OFFICE OF NAVAL RESEARCH
Contract No. N7ONR-39707
Project No. NR-031 395

THE CONSTRUCTION DETAILS OF AN AUTOMATIC RECORDING TEMPERATURE CONTROLLED TORSION MACHINE

FINAL REPORT
June 15, 1953

by

Sheldon Spachner
and
E. P. Klier

Department of Chemical Engineering
Glenn L. Martin College of Engineering and Aeronautical Sciences
University of Maryland
College Park

GENERAL DISCUSSION

In the determination of stress-strain curves by tension testing, stress is usually defined as the force applied per original area; the strain is the change of length with respect to the original length, or the change of area with respect to the original area. A more fundamental relationship may be obtained in the plotting of stress-strain data if the stress is taken as the force applied with respect to the instantaneous area, and the strain is considered to be the change in elongation with respect to the instantaneous length, (1)*. This type of stress and strain is known as true stress σ, and natural strain δ. The relation $\sigma = \sigma_0 \delta^m$ gives an approximation to the curve obtained in true-stress natural strain tests (2), although a curve obtained by use of this equation departs from the experimental true-stress natural strain curve in three intervals, (a) that in which elastic strain obtains, (b) that in which inhomogeneous plastic strain is encountered, and (c) that beyond the maximum load in which the specimen necks down. This expression, however, is useful in essaying the plastic behavior of metals.

It has been found that, in steel, a relation exists between the strain hardening exponent, n, and the temperature of the test, while the temperature, in turn, affects the fracture characteristics of the steel. There is thus indicated a possible relationship between the strain hardening exponent, n, and the fracture characteristics

---

* Underlined numbers in parentheses refer to bibliography at end of paper.

of the steel. Limited data now available (3) (4) indicate that within a given class of steels those steels with the lowest values of n at very low temperatures will have the highest transition temperatures in the impact test. In addition, there are indications that the strain hardening exponent is dependent upon the velocity of deformation (5).

It was the purpose of this work to investigate the relation of the strain hardening exponent to specimen temperature and strain rate. This would be accomplished by subjecting specimens to torsional forces at controlled temperature and strain rate, with torque-twist curves being obtained from these tests. By utilization of formulae developed in the following sections, it is possible to convert the torque-twist data to effective-stress effective-strain data, and therefrom obtain the strain hardening exponent. Since plastic theory dictates that a specimen subjected to tension will yield the same effective-stress, effective-strain curve as one subjected to torsion, a prediction may be made of the behavior of these specimens in tension.

Since under ideal conditions tension data would be desired in the evaluation of variations in the strain hardening characteristics of a steel with temperature and strain rate, the reasons for the use of torsion data will be advanced. Chief of these reasons is the brittleness of many steels when tested in tension at very low temperatures. This brittle range frequently extends to sufficiently high temperatures that the needed stress-strain data cannot be obtained in the tension test. A second major factor is the question of strain rate control. In the tension test elaborate control equipment is required to insure constant strain rates through the course of the test. This problem is not of major moment in the torsion test where a constant rate of twist insures constant strain rate.

The torsion machine which was constructed to allow the determination of the torque twist data for specimens at controlled temperature and strain rate will be described below following a presentation of the theory of plasticity on which the conversion of the data depends.

THEORY

I. PLASTIC FLOW BEHAVIOR (6) (7)

In the work that follows, stress will be defined as the ratio of the force applied to the instantaneous area (true stress, $\sigma = \frac{F}{A}$), instead of the usual ratio of force applied with respect to an initial area, (nominal stress, $S = \frac{F}{A_o}$). Strain will be defined as

$$\delta = \int_{l_o}^{l_i} \frac{dl}{l} = \ln\left(\frac{l_i}{l_o}\right)$$

instead of by the less exact

$$e = \int_{l_o}^{l_i} \frac{dl}{l_o} = \frac{l_i - l_o}{l_o} = \frac{\Delta l}{l_o}$$

where strain measurement is based upon a reference to the original elongation, l_o. If it may be assumed that the metals being tested are sufficiently fine grained to behave on a macroscopic scale as sensibly isotropic, three generalizations established in elastic theory may be modified to describe macroscopic plastic flow. These relationships are now presented.

A. <u>The Axes of the Principal Stress and Strain Coincide Throughout Deformation.</u>

This may be shown in brief fashion by considering a small isotropic cube which has a stress acting upon it. This stress may be resolved into three normal stresses acting parallel to the axes of the coordinate system that the block is in, viz., σ_x, σ_y, σ_z, and six shearing stresses, τ_{xy}, τ_{yx}, τ_{xz}, τ_{zx}, τ_{yz}, τ_{zy}. Equilibrium conditions demand that the shearing stresses obey the following relations.

$$\tau_{xy} = \tau_{yx} \quad ; \quad \tau_{xz} = \tau_{zx} \quad ; \quad \tau_{yz} = \tau_{zy}$$

The nine stress terms indicate that this matrix is symmetrical. It can be shown that a suitable rotation of the coordinate system containing the nine stresses will transform the number of stresses to three, or that the nine term matrix will be transformed to a three term matrix, - only the diagonal terms remaining.

$$\begin{pmatrix} \sigma_x & \tau_{yx} & \tau_{zx} \\ \tau_{xy} & \sigma_y & \tau_{zy} \\ \tau_{xz} & \tau_{yz} & \sigma_z \end{pmatrix} \longrightarrow \begin{pmatrix} \sigma_1 & 0 & 0 \\ 0 & \sigma_2 & 0 \\ 0 & 0 & \sigma_3 \end{pmatrix}$$

The transformation equation (sometimes called the Secular Equation) is:

$$\sum_{K=1}^{3} \left(T_{jK} - T_i{}' \delta_{jK} \right) \alpha_{Ki} = 0$$

where δ_{jk} is Kronecker's delta, and α is the direction cosine between axes. The three terms, σ_1, σ_2, σ_3, are known as principal stresses.

Hooke's law states that stresses are linear functions of pure strains, provided that the elastic limit of the material is not exceeded.

Strain is a more complicated quantity than stress, resulting in rotation as well as linear displacement. Strain may be represented as a sum of two dyadics. One is termed a pure strain dyadic, which considers linear displacements only, and is symmetric; the other is a non-symmetric rotation dyadic. Since Hooke's law relates only stress and pure strain, and since the pure strain dyadic is symmetric, we may relate three stress terms to three strain terms, δ_1, δ_2, δ_3. These strain terms will be called principal strains. The rotation may be chosen to be equal to that of the stress dyadic. Thus, we have three principal stresses parallel to three principal strains.

B. The Sum of the Principal Strains is Zero.

Consider the element $l_1 b_1 h_1$ bounded by the principal planes before straining. Since elastic strain is small compared to plastic strain in the test the volume may be treated as constant throughout the straining operation, $l_1 b_1 h_1 = l_2 b_2 h_2$ so that,

$$\frac{l_2 b_2 h_2}{l_1 b_1 h_1} = 1$$

Taking natural logarithms,

$$\ln\left(\frac{l_2 b_2 h_2}{l_1 b_1 h_1}\right) = \ln\left(\frac{l_2}{l_1}\right) + \ln\left(\frac{b_2}{b_1}\right) + \ln\left(\frac{h_2}{h_1}\right) = 0$$

or,

$$\delta_1 + \delta_2 + \delta_3 = 0$$

C. Relationship of Principal Strains to Principal Stresses.

Since the volume remains constant, an elongation in one direction must cause a contraction in another direction. Since the material under consideration is isotropic, there will be an elongation in one direction, equal contractions in the other two directions, or

$$\delta_1 = \frac{1}{D}\left[\sigma_1 - N(\sigma_2+\sigma_3)\right]$$

$$\delta_2 = \frac{1}{D}\left[\sigma_2 - N(\sigma_1+\sigma_3)\right]$$

$$\delta_3 = \frac{1}{D}\left[\sigma_3 - N(\sigma_1+\sigma_2)\right]$$

(N = Poisson's ratio in elastic theory)

Since $\delta_1 + \delta_2 + \delta_3 = 0$, N must equal 1/2 for a corresponding euqlity on the right side. The relations between principal stress and principal strain are then given by:

$$\delta_1 = \frac{1}{D}\left[\sigma_1 - \frac{1}{2}(\sigma_2+\sigma_3)\right]$$

$$\delta_2 = \frac{1}{D}\left[\sigma_2 - \frac{1}{2}(\sigma_1+\sigma_3)\right] \qquad (1)$$

$$\delta_3 = \frac{1}{D}\left[\sigma_3 - \frac{1}{2}(\sigma_1+\sigma_2)\right]$$

II. DEVELOPMENT OF EFFECTIVE STRESS - EFFECTIVE STRAIN RELATIONSHIPS

Squaring and adding the equations in section I, part C. yields:

$$\frac{3}{2}\left[\sigma_1^2 - \sigma_1\sigma_2 - \sigma_1\sigma_3 + \sigma_2^2 - \sigma_2\sigma_3 + \sigma_3^2\right] = \delta_1^2 + \delta_2^2 + \delta_3^2$$

Then,

$$\frac{2}{3}\left(\delta_1^2 + \delta_2^2 + \delta_3^2\right) = \frac{1}{D^2}\left(\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_1\sigma_3 - \sigma_2\sigma_3\right) =$$

$$= \frac{1}{2D^2}\left[(\sigma_1-\sigma_2)^2 + (\sigma_1-\sigma_3)^2 + (\sigma_2-\sigma_3)^2\right]$$

Extracting the square root,

$$\sqrt{\frac{2}{3}\left(\delta_1^2 + \delta_2^2 + \delta_3^2\right)} = \frac{1}{D}\left\{\frac{1}{2}\left[(\sigma_1-\sigma_2)^2 + (\sigma_1-\sigma_3)^2 + (\sigma_2-\sigma_3)^2\right]\right\}^{\frac{1}{2}}$$

Effective strain is defined as,

$$\bar{\delta} = \sqrt{\frac{2}{3}\left(\delta_1^2 + \delta_2^2 + \delta_3^2\right)}$$

Effective stress is defined as,

$$\bar{\sigma} = \frac{1}{\sqrt{2}}\left[(\sigma_1-\sigma_2)^2 + (\sigma_1-\sigma_3)^2 + (\sigma_2-\sigma_3)^2\right]^{\frac{1}{2}}$$

Then,

$$\bar{\delta} = \frac{1}{D}\,\bar{\sigma}$$

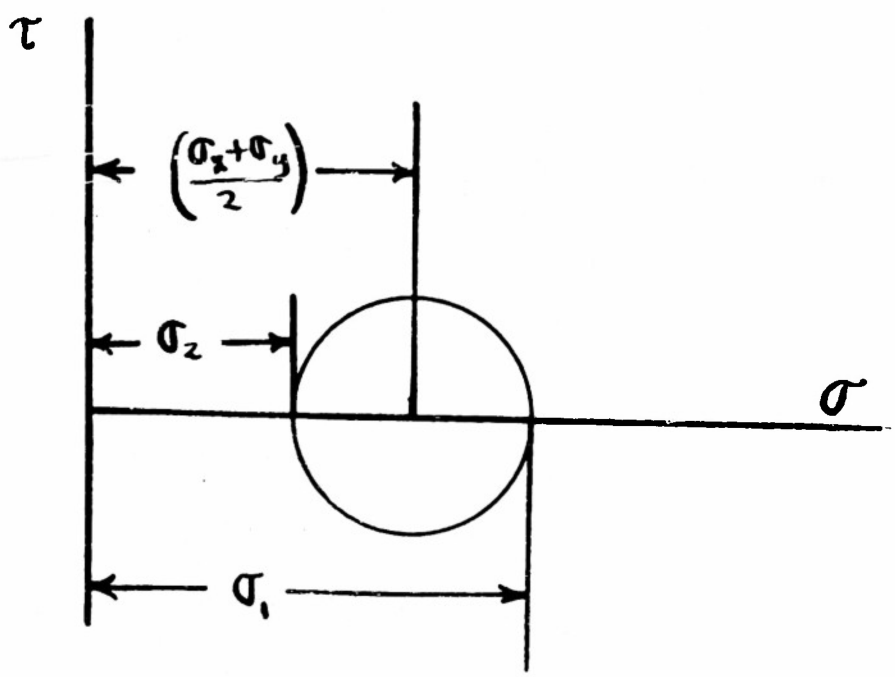
In the case of simple tension, since $\sigma_1 = \sigma$, $\sigma_2 = \sigma_3 = 0$ and $\delta_1 = \delta$, $\delta_2 = \delta_3 = -\frac{1}{2}\delta$. The equations $\bar{\sigma} = \sigma_1$ and $\bar{\delta} = \delta_1$ are then obtained.

III. RELATIONSHIP OF EFFECTIVE STRESS - EFFECTIVE STRAIN CURVES OBTAINED IN TORSION TO THOSE OBTAINED IN TENSION.

A. Expression of Effective Stress in Terms of Shearing Stress

In the case of simple torsion, it can be shown, by use of the stress dyadic, that $\tau_{max} = \frac{1}{2}(\sigma_1 - \sigma_3)$. One principal stress, σ_2, is zero.

The relation that σ_1 and σ_2 will bear to each other may be seen from a Mohr circle representation of these two stresses. (1)



It may be seen from the Mohr circle that for any value of τ, there are two values of σ. When τ is a maximum,

$\sigma = \pm\left(\frac{\sigma_x + \sigma_y}{2}\right)$. Since the values of σ are σ_1 and σ_3,

$\sigma_1 = \left(\frac{\sigma_x + \sigma_y}{2}\right)$, $\sigma_3 = -\left(\frac{\sigma_x + \sigma_y}{2}\right)$ or, $\sigma_1 = -\sigma_3$

Thus, $\tau_{Max} = \frac{1}{2}(2\sigma_1) = \sigma_1$

Now, substituting in the equation of effective stress,

$$\bar{\sigma} = \frac{1}{\sqrt{2}}\left[\sigma_1^2 + (2\sigma_1)^2 + \sigma_1^2\right]^{\frac{1}{2}} = \frac{1}{\sqrt{2}}\left[6\sigma_1^2\right]^{\frac{1}{2}} = \sqrt{3}\,\sigma_1 = \sqrt{3}\,\tau \quad (2)$$

For the case of strain, substitution in equation (1) yields:

$$\delta_1 = \frac{1}{D}\left[\sigma_1 + \frac{\sigma_1}{2}\right] = \frac{3\sigma_1}{2D}$$

$$\delta_2 = 0$$

$$\delta_3 = \frac{1}{D}\left[\sigma_3 - \frac{\sigma_1}{2}\right] = \frac{3\sigma_3}{2D}$$

$$\bar{\delta} = \sqrt{\frac{2}{3}\left(\delta_1^2 + \delta_2^2 + \delta_3^2\right)} = \sqrt{\frac{2}{3}\left(\frac{9\sigma_1^2}{4D^2}\right)^2} = \sqrt{3}\,\frac{\sigma_1}{D} \quad (3)$$

This gives the relation between effective strain and principal stress.

B. Expression of Effective Strain in Terms of Shearing Strain (8)

The shearing strain γ, is defined such that,

$$\tau = G\gamma = \frac{1}{2}(\sigma_1 - \sigma_3) = \sigma_1$$

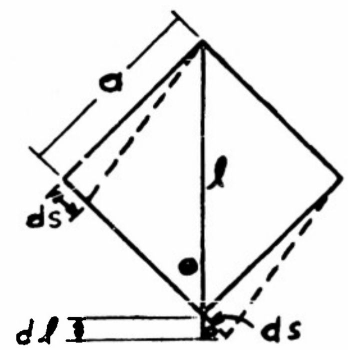
Substituting in eq. (3)

$$\bar{\delta} = \frac{\sqrt{3}\, G\gamma}{D} \qquad (4)$$

A relationship exists between D and the torsional modulus G. This may be shown as follows:

Assume a square of sides a subject to a shearing stress. It is known that, in such a case, the principal stress will act at an angle of 45° to the sides of the square. Therefore, considering the elongation of the diagonal l,

$$d\delta = \frac{dl}{l} = \frac{ds\cos\theta}{\frac{a}{\cos\theta}} = \frac{\frac{ds}{\sqrt{2}}}{\sqrt{2}\,a} = \frac{ds}{2a}$$



$\theta = 45°$

Shearing strain γ may be defined as,

$$d\gamma = \frac{ds}{a}$$

$$\therefore \; ds = \frac{d\gamma}{2}$$

Integrating,

$$\int_0^{\sigma} ds = \int_0^{\gamma} \frac{d\gamma}{2}$$

$$\delta = \frac{\gamma}{2} \qquad \gamma = 2\delta$$

Now, it is known that

$$\tau = G\gamma = 2\delta_1 G = \sigma_1 \tag{5}$$

From the first of equations (1) in section III part C.

$$\delta_1 = \frac{1}{D}\left[\sigma_1 - \frac{1}{2}(\sigma_2 + \sigma_3)\right] = \frac{1}{D}\left[\sigma_1 + \frac{1}{2}\sigma_1\right] = \frac{3\sigma_1}{2D}$$

or,

$$\sigma_1 = \frac{2D\delta_1}{3} \tag{6}$$

Equating equations (5) and (6),

$$2\delta_1 G = \frac{2D\delta_1}{3}$$

$$G = \frac{D}{3}$$

Substituting value of G in terms of D in equation (4) yields,

$$\bar{\delta} = \frac{\sqrt{3}\, D\, \gamma}{3D} = \frac{\gamma}{\sqrt{3}} \qquad (7)$$

C. Comparison of Torsion and Tension Results

It has now been demonstrated that the effective stress in torsion is directly proportional to σ_1 and that the effective strain in torsion is directly proportional to the shear strain γ which in turn is directly proportional to δ_1. Thus, effective stress and effective strain in torsion are directly proportional to σ_1 and δ_1.

In the case of simple tension, since $\sigma_1 = \sigma$, $\sigma_2 = \sigma_3 = 0$, and $\delta_1 = \delta$, $\delta_2 = \delta_3 = -\frac{1}{2}\delta$. $\bar{\sigma}$ is given by $\bar{\sigma} = \sigma$ and $\bar{\delta}$ by $\bar{\delta} = \delta$. Since $\bar{\sigma}$ and $\bar{\delta}$ in tension are also directly proportional to σ and δ, it follows that it is possible to predict what the effective stress, effective strain curve in tension will be from a knowledge of the effective stress, effective strain curve of the specimen in torsion.

IV. EXPRESSIONS FOR $\bar{\delta}$ AND $\bar{\sigma}$ IN TERMS OF TORQUE AND ANGULAR DISPLACEMENT OF SPECIMEN IN TORSION TEST (7)

A. Expression for $\bar{\delta}$ in Terms of θ.

From equation (7) it may be seen that the shearing strain $\gamma = \sqrt{3}\,\bar{\delta}$. γ is taken as the product of the angular twist of the bar per unit length, α, and the distance of the strained increment from the center of the bar, r.

Or, $\gamma = r\alpha$

Thus, $\gamma = \sqrt{3}\,\bar{\delta} = r\alpha$

$$\bar{\delta} = \frac{r\alpha}{\sqrt{3}}$$

Since $\bar{\delta}$ will be calculated for the surface of the torsion bar, r = radius of the torsion bar R, and,

$$\bar{\delta} = \frac{R}{\sqrt{3}}\,\alpha \qquad (8)$$

Since α = angular twist of the bar per unit length, $\alpha = \frac{\theta}{l}$, where θ is the angular displacement of one section of the bar with respect to another section, and l is the length of the bar over which strain is measured. Substituting in eq. (8),

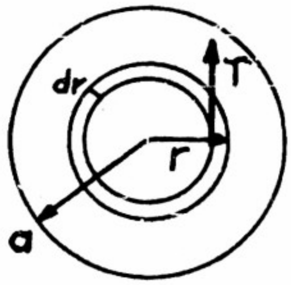
$$\bar{\delta} = \frac{R}{\sqrt{3}}\,\frac{\theta}{l} = K_1\,\theta \qquad (9)$$

B. Expression for $\bar{\sigma}$ in Terms of Torque T.

The torque T for any point located a distance dr from the center of the shaft is given by,

$$T = \tau r$$

The total torque may be found by integrating over the total cross-section of the specimen as shown below:



$$T = \int_0^a \tau r\, 2\pi r\, dr = 2\pi \int_0^a \tau r^2\, dr$$

Since $r = \frac{\gamma}{\alpha} = \frac{\gamma l}{\theta}$

$$T = 2\pi \int \frac{T \gamma^2 l^2}{\theta^2} \cdot \frac{l}{\theta} d\gamma = 2\pi \int_0^a \frac{T l^3}{\theta^3} \gamma^2 d\gamma$$

$$T\theta^3 = 2\pi \int_0^a l^3 T \gamma^2 d\gamma = 2\pi \int_0^a l^3 T \frac{r^2\theta^2}{l} \frac{r\,d\theta}{l} =$$

$$= 2\pi \int_0^a T l r^3 \theta^2 d\theta$$

Differentiating, $$\frac{d}{d\theta}\left(T\theta^3\right) = 2\pi T l r^3 \theta^2$$

Since all measurements are made with respect to the bar surface,

$r = a$.

Then, $$\frac{d}{d\theta}\left(T\theta^3\right) = 2\pi l a^3 T \theta^2$$

Or, $$\frac{1}{\theta^2}\frac{d}{d\theta}\left(T\theta^3\right) = 2\pi l a^3 T$$

Carrying out the differentiation,

$$\theta \frac{dT}{d\theta} + 3T = 2\pi l a^3 T$$

Or,

$$T = \frac{1}{2\pi l a}\left(3T + \theta \frac{dt}{d\theta}\right) = \frac{3}{2\pi l a^3}\left(T + \frac{\theta}{3}\frac{dt}{d\theta}\right) \tag{10}$$

$$T = K_2\left(T + \frac{\theta}{3}\frac{dt}{d\theta}\right)$$

Strain measurements will be made over a two inch gage length in the 1/3 inch section of the torsion bar, in order to rule out inhomogeneous plastic strain effects in the shoulder section of the bar. Thus, $\underline{l}$ will be taken equal to two inches.

APPARATUS

I. THE TORSION APPARATUS

A. General

This device is designed to operate with a minimum of attention of the operator, automatically plotting torque-twist curves during the course of the test, - shutting itself off when the total angular displacement becomes equal to 720°. Any rate of strain from 360° per hour to 180° per minute may be obtained, and held constant within 1%. Torque is measured by expansion of a ring dynamometer coupled to an adjustable lever at right angles to the axis of rotation of the bar. The torque is equal to the product of the force required to stretch the dynamometer and the length of the lever arm. An electrical signal corresponding to the expansion of the dynamometer is fed to the Y-arm of an X-Y recorder. If necessary, a DC pre-amplifier is used to boost the signal to the 10 millivolts required for full scale deflection for the X-Y recorder. A signal, obtained from a voltage dividing potentiometer connected to an angular extensometer, is connected to the X-arm of the recorder.

B. The Torsion Machine

Details of the torsion machine are shown in Figure No. 1.

Grips were machined from SAE 1040 steel. This steel has the following characteristics: (9)

Characteristic	Annealed
Yield Point in Shear	3.62×10^4 lb/in^2
Shear Modulus	1.17×10^7 lb/in^2

The formula for shearing unit stress S_s for a solid shaft (10) is:

$$S_s = \frac{16\,T}{\pi d^3}$$

T = torque

d = diameter

For T = 6,000 lb. in., and d = 1 in.,

$$S = 3.06 \times 10^4 \text{ lb/in}^2$$

The angle of twist of the grip shaft is given by:

$$\theta = \frac{32 T l}{E_s \pi d^4}$$

θ = angle of twist in radius

l = length of grip (in.)

E_s = shearing modulus of elasticity

E_s = minimum: 1.17×10^7 lb/in^2

$$\theta = \frac{32 \times 6 \times 10^3 \times 12}{1.17 \times 10^7 \times 3.14 \times 1}$$

$$= 6.25 \times 10^{-2} \text{ radians} \cong 3.6^\circ$$

The unit shearing stress calculation indicates that the grip shaft will withstand a load of 6,000 in. lb. The angle of twist calculation shows that, if maximum shearing stress is not attained before a 360° angular displacement, the error in measured angular displacement due to twisting of the grips.

will be less than 1%.

The upper grip of the torsion machine may be raised to permit the positioning of a specimen without disturbing the coolant container or its contents. This permits faster specimen changing and conserves coolant.

C. <u>Motor Drive</u>

A 1 H.P. 1725 R.P.M. motor is connected through a 3:1 gear pair to an hydraulic speed reducer. The speed reducer transmits between 0 and the input R.P.M. in either direction. (The 3:1 gear pair is used to limit the input R.P.M. to less than the 750 R.P.M. maximum input for the hydraulic speed reducer.) The speed reducer connects to the drive shaft of the torsion machine through two reducing worm gear drives of 20:1 and 50:1 respectively.

Power requirements for the torsion apparatus may be calculated as follows:

$$P = T\omega = T \cdot 2\pi n$$

P = power (ft lb/sec)

At maximum strain rate, $\underline{n}$ = 1/120 rev/sec

T = torque (lb-ft)

At maximum torque, T = 500 lb-ft

W = angular velocity (rad/sec)

P = (500) (2)/120 = 26.2 ft. lb/sec

n = revolutions/sec

Assuming a 50% loss in power due to friction in the 50:1 worm drive assembly,

$$P_1 = 52.4 \text{ ft. lb/sec}$$

Assuming a 50% loss in power due to friction in the 20:1 worm drive assembly,

$$P_2 = 104.8 \text{ ft. lb/sec}$$

Assuming another 50% loss in power due to inefficiency in the hydraulic speed reducer,

$$P_t = 209.6 \text{ ft. lb/sec}$$

A 1/2 H.P. motor delivers 275 ft. lb/sec. Therefore, a 1/2 H.P. motor is capable of supplying the power required. However, high starting torque is required for the speed reducer. Hence, a one H.P. motor was selected for use.

Torque requirements on various drive shafts may be determined as follows:

The expression for the maximum torsion shaft torque may be given by, $T_1 = \frac{P_1}{2\pi n_1} = 6000$ lb. in.

The expression for the torque applied to the shaft turning the worm which meshes with the torsion machine drive gear is

$$T_2 = \frac{P_2}{2\pi n_2}$$

But $n_2 = 50\, n_1$

and $P_2 = 2P_1$ due to friction loss.

Then

$$T_2 = \frac{2P_1}{50 \cdot 2\pi n_1} = \frac{1}{25} T_1 = 120 \text{ lb.}$$

Therefore, 3/4 inch shafting will be sufficient for this worm drive.

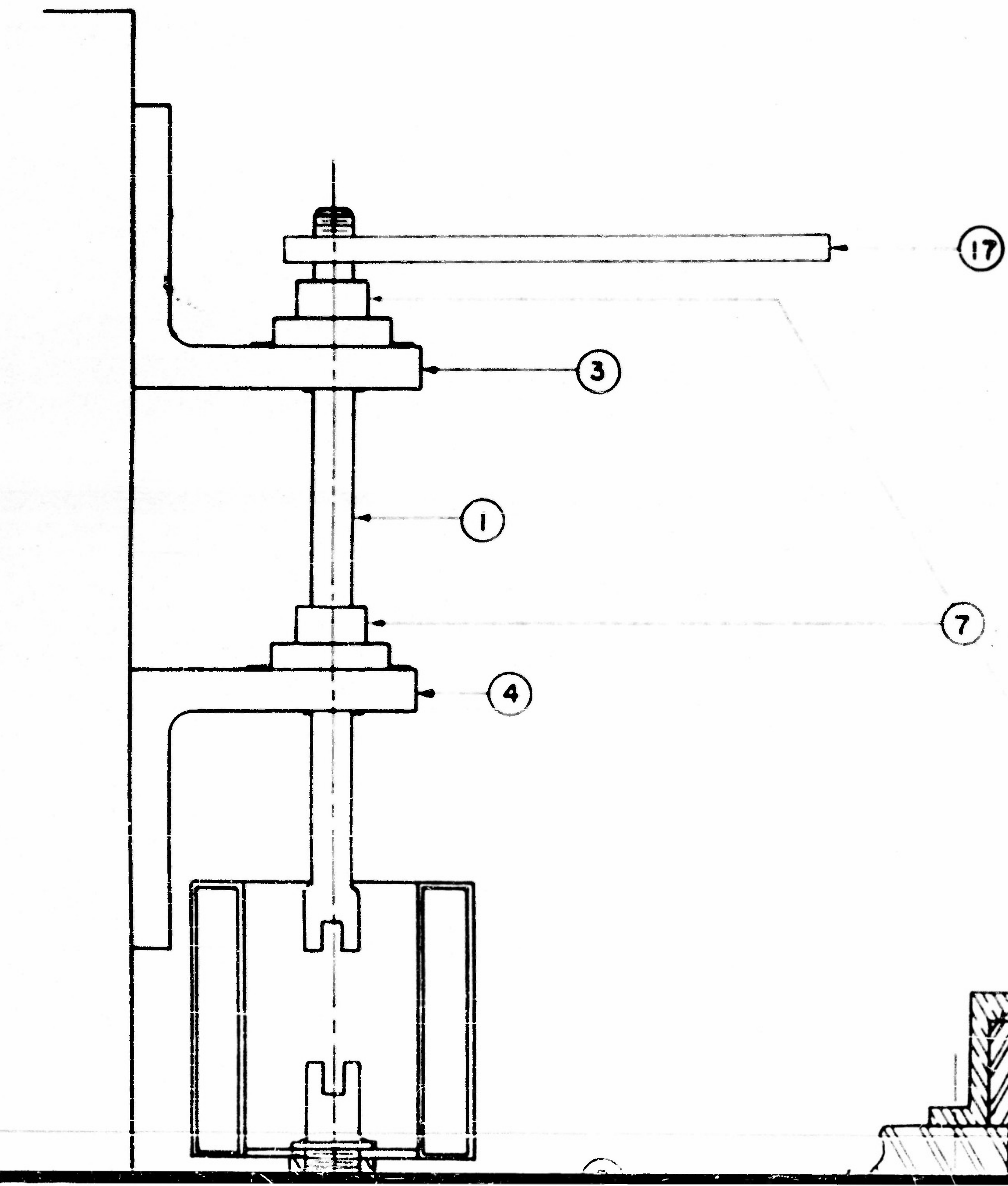
A similar calculation for the applied torque on the worm driving the 20 tooth gear shows the maximum torque that this worm must exert to be 12 lb. in. Since this torque will not be exceeded at any point between the speed reducer and the motor shaft, torsional stress and strain present no problem at any point in the system.

C. The Ring Dynamometer (11)

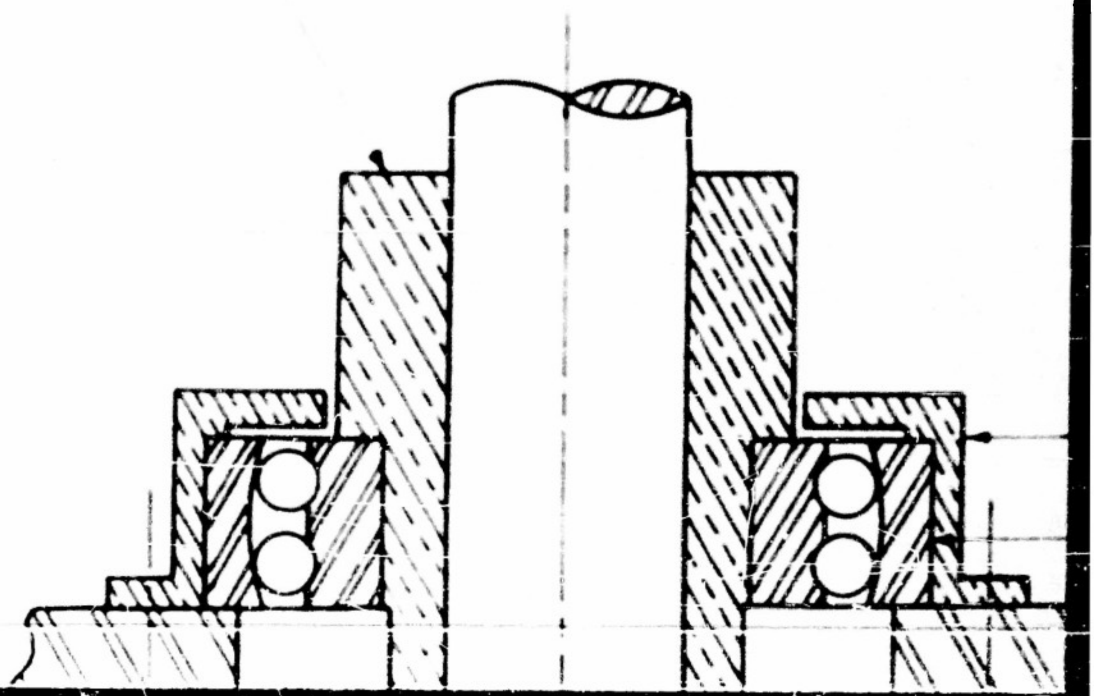
The dynamometer is constructed from medium carbon steel fully killed, and annealed to a dead soft condition after machining to eliminate internal stresses and to minimize hysteresis. As may be observed from Fig. No. 2, a tensile load applied to the dynamometer will produce certain stresses at the inner and outer fibers of the curved portions of the dynamometer. As long as these stresses are kept within the elastic limit of the material, the strains produced will be proportional to the applied tensile load. This ring is claimed sensitive to less than one pound in a range between 0 and 1000 lb. The designers of the ring claim an average error of only 0.36% for the entire calibration range, and a maximum error of 0.95%.

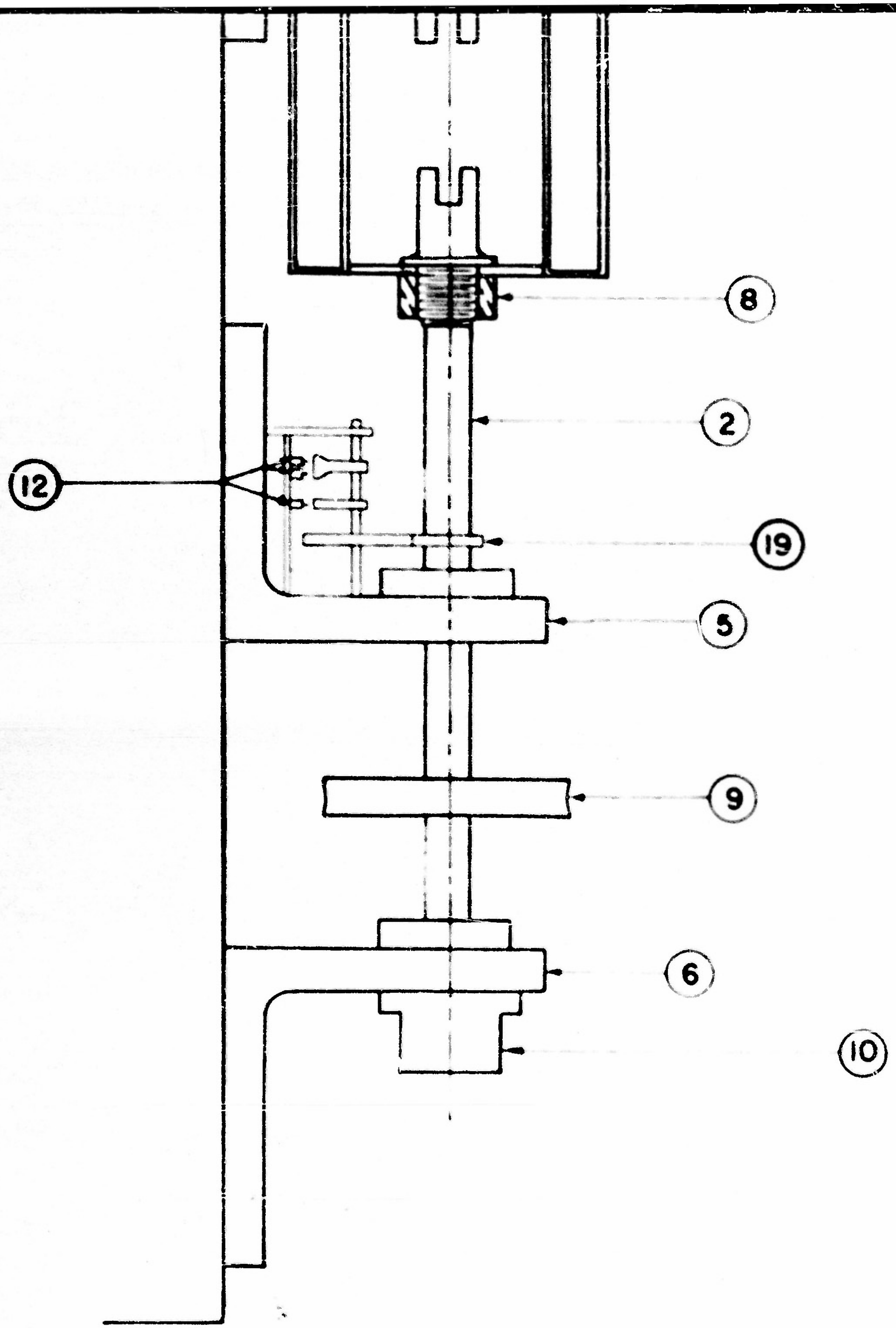
Four A-1 SR-4 strain gages are attached to the ring as shown in Figure No. 2. The inner two gages are connected in series and the outer two gages are connected in series. When the ring is extended, the inner gages elongate; the outer gages contract. The elongation is numerically equal to the contraction. Use of two gages in series, while not increasing the ratio of $\Delta R/R$ does give a greater total change in resistance, ($2\Delta R$). For a given current, twice the voltage change will thus be observed, if two gages are series connected.

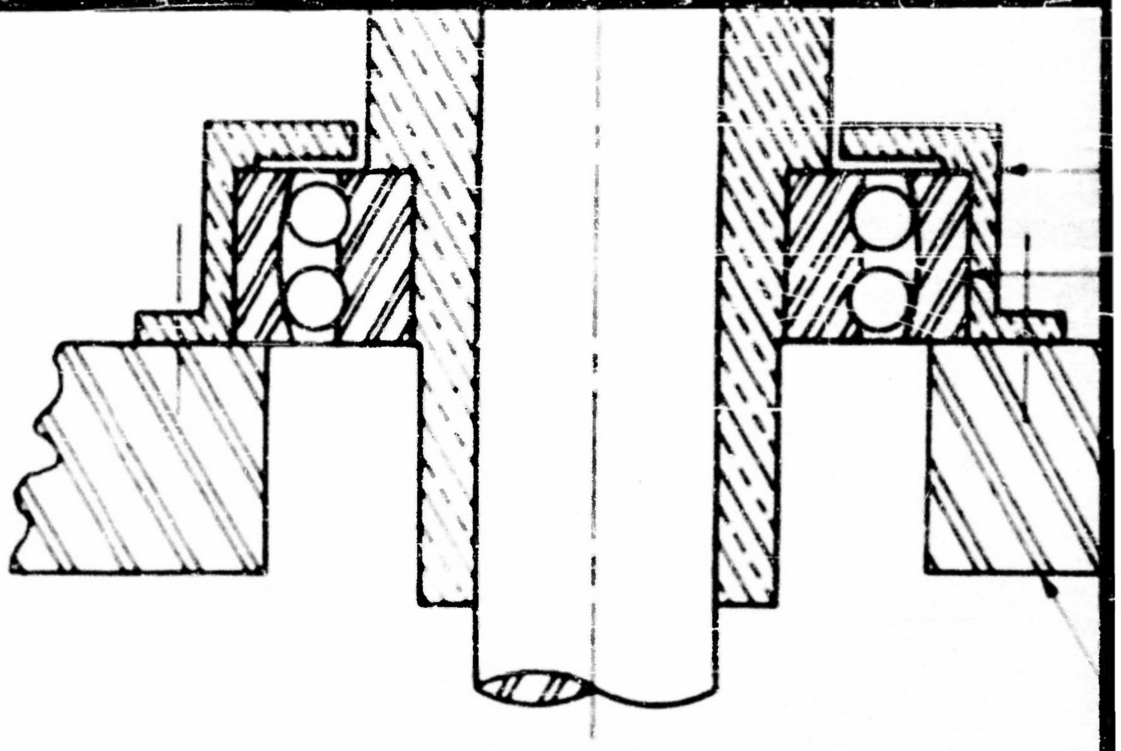
The ring will, of course, have two mountings. The wall mounting takes the form of a threaded stud which may be placed in any one of twelve holes spaced one inch apart in a steel plate welded to an "I" beam in the building. The mounting



NO.	QUAN	ITE
1	1	UPPER SHAFT
2	1	LOWER SHAFT
3	1	TOP BEARING
4	1	SECOND "
5	1	THIRD "
6	1	BOTTOM "
7	2	SLIDE BUSHIN
8	1	COOLING CAN
9	1	$6\frac{1}{4}$" O.D. 60 TO
10	1	THRUST BEAR
11	4	BEARING RETA





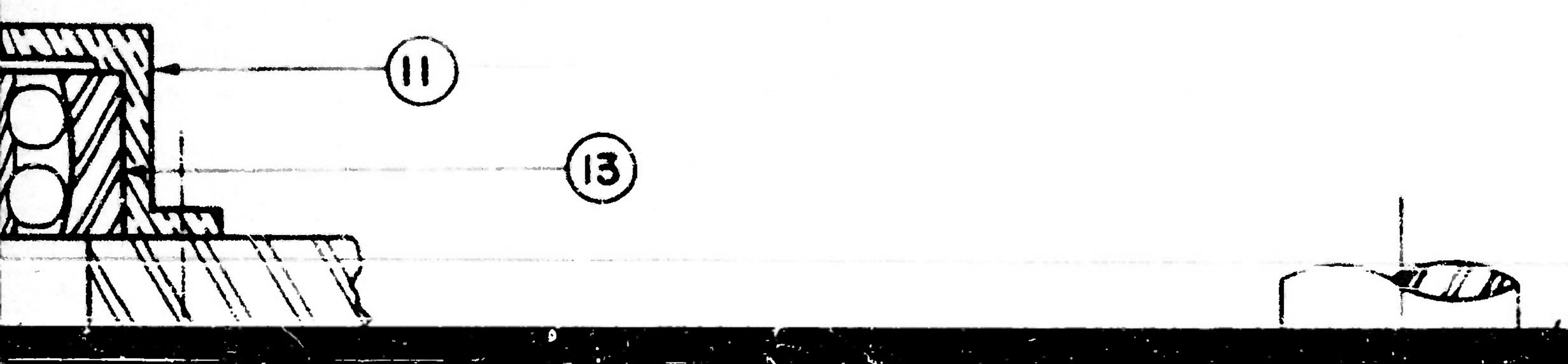


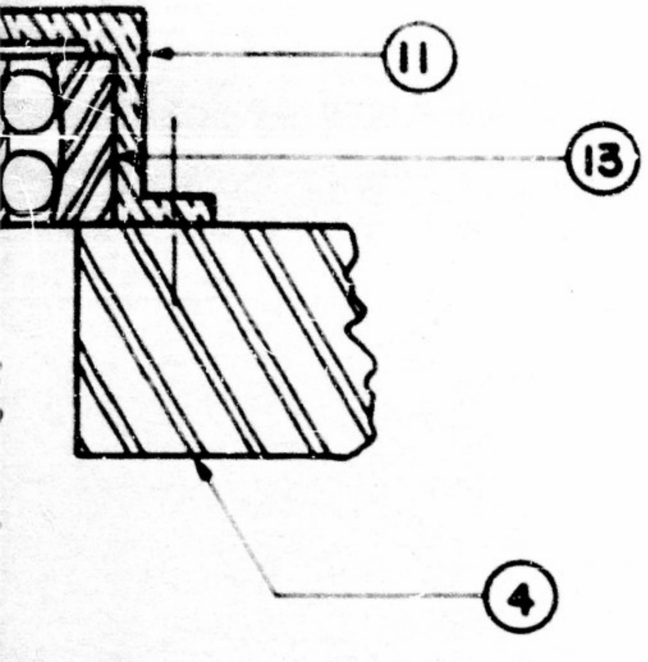
UPPER BEARING ASSEMBLY

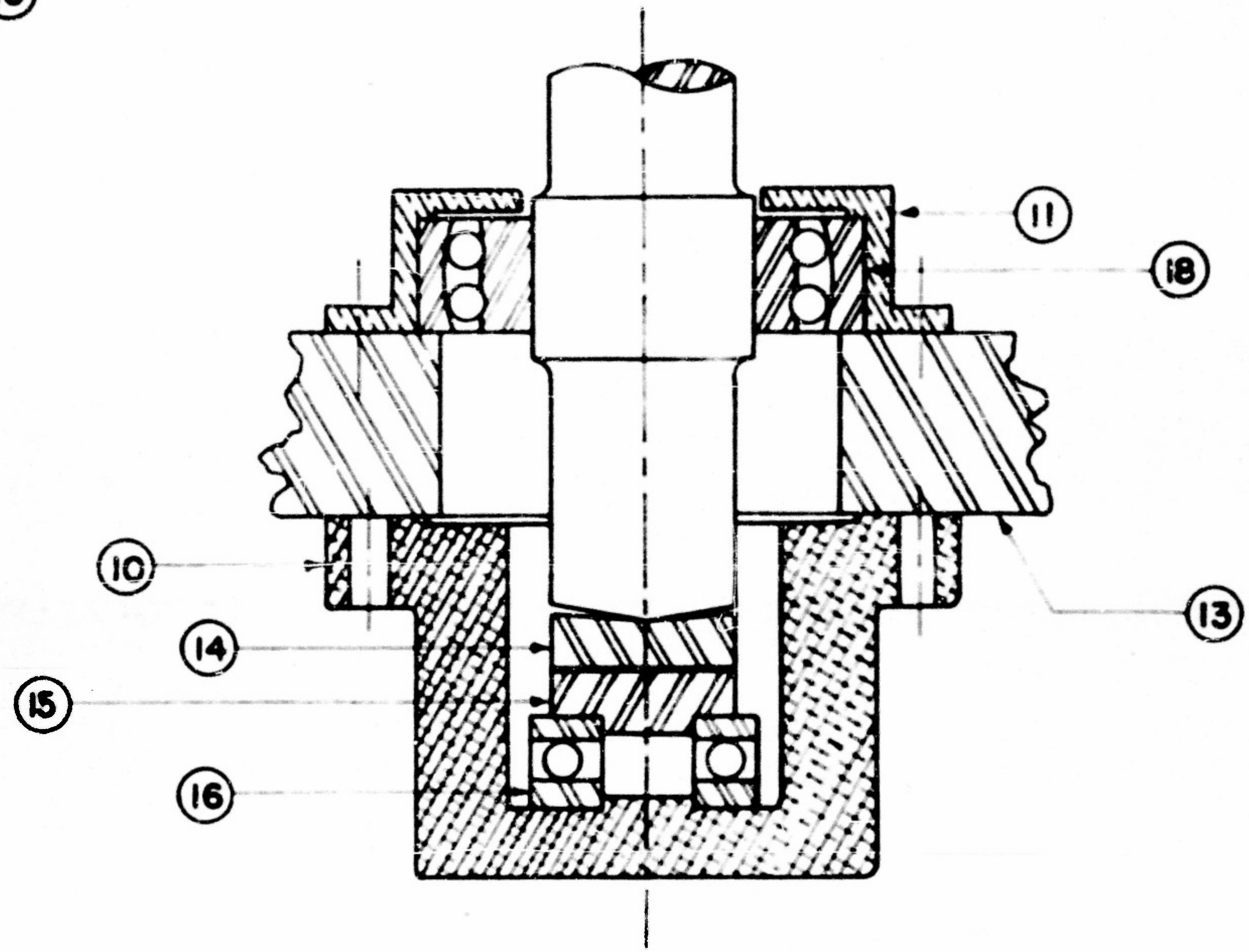
SCALE $\frac{1''}{4} = 1''$

ITEM	MATERIAL	NO.	QUAN	ITEM	MATERIAL
PER SHAFT	STEEL	12	3	MICROSWITCH	
VER SHAFT	"	13	3	L206 FAFNIR BEARING	STOCK
BEARING HOUSING	"	14	1	SLIDE BUSHING	STEEL
OND " "	"	15	1	THRUST BEARING BUSHING	"
RD " "	"	18	1	L208 FAFNER BEARING	STOCK
TOM " "	"	16	1	E-1 AETNA BEARING	"
DE BUSHING	BRONZE	17	1	LEVER ARM TO CONNECT	
OLING CAN LOCK NUT	"			TORSION MACHINE TO RING	
O.D. 60 TOOTH GEAR	BRONZE			DYNAMOMETER	STEEL
RUST BEARING HOUSING	AL	19	1	2:1 REDUCTION GEARS	
ARING RETAINER	BRASS				

3





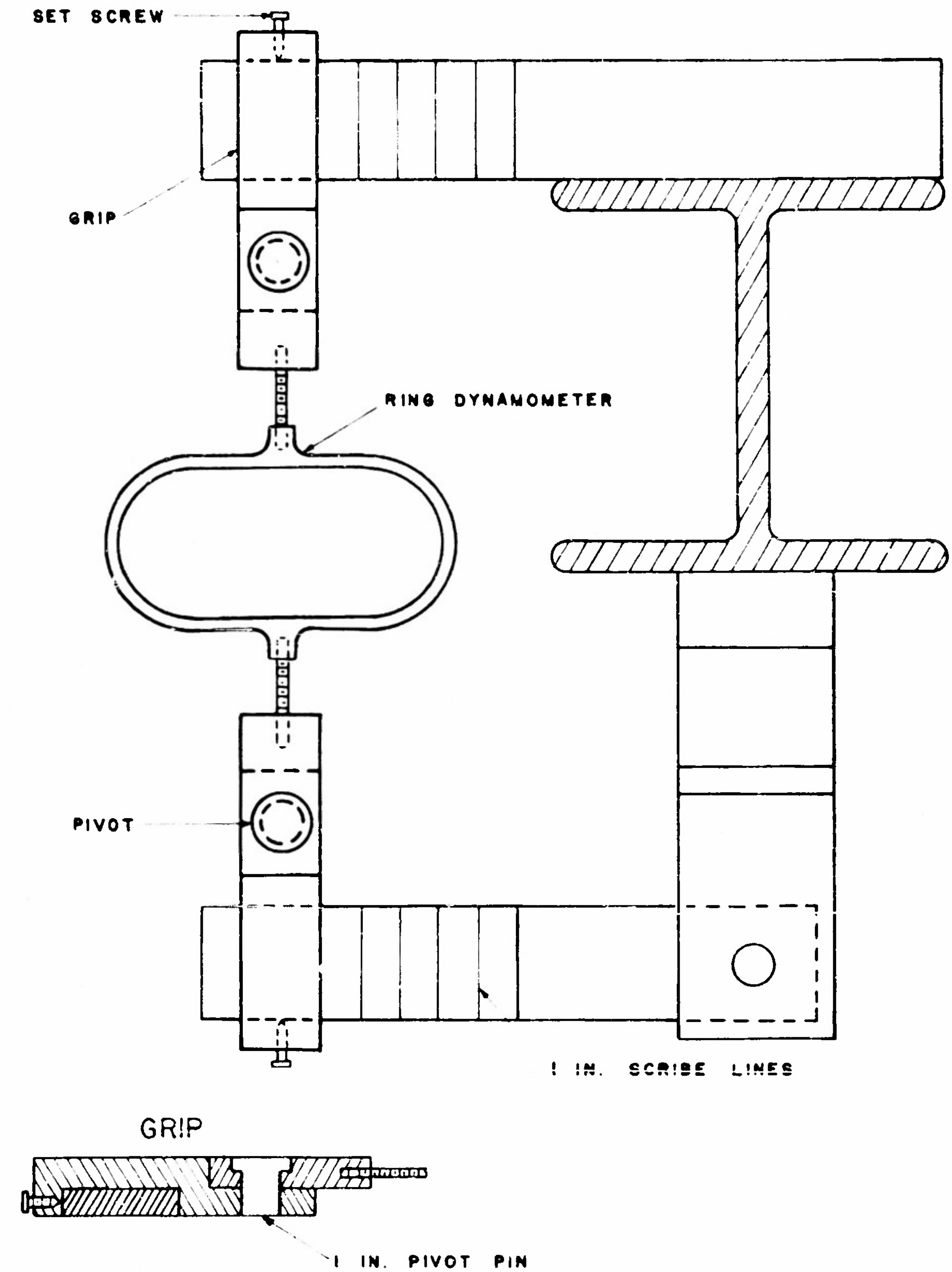


LOWER BEARING ASSEMBLY

DETAIL SCALE 1"=1"

DESIGNED BY-	
DRAFTED BY-	
C & D-	
DATE-	

RING DYNAMOMETER MOUNTING



SCALE $\frac{1}{4}$ IN. = 1 IN.

DESIGNED BY—	
DRAFTED BY—	Thayer
OKD	
DATE	

FIGURE NO.

attached to the lever arm of the torsion machine has a flexible joint to allow for a slight angular displacement of the lever. This displacement comes about due to an elongation of the ring as the load is applied. Thus, the ring stud will not make a right angle with the lever when a specimen is placed in torsion. Since the torque formula $\vec{T} = \vec{r} \times \vec{F} = |rF|$ used in determination of the ordinate values assumes a right angle at this point, some error will ensue. This error may be calculated as follows:

Let the lever arm length be given by r, the distance of the lever to the wall (ring dynamometer width plus pivots) by d. r and d are at right angles to each other before a load is applied. As a load is applied, d elongates to $d + k$ and r rotates about a point A. These conditions are shown below in Figure No. 4.

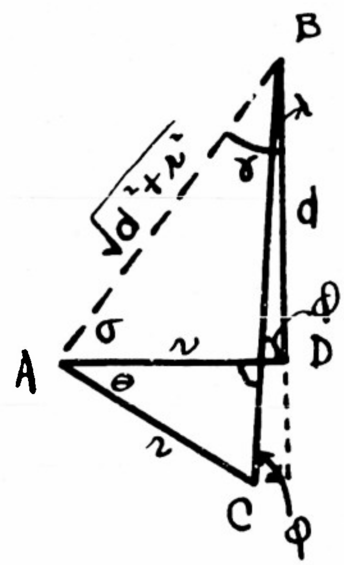


Figure No. 4

After rotation, the torque is given by $Fr \sin \phi$. The dependance of ϕ on the original length and elongation may next be investigated.

The three sides of triangle ABC are known. The sine of angle BAC may then be calculated by the formula

$$\sin A = \frac{2}{r\sqrt{d^2+r^2}}\left[s\left(s-[d+k]\right)\left(s-r\right)\left(s-\sqrt{d^2+r^2}\right)\right]^{1/2}$$

where $$s = \frac{r + d + k + \sqrt{d^2+r^2}}{2}$$

Sine of angle ABC is determined from the law of sines.

$$\sin B = \frac{r}{d+k} \sin A$$

From Triangle ABD,

$$\sin \gamma = \frac{r}{\sqrt{d^2+r^2}}$$

Now,

$$\gamma - B = \lambda$$

$$\phi = 90^\circ - \lambda = 90^\circ - (\gamma - B)$$

Thus, ϕ has been determined as a function of r, d, and k.

From a consideration of triangle BAD,

$$\sin \sigma = \frac{d}{\sqrt{d^2+r^2}}$$

For the experimental set-up used,

r_{max} = 12 in.

d = 20.5 in.

k = 0.25 in. (maximum elongation of ring dynamometer)

Computation shows that $\phi = 90^{\circ}$ with -0.01%. Consequently, the expression $T = F\underline{r}$ may be used with insignificant error.

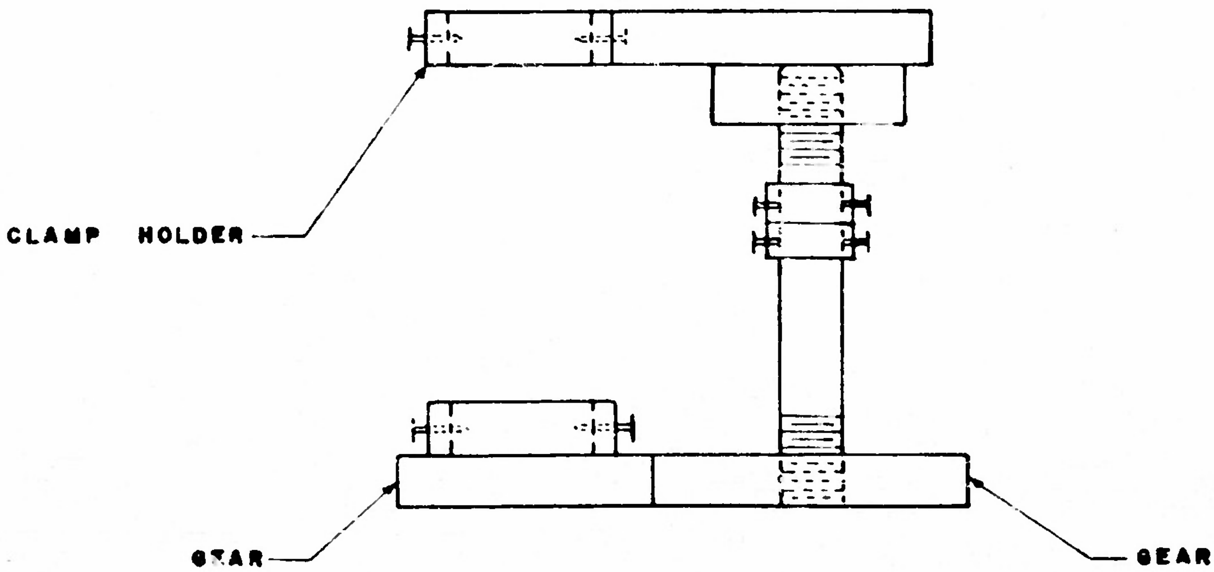
D. Strain Measurement

Strain measurement is obtained by means of a potentiometer, capable of turning through an angle of 360°, linear within $\pm$ 0.5%. Since the strain is measured through a 720° angle, and is measured over a two inch gage length in a section of smaller diameter than either end of the test bar, a special grip is needed. Constructional details are as follows:

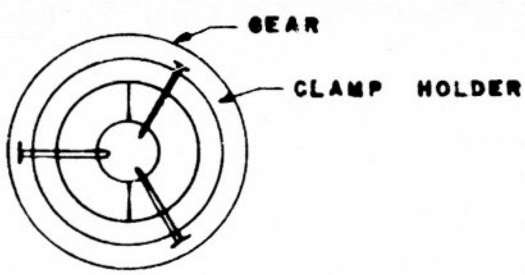
A 1-1/4 inch diameter gear is attached to the 1/3 inch diameter section of the test bar. To accomplish this, a 3/4 inch hole is drilled in the gear. A housing accommodating a two piece removable clamp is then brazed onto the gear. This gear meshes with a gear having twice the number of teeth of the first gear (approximate size, 1-3/4 in.). This second gear is attached by means of a short vertical shaft to the potentiometer. The potentiometer housing is attached to a horisontal shaft which, in turn, is attached to the test bar by means of another two piece removable clamp. The clamps are spaced two inches apart on the test bar. The two gears should be approximately 1/4 inch thick to permit fast lineup and positive meshing.

This extensometer has the following advantages: (1) It is sufficiently compact to allow total immersion of the extensometer in any coolant used without alteration of the coolant container, and (2) it may be quickly and easily fastened to the test bar. In practice, two extensometers may be made. While

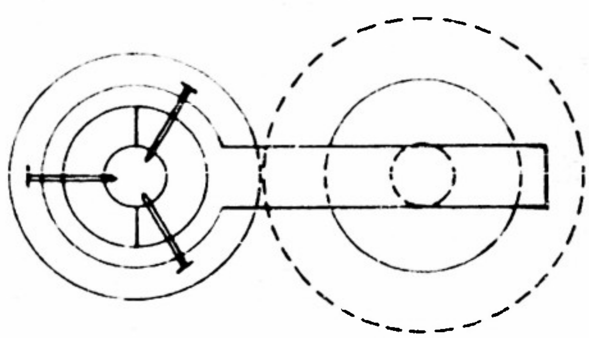
ANGULAR EXTENSOMETER



SIDE VIEW



TOP VIEW
OF LOWER GRIP



TOP VIEW

DESIGNED	BY—
DRAFTED	BY—
OKD	
DATE	

one extensometer measures the strain of a test bar undergoing torsion, another extensometer may be fastened to the next test bar to be used. Upon fracture of the first test bar, the second bar and its extensometer may be slipped into the grips. Connecting the three wire extensometer plug completes the installation. Thus, it will be possible to replace specimens and be ready for a second test in two minutes or less. Since it is possible to replace a specimen without disturbing the coolant container, by lifting the upper grip of the torsion machine, it will be possible to effect a change of specimens with little loss of coolant.

A simple jig and inking device may be prepared which will place two marks on the specimen two inches apart. The marks will be the same on all test specimens. The extensometer clamps will be fastened on these marks, assuring that strain measurements will be made on the same two inch gage length on all specimens.

A schematic diagram of the electrical circuit for recording angular displacement is shown, along with design data.

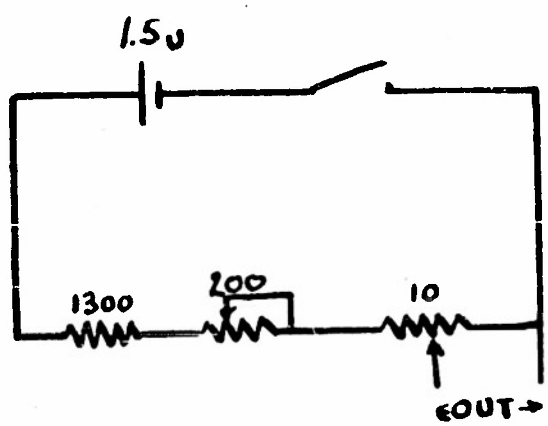


Figure No. 6

Since 0.01 volt will drive the X-arm of the X-Y recorder full scale,

$$iZ_{max} = 0.01 \text{ v.}$$

Assume $i = 10^{-3}$ amp. The resistance Z of the slide wire is then given by,

$$Z = \frac{10^{-2}}{10^{-3}} = 10 \text{ ohms}$$

$$V_r = 1.5 - 0.01 = 1.49 \text{ v.}$$

$$R = \frac{1.49}{10^{-3}} = 1500 \text{ ohms}$$

Resistor R is broken into a fixed and a movable part to prevent an accidental serious overloading of the X-Y recorder.

E. Automatic Shut-Off Control

Since testing conditions require that a two inch gage length of the 1/3 inch section of all test bars be rotated through an angle of 720° and then stopped, an automatic shut-off control on the torsion motor drive is desirable when this angle of 720° is reached. To accomplish this, a two-to-one gear assembly turned by the motor driven grip is mounted on the torsion tester. When this grip has turned through an angle of 720°, two arms mounted to the shaft of the larger gear actuate two micro-switches. Details are shown in Fig. No. 1. One of these switches opens the 115 v. circuit supplying current to the torsion motor, X-Y recorder, solonoidal gas valve, and thyratron circuits (Part F.) The other switch is used to open any auxiliary battery circuits employed.

F. Temperature Control (12)

Since temperature as well as strain rate must be held constant for any one specimen test, some form of temperature control is necessary. The temperatures used will be -196° C. (liquid nitrogen), -150° C. (freon 12 cooled with liquid nitrogen), -100° C. (alcohol and ether combination), -78° C. (alcohol and ether combination cooled with dry ice), and 20-30° C. (room temperature).

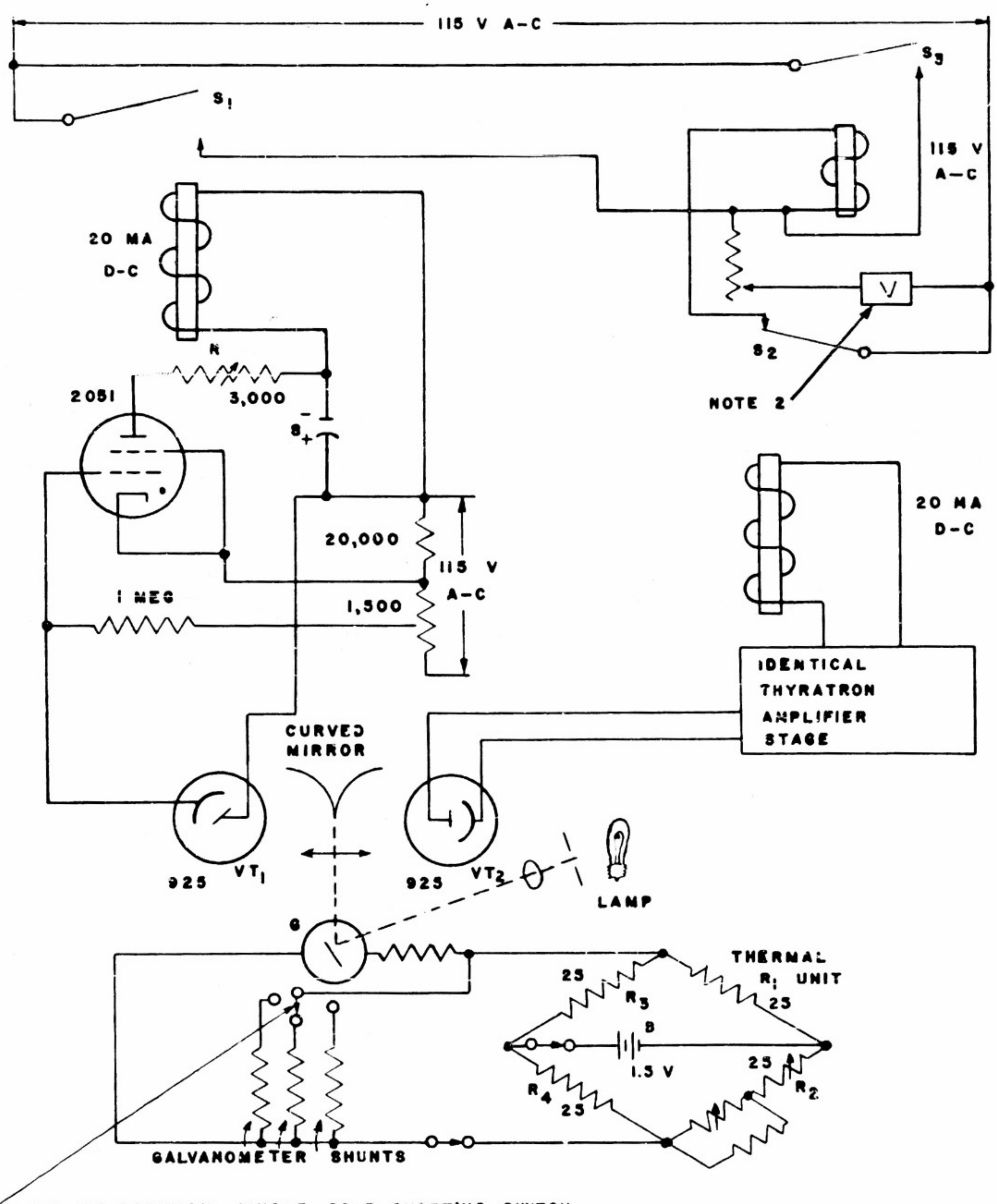
In keeping with the rest of the apparatus, the temperature control is automatic, the control mechanism being actuated when the temperature changes $\pm$ 0.25° C. The sensing element is a 25 ohm platinum resistance thermometer.

In this circuit, R_2 is set at the value R_1 the resistance thermometer will have at the particular temperature desired. Then, relay S_3 is manually closed. This sends a current to magnetic solenoid V, actuating the solenoid. When the temperature starts to go below the setting, the galvanometer swings in a clockwise direction, activating phototube VT_1, causing S_2 to open. This removes the current from solenoid V. When the temperature increases above the predetermined setting, phototube VT_1 is actuated, causing S_1 to close and S_2 to close. Solenoid V is again actuated.

The arrangement by which liquid nitrogen is introduced into the cooling container, pumping method, reservoir and reservoir control are shown in Figure No. 8.

A reservoir has been placed between the liquid nitrogen supply and the coolant container to minimize time lag between temperature control signal and delivery of the liquid. The

TEMPERATURE REGULATOR CIRCUIT



NOTE 1: 3 POSITION SINGLE POLE SHORTING SWITCH

NOTE 2: SOL...OIDAL GAS VALVE

DESIGNED BY—	[illegible]
DRAFTED BY—	Thayer
OKD	
DATE	

FIG NO.

liquid level in the reservoir is controlled by means of an aluminum ball float shown in Figure No. 8a. The associated electrical circuit is shown in Figure No. 7. When the ball makes contact with the lower stud, a circuit is closed, actuating a locking relay. This, in turn, closes exhaust valve V and passes current to the nichrome heater coil immersed in the liquid nitrogen supply container. The heater vaporizes a small amount of the liquid, building up a pressure in the supply container sufficient to pump the liquid into the reservoir. When the sphere rises high enough to contact the upper stud, another relay coil breaks the circuit to the nichrome heater and to valve V, allowing any built up pressure to fall to atmospheric pressure. A locking relay is used as the control to prevent a starting of the liquid nitrogen pump due to an accidental interrupting of relay current.

A snug-fitting cap covers the supply container opening. This cap is equipped with a small thin rubber diaphragm, which will blow out if the pressure in the container goes over $20\ lb/in^2$.

Freon 12 presents a somewhat different problem. In this case, the gas must be liquified by cooling with dry ice. If further cooling is desired, liquid nitrogen is admitted into a jacket surrounding the liquified freon which in turn surrounds the specimen. If cooling below the dry ice temperature is required, valve V controls the amount of liquid nitrogen admitted to the jacket. If the dry ice temperature is required, valve V

LIQUID NITROGEN SUPPLY SYSTEM

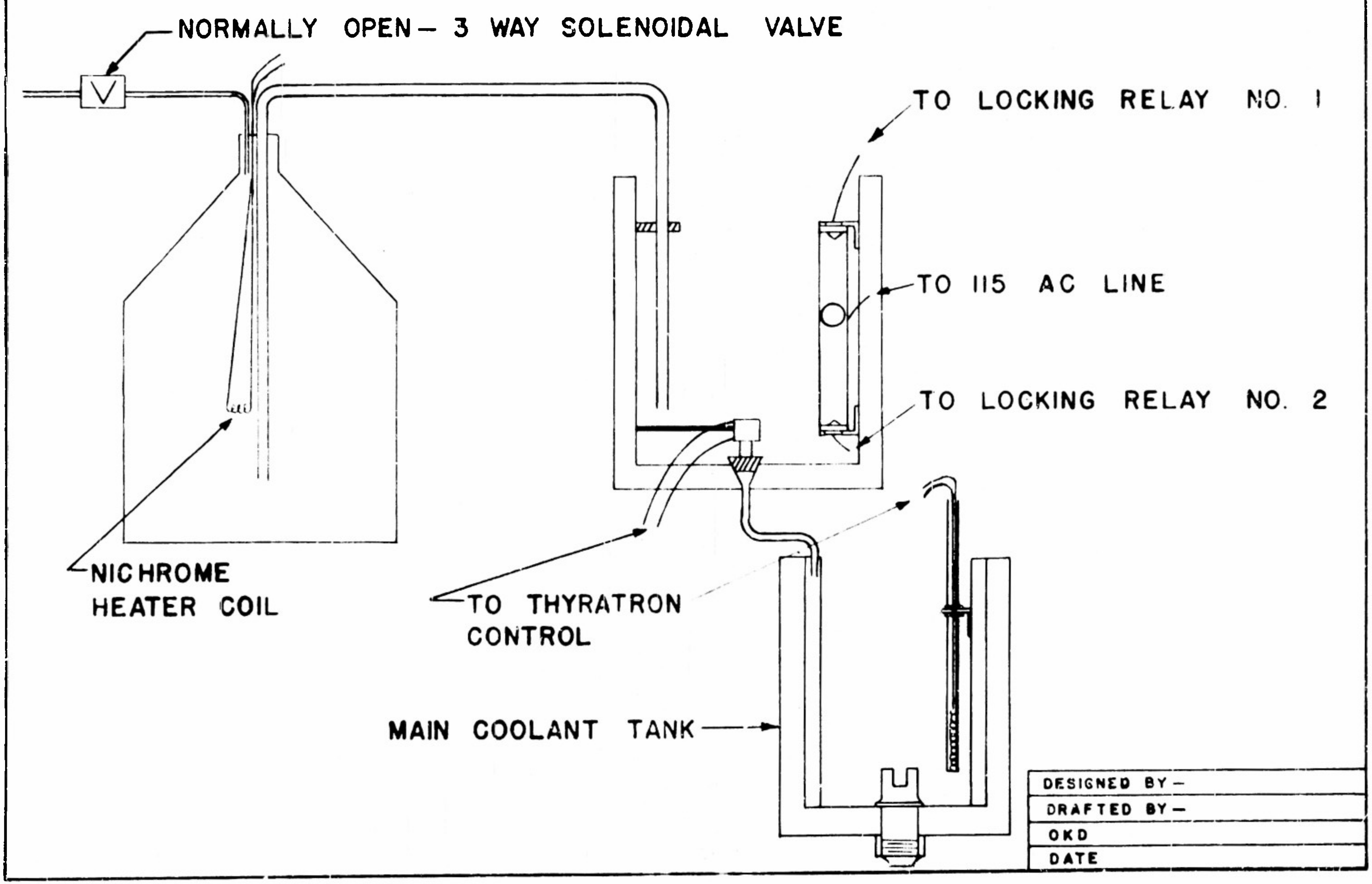
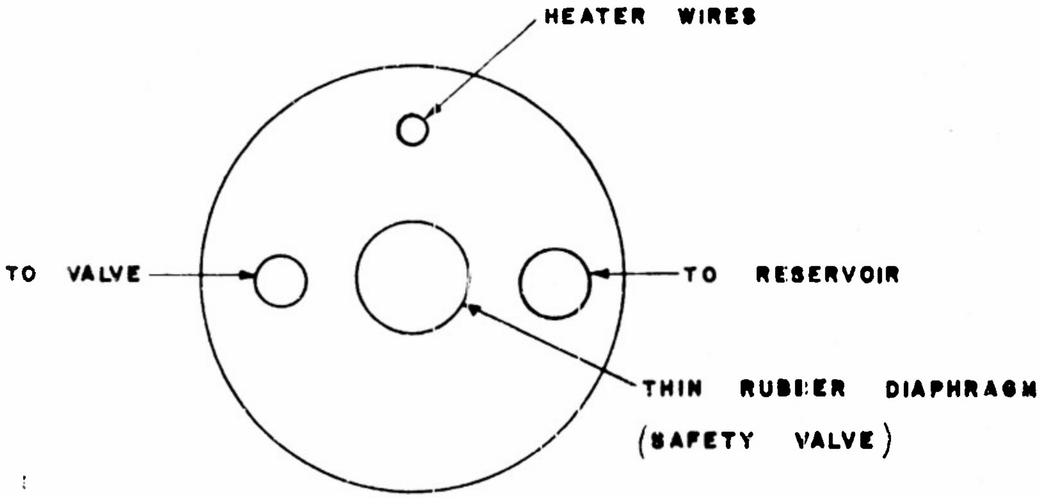
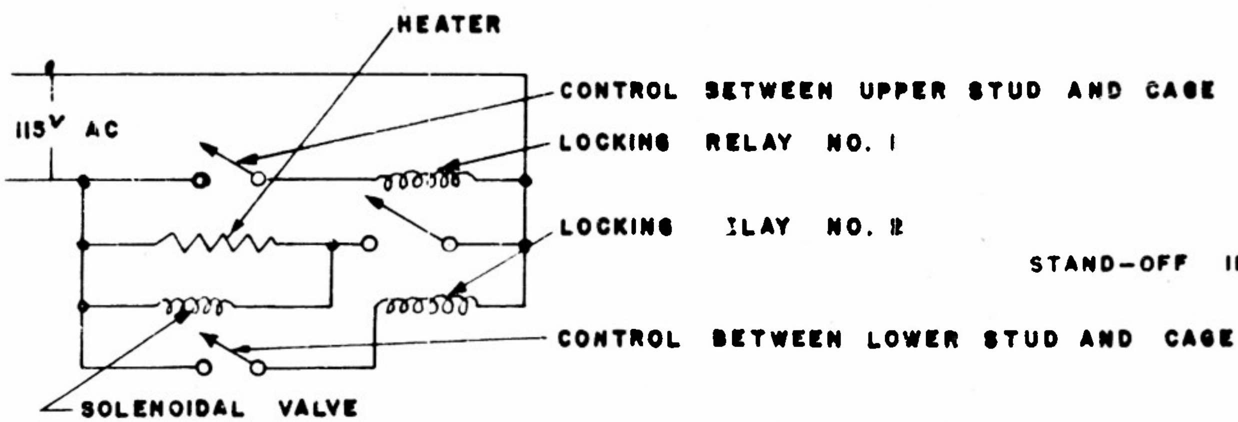


FIGURE NO

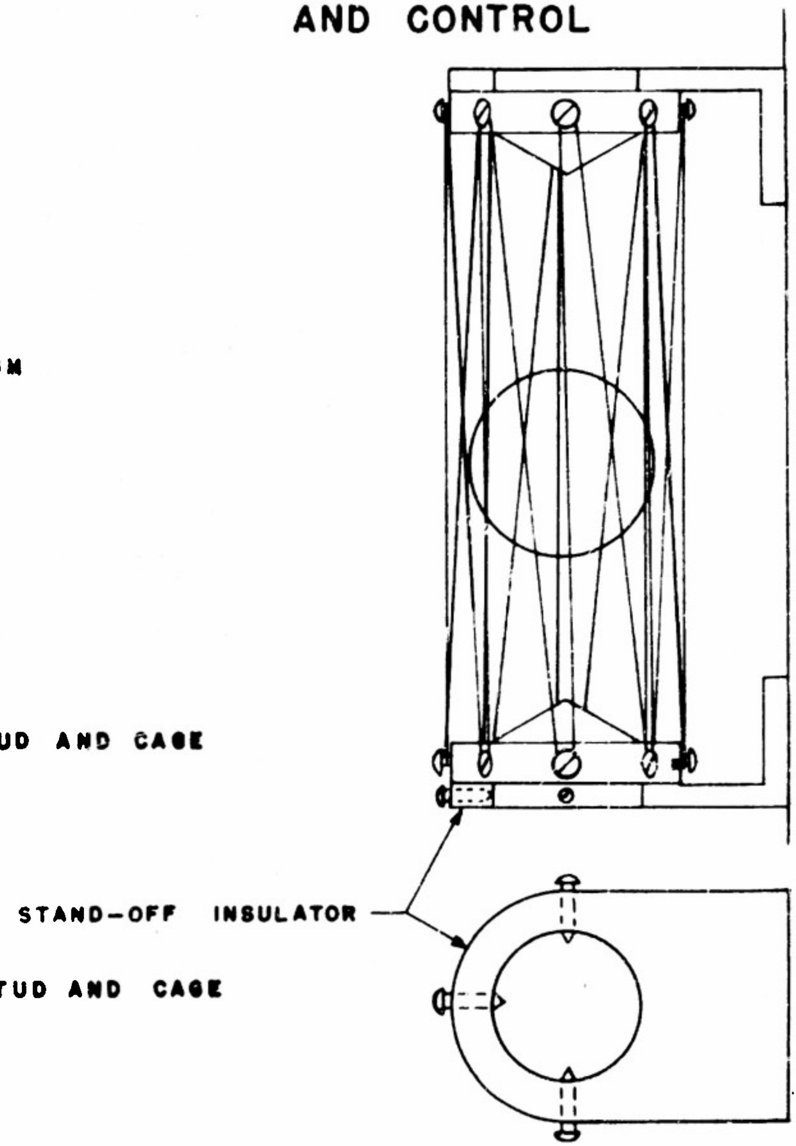


DEWAR FLASK CAP



RESERVOIR FLOAT CONTROL CIRCUIT

RESERVOIR FLOAT GAUGE
AND CONTROL



DESIGNED BY—	
DRAFTED BY—	Thayer
OKD	
DATE	

FIGURE NO.

controls the amount of gas coming from the freon tank. An additional liquid valve may be required for fine control. However, this is to be avoided if possible, due to possible difficulties with the valve at low temperature operation.

Both torsion grips act as heat sources; the coolant acts as a sink. To prevent rapid loss of coolant, two auxiliary containers, 6 inches in diameter, containing dry ice, are attached to the grips. To further reduce coolant losses, the test specimen and angular extensometer may be pre-cooled with dry ice to -78° C. before being placed into operation. This cooling may be accomplished while another specimen is being tested, so that no loss in time will result.

G. Determination of Rate of Twist of Specimen

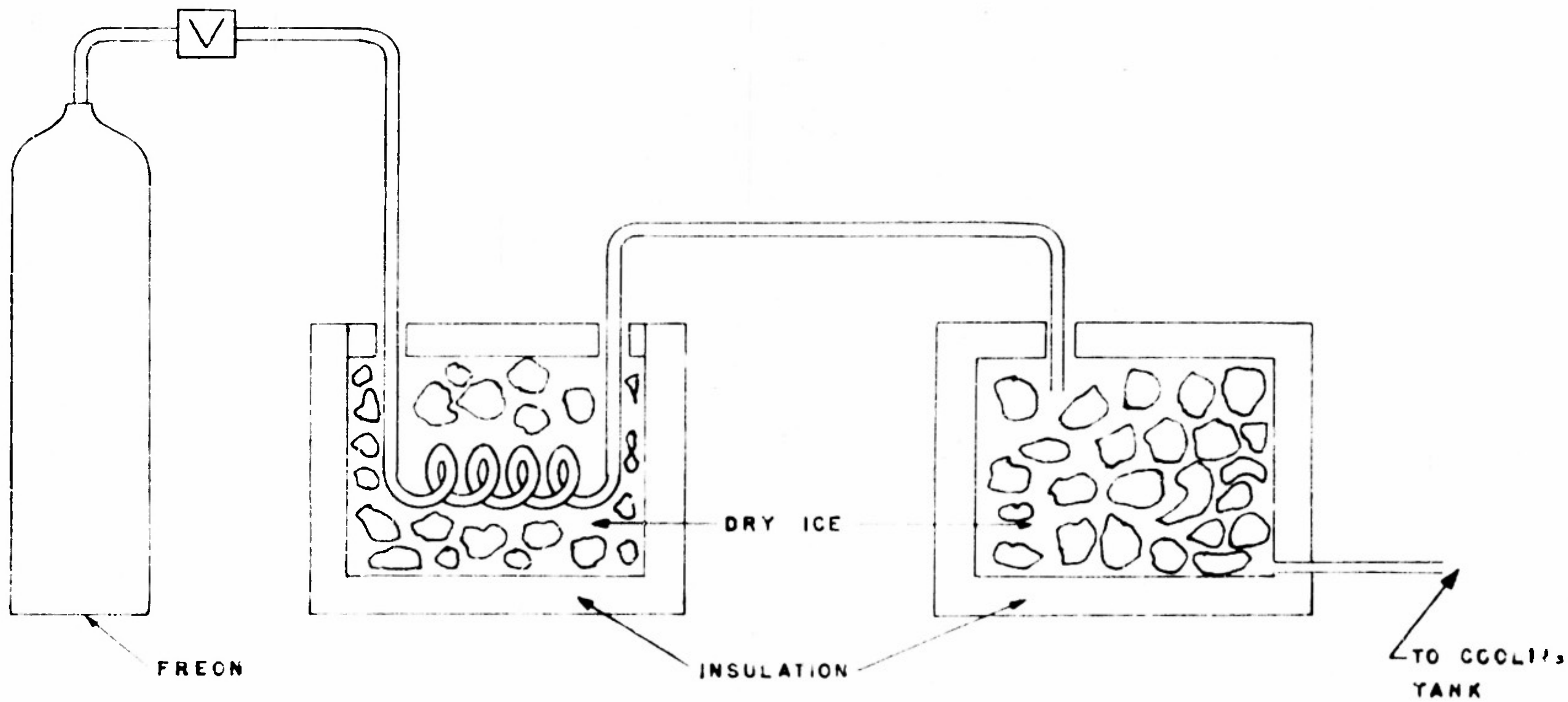
The rate of twist of the specimen will be determined by means of a magneto type tachometer mounted between the speed reducer and the transmission. Transmission ratios will be accurately determined, and a calibration chart established, translating desired strain rates at which the torsion grip will rotate into tachometer readings. This arrangement permits the use of a standard tachometer without a special gear train.

H. The X-Y Recorder

The X-Y recorder used is made by Leeds & Northrup, and is a modified Series 6000 Speedomax. Its specifications are as follows:

Record:	Continuous line
Ranges:	X-Axis (pen travel) 0-10 m.v.
	Y-Axis (chart travel) 0-10 m.v.

LIQUID FREON SUPPLY SYSTEM



DESIGNED BY–	
DRAFTED BY–	
OKD	
DATE	

FIGURE NO.

Response Speed:	X-Axis – 1 second for full scale balance (9 7/8")
	Y-Axis – 4 seconds for full scale balance (10")
Chart:	100 uniform divisions. 10" wide and 10 divisions per inch in both horizontal and vertical directions
Input Impedance:	From 1 ohm to 2000 ohms without affecting accuracy, sensitivity, and speed of response.

The strain recording circuit has been designed for use with any high impedance servo amplifiers having sufficient sensitivity to record full scale deflection for an input signal of 10 m.v. D.C.

REFERENCES

1. Hollomon, J. H.: "Tensile Deformation", Trans. AIME. 162 (1945) pp. 268 to 289.
2. Ludwik, P.: "Elemente der Technologischen Mechanik", Berlin (1909). Julius Springer.
3. "Correlation of Laboratory Tests with Full Scale Ship Plate Fracture Tests", Ship Structure Committee Report Serial No. SSC-19, June 1950.
4. Klier, E. P.: "The Tensile Properties of Selected Steels as a Function of Temperature", submitted to A.S.M.
5. Gensamer, M.: "Strength and Ductility", 20th Campbell Lecture, Trans. A.S.M. 36 (1946) pp. 30 to 60.
6. Gensamer, M.: "Strength of Metals under Combined Stresses", American Society for Metals (1941) Cleveland, Ohio.
7. Nadai, A.: "Theory of Flow and Fracture of Solids", McGraw-Hill (1950) New York.
8. Sachs, G.: "Spanlose Formung der Metalle", Edwards Brothers (1937) Ann Arbor, Mich.
9. Lyse, I. and H. Godfrey: "Shearing Properties and Poisson's Ratio of Structural and Alloy Steels", Proc. A.S.T.M. (1933) 2, p. 274.
10. Seely, F.: "Advanced Mechanics of Materials", J. Wiley & Sons, (1941) New York.
11. Low, Jr. and F. Garofalo: "Precision Determination of Stress-Strain Curves in the Plastic Range". Penn. State College (1945)
12. Roof, J.: "A Thermostatic Control System", Electronics (Oct. 1943)