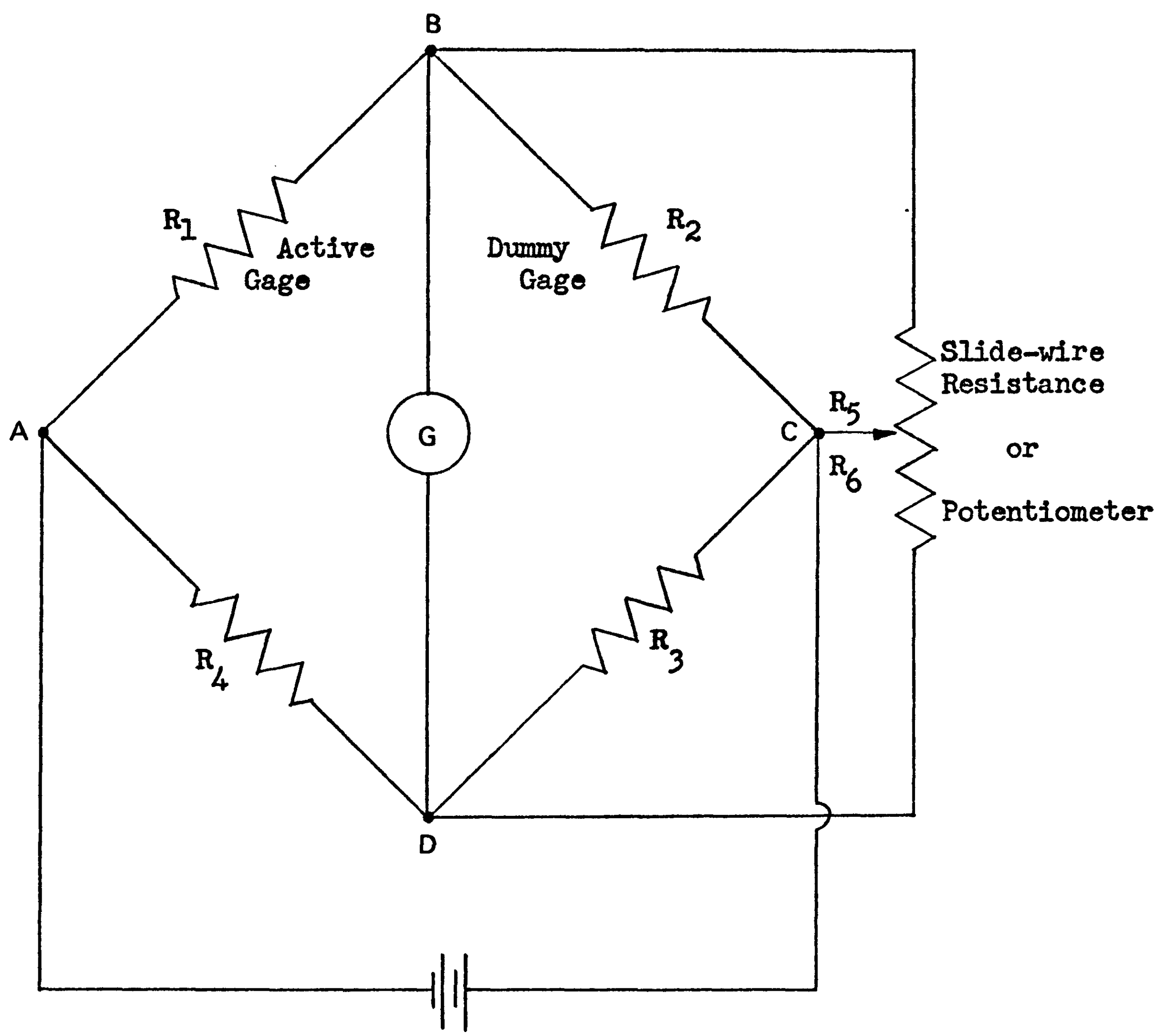


Figure 11. Null-balance Wheatstone Bridge.

reading zero. When R_1 is increased because of a strain, a voltage is indicated on G. The slide-wire resistance is adjusted until the bridge is again balanced, i.e. G registers no voltage. The calibrated adjustment is proportional to the change in resistance of the active gage. The strain can thus be read directly on the potentiometer dial. A similar explanation can be used to show how the dummy gage compensates for temperature changes.

Specimen Loading

A reservoir containing hydraulic fluid was placed above the tensile-testing device, and a siphon tube was filled and run from the reservoir into the reservoir for the jack which was to be used for pumping hydraulic fluid throughout the system. This jack was on the opposite end of the system from the high-pressure gage. The reason for this arrangement was to insure that there were no blocked-off fluid passages in the system. If the gage registered an increasing pressure with continued pumping, there could be no blocked passages in the system.

Before actual testing could begin, several preliminary steps had to be taken.

1. All pins had to be measured for equal extension on either side of the plate. The pins were then marked so that a quick visual inspection could ascertain whether or not they were properly located.

2. The jacks had to be checked for alignment under their respective pins.

3. The distance of each set of jacks from the specimen face had to be accurately measured. The uniformity of the distance was checked.

4. Each potentiometer for each strain gage had to be nulled at zero strain.

5. The system had to be pressurized and hydraulic oil bled off until all air was removed from the system. This procedure was facilitated by successively pumping all ten jacks, starting with the jack farthest from the release valve.

6. Finally, the load had to be applied and released a minimum of about five cycles to eliminate residual tensile stresses in the adhesive[9]. After cycling, the potentiometers had to be checked at zero load and zeroed if required.

The actual testing of the specimen was then started. Theoretically all that is necessary is to actuate one jack of the system, and the resulting load on the specimen will be uniform, i.e. no bending of the specimen from either side or either end. However, the strains will not be expected to be uniform along the width of the specimen due to the necessary design stress concentration, which was discussed previously, of approximately 1.5. It would be expected that strain gages located directly opposite each other on either side of the specimen would record the same value.

Practically there are several reasons why the loading of the specimen would probably not (initially) give equivalent strains on opposite sides of the specimen. These are not based on incorrect theory but are based mainly on machining and assembly inaccuracies as would be expected. First, the strain-gage locations are critical, especially those close to the fillet radius where the stress gradient is steepest. Second, the location and straightness of all holes is quite critical. Third, the transverse spacing of the jacks could vary as well as their spacing from the specimen. All of the above difficulties are due to a lack of precision in measuring and/or machining. Another source of inaccuracy comes from plate warpage. This warpage occurs in the plate-specimen and upper and lower bearing plates. A final source of inaccuracy comes from the jacks themselves. Most of this inaccuracy is in the diameter of the cylinder. Since little can be done about this, it is assumed to be small enough of a variation to be neglected. The variability of the friction is likewise probably insignificantly small.

Since complete accuracy in even one of these areas is impossible, a testing adjustment must be made. This adjustment will be possible only if the above inaccuracies are not excessive, and they were determined to be not excessive in this tensile-testing device.

Little strain variation was noted across the transverse face of the specimen. Considerable variation was noted by comparing the average strain on one face to the average of the other face. A shift of approximately 0.05 in. of the entire jack assembly toward the less-strained face gave satisfactory concentric loading, as shown in Figure 13.

The device was not taken to its full design capacity since the specimen would quite probably have fractured before reaching that capacity, and no auxilliary equipment was designed to provide the necessary safety to the operator in case of a fracture failure. The maximum load placed on the device was about 370,000 lb as calculated from pressure measurements, since at this load the strain gages had far exceeded their range and would therefore give erroneous results. It is felt that this load, although only 92 percent of the design load, was sufficient to prove out the device.

The strain rate was non-uniform due to the nature of the loading system. A rough calculation gave an average strain rate of 4.0×10^{-5} per sec. The strain rate was controlled by the rate of the hydraulic system pressure increase. The specimen was loaded to correspond to a hydraulic pressure of 500 psi and held while strain readings were recorded. The specimen was strained to correspond to an additional 500 psi pressure, and so on to the maximum pressure recorded.

After all tests were completed, each piece of equipment comprising the test device was inspected and measured. All pins appeared to be in original condition with no noticeable nor measurable permanent deflection. All hydraulic tubing was the original diameter. Holes in the plate were still round as closely as could be measured. Upper bearing plates had slight indentations, which were to be expected from the extremely high contact stresses (-274,000 psi principal stress).

Although no specific safety mechanism was built into the device, the design was such that the weakest links in the device for catastrophic failure are the junctions of the tubing with the tees. This statement is true if the assumption is made that the specimen is closely watched so that appreciable necking does not occur at the fillets. In order to provide safety to the operator against a hydraulic failure at these junctions, large cloths were wrapped around them. One failure at a junction actually did occur at a gage pressure of approximately 5500 psi. An analysis of the failure showed a collapsed flare due to insufficient flaring. All tubing was then removed and flared to the maximum extent. No subsequent failures occurred up to 8500 psi, the maximum pressure obtained in the system.

Data

The high-pressure gage was calibrated by first attaching it to a hydraulic jack. The jack was then placed in a tensile-testing machine and loaded to 40,000 lb. Load readings were taken at 1000-psi intervals, and a graph was drawn to convert gage pressure to load (see Figure 12). Load obtained from this graph could be compared to the load obtained from strain-gage measurements.

Data from the eight strain-gage readings are shown in Figure 13. The strain at which yielding of the specimen began can be seen by noting the deviation of the curves from a straight line. The curves fall into two distinctly different groups according to their slope. This behavior was anticipated, since at a given total load the strain gages located nearer the root of the fillet will be more highly strained than those further from the root.

A very interesting feature of the curves that was not anticipated is the location of the yield point (deviation from a straight line). Figure 13 shows that apparently the yield point for all curves occurs at about 6500 psi or 275,000 lb total load rather than at a given value of strain (and therefore stress). This apparent discrepancy will be discussed later.

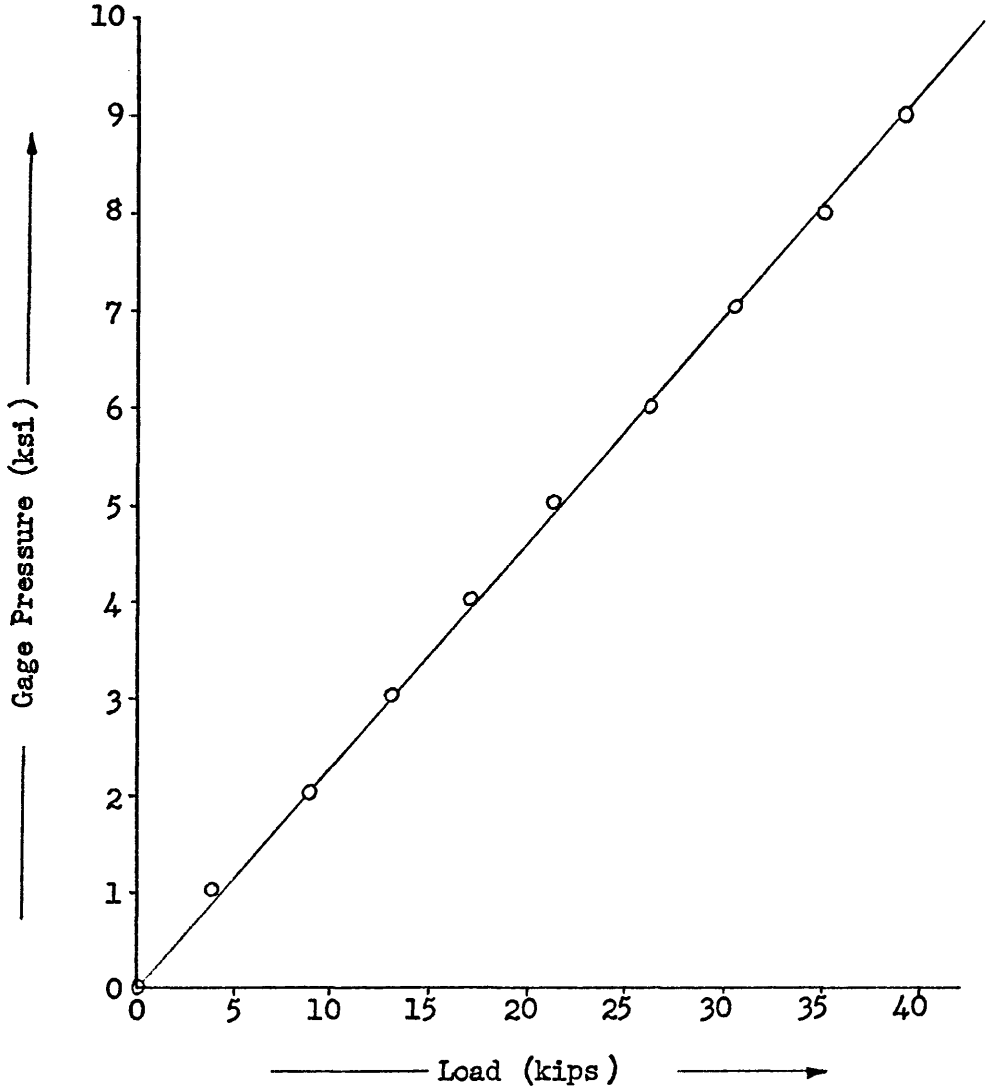


Figure 12. Graph for Converting from Gage Pressure to Load.

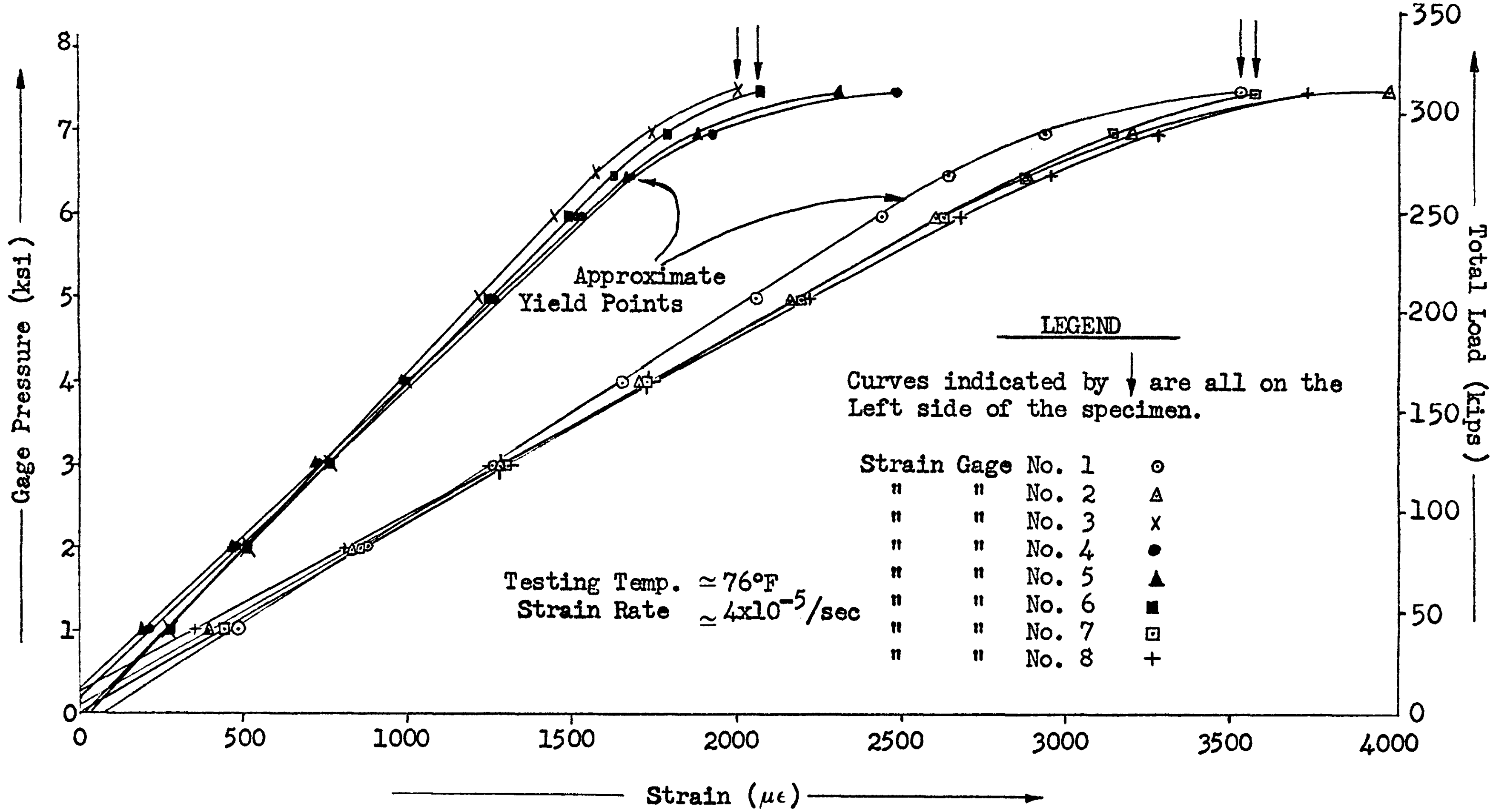


Figure 13. Load-Strain Curves Showing Degree of Concentricity of Loading of Specimen.

RESULTS AND ANALYSIS

The concentricity of loading gave results which exceeded the required concentricity. This degree of concentricity was obtained with only very little repositioning of the jacks.

Theoretical Background

The bases for the design were very simple, being equilibrium of force, equilibrium of moment, and Pascal's principle — all of which come from basic physics.

Equilibrium of force was used by taking each jack as a free body. The force of the ram against an upper pin must equal the force of the base of the jack against the lower pin for each jack, neglecting the weight of the jack, which for a 20-lb jack is only 1 percent of the load at a load of one ton. This principle was also applied to the specimen. The error due to the weight of the plate and pins above the specimen was approximately 80 lb or 0.4 percent at a load of one ton per jack.

Equilibrium of moment was used to eliminate any bending of the specimen in either the width or thickness direction. The extent to which this equilibrium was attained can be ascertained by comparing the curves of Figure 13.

Pascal's principle was relied upon to provide an equal pressure throughout the system. If one assumes that this principle holds and that friction is negligible, the forces exerted by the rams will be equal.

Data Interpretation

Data for this experimental work are shown in Figure 13. Several important pieces of information can be obtained from this data.

1. The concentricity of the load on the specimen can be determined by measuring the standard deviation of the curves from an average curve for each set. The maximum standard deviation occurs near the yield points for both sets of curves. The deviations measured in terms of strain are converted to stresses. The maximum standard deviation for the inner strain gages at 275,000-lb load is 48 microstrain or 1440 psi from the average of 1635 microstrain or 49,000 psi. This produces edge strains that are about 3.5 percent different from centerline strains for the inner portion of the specimen. A similar calculation gives 4.5 percent eccentricity at the fillet root. Examination of the curves shows that the principal cause of this

eccentricity is due to one strain gage. By disregarding the reading from that strain gage, an eccentricity of 1.5 percent is obtained. Formal statistical tests[10] show that the strain gage is an outlier at a 0.95 significance level. The reason for this is not certain, but probably the alignment of the gage with the tensile axis was not precise enough. This alignment is rather critical in high-stress gradient regions such as where this strain gage was located.

For comparison, concentricity measurements on cylindrical specimens using standard ball seat fixtures have been shown[11] to give eccentricities of approximately 30%.

2. The second point to notice is that all gages on the left side of the specimen are slightly but consistently less strained than those on the right side above a load of approximately 175,000 lb. This effect on eccentricity could be remedied by a very slight repositioning of the hydraulic jacks. However, it is felt that the eccentricity was at a low enough value without this minor adjustment; therefore the adjustment was not made.

3. The final point of interest is the location of the yield points, as was mentioned previously. The yield point, as determined from the strain by using

Young's modulus, for the steel specimen at the location of the inner gages is 48,800 psi, whereas the yield point for the steel near the fillet root is 79,000 psi. Table 1 showed that the material was homogeneous as determined by the yield point to within ± 1,000 psi, which is only 3 percent of the variation observed here! Obviously inhomogeniety is not the answer. The answer apparently lies with the differences in the restraint at the different locations. Wessel's[12] data show a room-temperature increase in yield strength of approximately 47 percent for notched mild steel. This value is somewhat less than the approximately 83 percent increase observed in this specimen. However, his specimens were much smaller, and therefore it would be expected that his specimens did not have the same amount of restraint as in this case. The result is not due to gage errors since the measurements agree with the modulus of elasticity within approximately 2 percent as calculated from Figure 13.

Error Analysis

Most errors in the various parts were discussed when the design of those parts or the testing of the specimen was explained. Although there were many sources of possible error in construction, none were so excessive that satisfactory results could not be obtained by a slight adjustment of jack location.

A deviation of 0.005 in. was measured in the horizontal alignment of pin holes. This deviation caused unequal loading to the extent that the bearing plates had to bend twice that amount before the corresponding pin began to transmit load to the specimen.

The horizontal spacing of pin holes was accurate to within measuring capabilities ($\pm$ .010 in.).

The pressure gage was accurate to within approximate $\pm$ 50 psi. This pressure value is equivalent to an error in total load of $\pm$ 2250 lb. The curves of Figure 13 are therefore in error by more than 5 percent below a total load of 45,000 lb (1100 psi gage pressure). They are increasingly more accurate with increasing gage pressure up to 7500 psi where the maximum error due to gage readings is 0.7 percent.

The best accuracy that can be expected with an electrical strain-recording system is 2 to 3 percent according to Stein[13]. It is felt that the error is certainly no less than 3 percent, since this was the first time the author has installed electrical strain gages and used them to determine strains. Probable error in location of one strain gage was discussed previously under Data Interpretation.

The measured amount of deflection at 92 percent of full load was 0.15 in. compared to the calculated value of 0.264 in. at full load. Although not strictly an error, the comparison of these two values show quantitatively the degree of conservativeness in assuming point loading at the ends of the pins.

Most significant sources for error lie in precision of machining and location of components of the tensile-testing device. And, as has been stated, these were all corrected by slight relocation of the hydraulic jacks. What this implies, of course, is that the original location of the jacks <u>cannot</u> be as precise as may be desired; otherwise no movement to correct for machining errors is possible.

Of passing interest was the comparison of the measured stress concentration factor with the theoretical value of 1.50. Figure 14 shows the average strain in the cross section as well as the strain distribution. The strain concentration determined from this graph is 1.48. The measured value is less than 1 percent lower than that obtained from Neuber's nomograph.

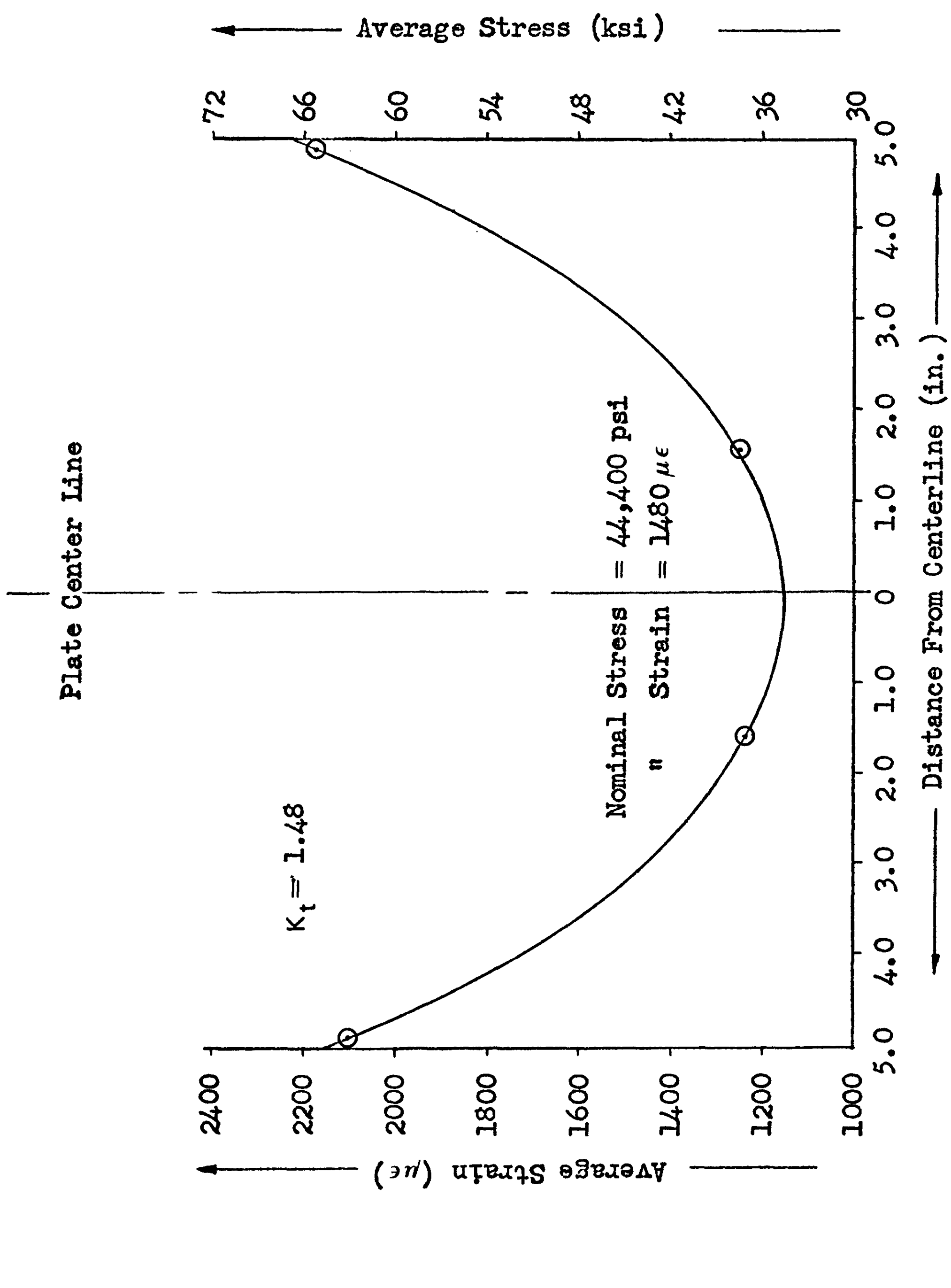


Figure 14. Graph for Determination of Stress Concentration Factor, K_t.

Further Study

Since the tensile-testing device was built to determine feasibility of a much larger device, no further design is required, because it has been successfully proven that sufficient concentricity of loading is possible with such a device. Further use of the device is planned, however. Brittle fracture studies will be conducted to investigate the gross strain concept.

What must be done, however, is to design and construct a full-scale (approximately 30,000,000-lb) device so that specimens of the size to be used in large steel structures can be tested. The complete design of this full-scale device is not intended to be a part of this report. However, rough calculations of both the approximate size and the approximate cost are included here.

1. 400,000-lb capacity tensile-testing device

A. Weight

1. Hydraulic Jacks (10 at 15 lb)	150 lb
2. Plate-specimen	170 lb
3. Bearing plates	200 lb
4. Pins	100 lb
Total Weight	620 lb

B. Cost

1.	Machining (100 hrs. at $10/hr.)	$1000
2.	Hydraulic Jacks (10 at $40 each)	400
3.	Heat Treating (Specimen and Pins)	50
4.	Tubing, fittings, valve, gage, misc.	150
5.	Materials: Plate-specimen, Pins, Support and Stress-distribution Plates	200
	Total Cost	$1800

2. 30,000,000 lb tensile-testing device

A. Weight

1.	Hydraulic Jacks (20 at 150 lb)	3,000 lb
2.	Plate-Specimen	65,000 lb
3.	Bearing Plates (Wide-flange Beams)	5,000 lb
4.	Pins	30,000 lb
	Total Weight	100,000 lb

B. Cost

1.	Machining (1000 hrs at $10/hr)	$ 10,000
2.	Hydraulic Jacks (20 at $4400 ea.)	88,000
3.	Heat Treating (Specimen and Pins)	10,000
4.	Tubing, fittings, valve, gage, misc.	2,000
5.	Materials: Plate-specimen, Pins, Bearing Plates	40,000
	Total Cost	$150,000

This projected cost and weight for a 30,000,000-lb device are very rough estimates and should not be considered more accurate than about ± 50 percent since actual costs were not obtained on any equipment except the hydraulic jacks. Also, no design costs were included. However even if the costs and weights do exceed the estimates by 50 percent, the total cost would be only $225,000 and the total weight 150,000 lb. These values are extremely low when one considers the capacity of the device. The cost of a 50,000-lb capacity commercial tensile-testing machine is approximately $15,000. The projected cost for a 30,000,000-lb commercial testing machine is \$9,000,000, assuming such a machine were available, which is not the case. This cost is approximately 60 times the estimated cost of the above 30,000,000-lb device. The concession one has to make for the tensile-testing device over a commercial tensile-testing machine is, of course, versatility.

CONCLUSIONS

1. With proper positioning of the jacks, the tensile testing device produced edge strains that were about 3 percent different from the centerline strain. This represents an improvement of a factor of ten over what is obtained with standard ball seat fixtures, which typically give edge strains 20-40 percent different from centerline strain.

2. This device weighed 620 pounds and would cost \$1800 to have it built. It is estimated that a geometrically-similar device of 30,000,000-pound capacity would weigh less than 100 tons and would cost less than \$250,000. Because of problems of hardenability of steels, there would have to be some minor changes in design.

3. For a notched specimen, the measured stress concentration factor of 1.48 compares favorably with values of 1.50 and 1.53 found in the literature.

4. Because of the restraint of the enlarged ends of the test specimen, the average elastic limit for the notched specimen was 80 percent higher than for an unnotched specimen. This is surprisingly high, but is not due to gage errors, since the measurements agree with the known modulus of elasticity of 30×10^6 psi.

APPENDIX I — DESIGN CALCULATIONS

Design of Pins

Against Bending Failure

Flexure Formula: $S = \dfrac{M\ c}{I}$

S = maximum fiber stress (psi)

M = maximum moment (in-lb)

c = distance from neutral axis to extreme fiber (d/2 in.)

I = moment of inertia (in.^{4})

$I = (\pi/64)\ d^4$ for rounds where d is diameter of the round.

$$150{,}000 = \frac{(2.5)\ (40{,}000)\ d/2}{\left(\frac{\pi}{64}\right) d^4}$$

$d = 1.9$ in.

Against Shearing Failure

$$\tau = \frac{F_s}{A} = \frac{F_s}{\left(\frac{\pi}{4}\right) d^2}$$

τ = maximum shearing stress (1/2 yield design strength-75,000 psi)

F_s = maximum shearing force (40,000 lb)

A = shear area (in.^{2})

d = pin diameter (in.)

<u>d = 0.82 in.</u>

<u>Against Bearing Failure in Plate</u>

$$S_b = \frac{F_b}{A} = \frac{F_b}{t \times d}$$

S_b = maximum bearing stress (use 1/2 of allowable from Table 1-45,000 psi)

F_b = maximum bearing force (40,000 lb)

A = bearing area (in.^{2})

t = plate thickness (1 in.)

d = pin diameter (in.)

$$d = \frac{F_b}{S_b}$$

<u>d = 0.89 in.</u>

<u>Against Deflection for 1.9 in. diameter pin</u>

A. Top pins

$Y_{Total(Top)}$ = bending + shear

$$Y_{Total(Top)} = \frac{PL^3}{3EI} + \frac{3PL}{2AG}$$

$Y_{Total(Top)}$ = Total deflection from bending and shear of top pins (in.)

P = load (40,000 lb)

L = length of pin from plate (2.5 in. for top and 7.0 in. for bottom pins)

E = Youngs Modulus—30 x 10^6 psi

G = Shear Modulus—12 x 10^6 psi

A = cross-sectional area of pin (2.84 $in.^2$)

I = moment of inertia (0.640 $in.^4$)

$$Y_{Total(Top)} = \frac{(40,000)(2.5)^3}{(3)(30x10^6)(\frac{\pi}{64})(1.9)^4} + \frac{(3)(40,000)(2.5)}{(2)(2.84)(12x10^6)}$$

$$Y_{Total(Top)} = 0.0108 \text{ in.} + 0.00440 \text{ in.}$$

$$Y_{Total(Top)} = \underline{0.0152 \text{ in.}}$$

B. Bottom Pins

$$Y_{Total(Bottom)} = \frac{(40,000)(7.0)^3}{(3)(30x10^6)(\frac{\pi}{64})(1.9)^4} + \frac{(3)(40,000)(7.0)}{(2)(2.84)(12x10^6)}$$

$$Y_{Total(Bottom)} = 0.237 + 0.01232$$

$$Y_{Total(Bottom)} = \underline{0.249 \text{ in.}}$$

<u>Contact Stress Calculation On Pin</u>

$S_b = -2 \ \mu \ (\frac{b}{\Delta})$ (Ref. 6, p.366)

S_b = bearing stress (psi)

μ = Poisson's ratio (0.33)

b = $\sqrt{\frac{2_q \Delta}{\pi}}$ = semi-minor axis of contact ellipse (in.)

q = load per inch of beam = $\frac{40,000 \text{ lb}}{2.25 \text{ in.}}$ = 17,800 lb/in

q = rigidity factor($6.54 \times 10^{-8} \frac{\text{in}}{\text{psi}}$)

S_b = <u>-274,000 psi</u>

Design of Margin

$$a = \frac{F_b}{2 \times \tau \times t} \quad \text{(Ref. 1, p. 437)}$$

a = minimum margin (See Figure 3) to prevent tearout (in.)

F_b = maximum bearing force (40,000 lb)

τ = maximum shearing stress (15,000 psi)

a = 1.33 in. minimum

LITERATURE CITED

1. Higdon, A., Ohlsen, E. H., and Stiles, W. B., Mechanics of materials: New York, Wiley & Sons, 502 p. (1960).

2. Parker Hannifin - Parker Tube Fittings Division, Tube fittings manual: Cleveland, Ohio, (1962-1967).

3. Parker Seal Company, O-ring handbook: Culver City, California, (1965).

4. American Society for Testing and Materials Standards, Part 31: Philadelphia, Pa., ASTM, p. 8-10 (1968).

5. Hall, A. S., Jr, Holowenko, A. K., and Laughlin, H. G., Schaum's outline of theory and problems of machine design: New York, Schaum Publishing Co., p. 47 (1961).

6. Seely, F. B., and Smith, J. O., Advanced mechanics of materials: New York, Wiley & Sons, 680 p. (1967).

7. Rosenstein, A. H., Brittle fracture in welded plate specimens: Colorado School of Mines, Thesis No. T-1014, p. 22 (1964).

8. Clark, D. S., and Varney, W. R., Physical metallurgy for engineers: New Jersey, Van Nostrand, 629 p. (1962).

9. Dally, J. W., and Riley, W. F., Experimental stress analysis: New York, McGraw-Hill, 520 p. (1965).

10. Johnson, N. L., and Leone, F. C., Statistics and experimental design: Volume 1: New York, Wiley & Sons, 523 p. (1964).

11. Ebert, L. J., Lubahn, J. D., and Sachs, G. D., "Notched Bar Tensile Test Characteristics of Heat-Treated Low Alloy Steels" in Trans. of American Society for Metals: Volume 33: ASTM, p. 340-395 (1944).

12. Wessel, E. T., American Society for Testing Materials: ASTM, p. 56 (1956).

13. Stein, P. K., Metals engineering design: ASME Handbook: New York, McGraw-Hill, p. 479 (1965).