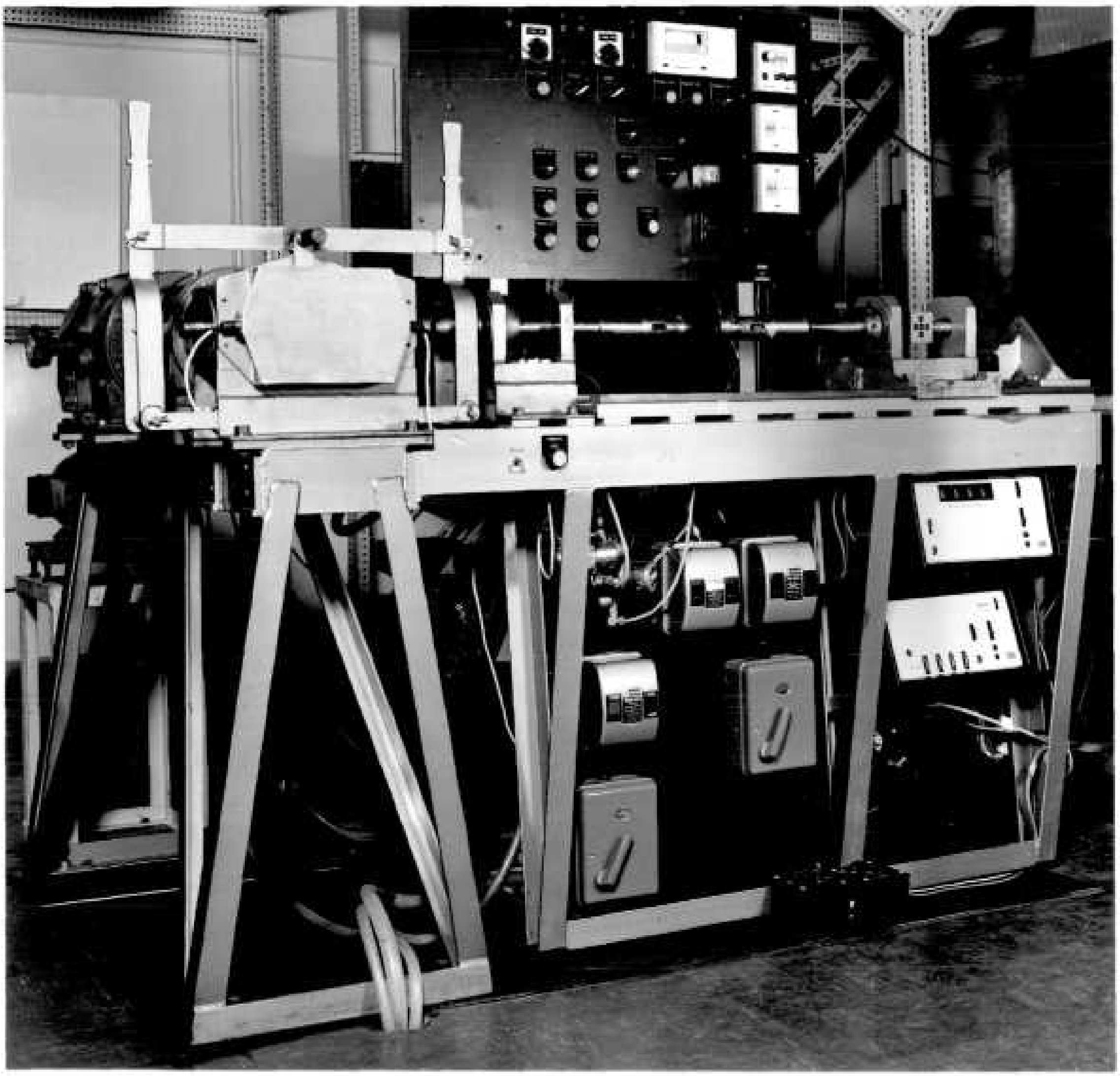

General view of the finished machine
with its bed cleared in preparation for the fitting of a
special heating system.

THE DESIGN AND DEVELOPMENT OF A METHOD FOR INVESTIGATING THE PROPERTIES OF MATERIALS IN PLASTIC TORSION.

A dissertation submitted for the degree

of

Master of Philosophy

of the

University of London

by

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SUMMARY.

An analysis is given of the various factors which influence the suitability and accuracy of results derived from testing a solid cylindrical sample of a material in conditions of plastic torsion.

Recommendations are made for eliminating or reducing the influences of limiting factors with respect to the following.

(i) Geometry of the specimen.

(ii) The mode of construction of a suitable testing machine.

(iii) Suitable test procedure.

The difficulties associated with manufacture, within a University department, of a torsion testing machine, suitable both for laboratory teaching and post graduate research in plastic torsion at various controlled temperatures, are outlined and then rationalized into an appropriate design specification. The specification is particularly concerned with ease, simplicity and safety of use by unskilled operators; but tries to allow for future needs in research.

Alternative possible solutions to particular problems of the details of design of a torsion machine are considered in relation to each other and a positive solution to each problem is selected within the limitations of the specification.

In the appendices the final detailed drawings are reproduced, photographically, to a reduced scale, a satisfactory test procedure is summarized and, finally, certain calculations which diverge from conventional design practice are outlined in relation to the finished dimensions of the relevant machine components.

ACKNOWLEDGMENTS.

Grateful acknowledgment is made of the help given by Dr. F.A.A. Crane, as supervisor and of the advice and help given by Dr. D.L. Thomas.

Thanks are also due to my wife for the typing of this dissertation.

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CONTENTS.

1. ADVANTAGES AND LIMITATIONS OF TESTING PLASTICITY BY TORSION.

There are valid reasons for using torsional straining methods for assessing plasticity in preference to other methods. These reasons refer both to metallurgical advantages and to operational advantages as outlined below under the respective headings.

1.1. Metallurgical Advantages and Limitations.

Metallurgically there are five principal merits of the torsional straining method when compared with other methods such as direct compression or tension.

(i) Stress and strain conditions imposed by torsional loading in plastic flow are less complex and more readily interpreted than for any other system of loading. This evolves from the fact that the deforming stress is basically shearing in nature. Of course there are tensile and compressive components but they are of such an intensity and orientation that, with suitably proportioned, suitably loaded, test pieces manufactured from reasonably homogeneous material they do not greatly modify the mode of plastic flow until the point of fracture is approached. On the other hand, except for extremely ductile materials the ultimate criterion of failure is usually likely to be the localized resistance to tearing under tension.

Stress distribution under torsional loading is well understood for elastic stress conditions and is outlined in many textbooks on mechanics. (1,2,3,4,5.) It is open to quite rigorous solution, giving shearing, tension and compression stresses numerically equal in intensities. However, as plastic flow develops, solution becomes progressively less rigorous because of the uncertainties regarding the stress/strain characteristics of the material (4,5,6.). In elastic loading the shearing stress is a maximum at the outer surface of a cylindrical specimen subjected to axial torsion, see Fig.1.(a), and zero at the centre; the maximum shearing stress is related to the torque by:

$$\tau = \frac{T}{Z_p} \qquad (1)$$

where, τ is the maximum shearing stress, T is the torque and Z_p is the polar modulus of the section (equal to $\frac{\pi d^3}{16}$, d being the outside diameter of the section)

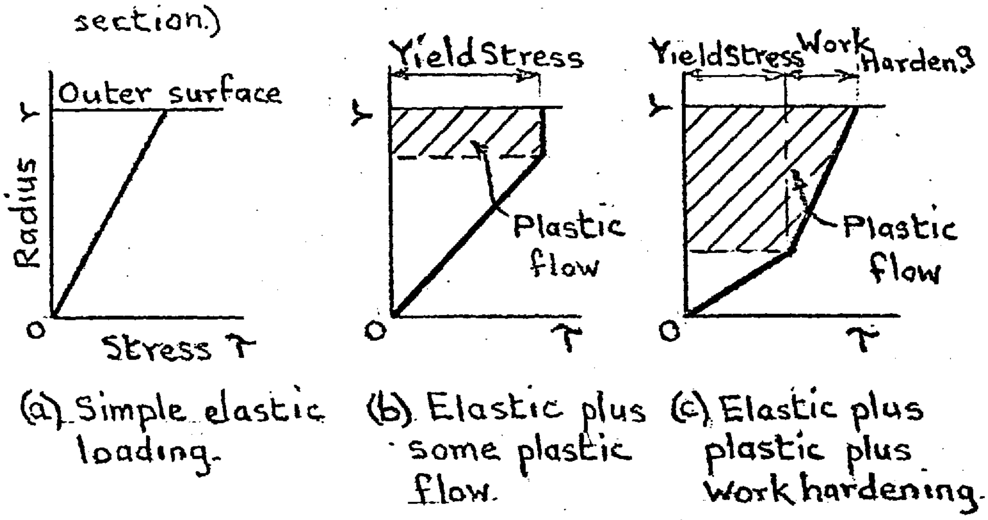


FIG.1. Stress distribution in the transverse planes of a torsionally loaded solid cylinder as plastic flow develops.

The situation changes as plastic flow begins, see Fig.1.(b), when the yield stress is exceeded at the outer surfaces. The stress distribution pattern is no longer triangular and a stress "bulge", of somewhat problematic shape depending on both material and conditions of testing, (such as temperature and strain rate, see below), begins to develop; so it becomes necessary to adopt some arbitrary assumptions with respect to distribution of plastic flow stress. The simplest assumption to make is that the material is ideally plastic and the yield stress remains constant after a completely elastic rise in stress to the point at which general yield occurs (i.e. τ = yield stress of the material at the relevant temperature). The theoretical torque, required to generate this form of stress distribution, depends on the location of the transition from elastic to plastic behaviour.

If such ideal plastic flow is pictured as developing right to the centre of the specimen in conditions of fully plastic flow, such that the elastic component becomes insignificant, then the situation would be that shown in Fig.2.(a).

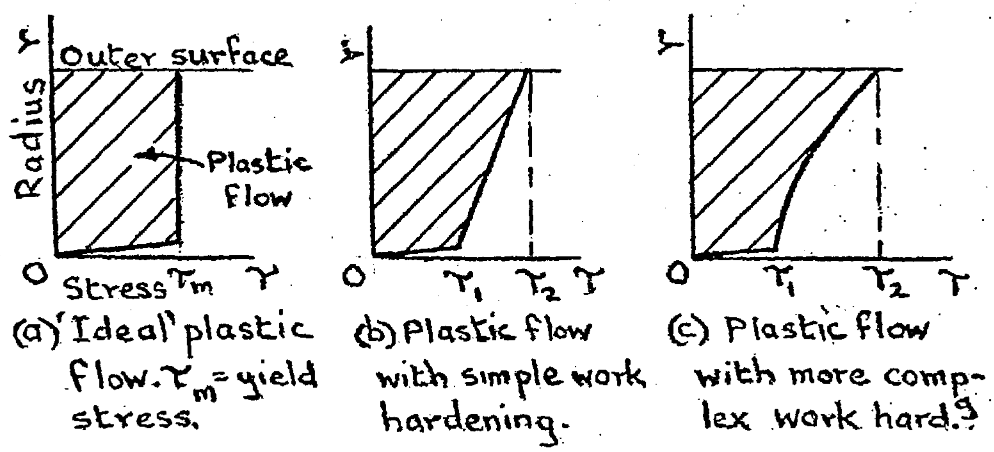


FIG.2. Stress distributions in solid cylindrical sections undergoing massive plastic flow in torsion.

The torque is then given by :-

$$\Gamma = \tau_m \times \frac{\pi d^2}{4} \times \frac{d}{3}$$

$$= \tau_m \times \frac{\pi d^3}{12}$$

where $\frac{d}{3}$ is the radius of gyration for a solid circular section and τ_m is the shearing yield stress

$$\therefore \tau_m = \frac{12\Gamma}{\pi d^3} \quad \text{- - - - - - - - - - - -} \quad (2)$$

The accuracy of this assumption clearly varies appreciably with the material, because of the differences in real stress/strain characteristics, and with the particular conditions in which a test is made, because the characteristics will vary with temperature and with straining rate.

Most materials show some degree of work hardening with increasing amount of plastic strain depending on the temperature at which the straining takes place. In work hardening the yield stress is sometimes nearly proportionate to the total amount of plastic strain (i.e. the work hardening part of the stress/strain curve is linear); therefore, in these conditions, the stress distribution diagram for a circular section under plastic torsion would take the form shown in Fig.1.(c), simplifying to that shown in Fig.2.(b); because the elastic range is usually insignificant once full plastic flow begins. The torque in this case is a sum of the torque due to basic yield plus the torque component due to the effects of work hardening. If the yield stress rises uniformly with total strain, as shown in Fig.2.(b), the work hardening component is similar in magnitude to the equivalent elastic torque (Equation 1).

Hence:-

$$\Gamma = \frac{\pi \tau_1 d^3}{12} + (\tau_2 - \tau_1) \times \frac{\pi d^3}{16}; \quad \text{where } \tau_1 = \text{minimum yield stress}$$

$$= \pi d^3 \left(\frac{\tau_1}{12} + \frac{\tau_2 - \tau_1}{16}\right) \quad \text{and } \tau_2 = \text{maximum yield stress}$$

$$= \frac{\pi d^3}{16}\left(\frac{\tau_1}{3} + \tau_2\right)$$

$$= \frac{\pi d^3}{48}(\tau_1 + 3\tau_2)$$

$$\text{or } \tau_1 + 3\tau_2 = \frac{48\Gamma}{\pi d^3}$$

$$\text{but } \tau_2 = \tau_1 + \frac{dh}{2} \quad \text{where } h = 2\left(\frac{\tau_2 - \tau_1}{d}\right)$$

$$\therefore \tau_1 + 3\left(\tau_1 + \frac{dh}{2}\right) = \frac{48\Gamma}{\pi d^3}$$

$$\therefore 4\tau_1 + \frac{3dh}{2} = \frac{48\Gamma}{\pi d^3}$$

$$\therefore \tau_1 = \frac{12\Gamma}{\pi d^3} - \frac{3dh}{8} \quad \text{- - - - - - - - - (3)}$$

Of the three variables (the torque Γ, the minimum yield stress τ_1, and the slope h) the value of ~~the~~ two must be known before this equation can be solved. The torque Γ may be derived directly by experiment and τ_1 is likely to be about equal to the static shearing yield strength of the material at the relevant test temperature, also obtainable by suitable experiment; but h is of more doubtful value since it is dependent, for its accuracy, both on the uniformity of the rate of work hardening, derived from the appropriate stress/strain diagram for the material, and on the value of the total strain at the surface of the test cylinder, because the latter governs the value of the maximum yield stress that is attained.

i.e. $\tau_2 - \tau_1 = \mathcal{E} x$; where x is the rate of work hardening derived from the slope of the torsional plastic stress/strain diagram and $\mathcal{E}$ is the total surface shear strain (assumed to be very much greater than the elastic strain, permitting the latter to be ignored). However, $\mathcal{E} = \frac{\pi N d}{l}$, where N = number of twists given to the specimen, l = the axial length of the test specimen and $\frac{d}{2}$ = the radius of the outer surface of the test cylinder (always proportional to $\mathcal{E}$).

Substituting in:- $h = 2\left(\frac{\tau_2 - \tau_1}{d}\right)$ (see above)

$$h = \frac{2\mathcal{E} x}{d}$$

$$= \frac{2\pi N x}{l}$$

Thus, the value of h can be derived from the relevant stress/strain curve for the material obtained in tension and converted to shearing by the use of a suitable yield criterion.

However, if the rate of work hardening of the material varies significantly with the amount of strain, it is apparent that to adopt equation (3) too arbitrarily could lead to massive inaccuracies in the derived results.

It can be seen that not only the total strain but also the rate of straining within a test cylinder both vary with the radial position.

$$\gamma = \frac{\phi r}{l} \qquad (4)$$

where γ is the shear strain at radius r, ϕ is the angle of twist, in radians, of the cylindrical section and l is the axial length of cylindrical section under consideration

$$\text{or } \gamma = \frac{2\pi N r}{l} \qquad (5)$$

where N is the total number of twists given to the specimen.

$$\therefore \dot{\gamma} = \frac{2\pi n r}{l} \qquad (6)$$

where $\dot{\gamma}$ is the strain rate at radius r and n is the number of twists per unit time.

(If l is made equal to $2\pi r$, then N and n are numerically equal to the total strain and rate of strain, respectively, at the surface of the specimen and this can be very convenient in experimental work).

The inherent variation in rate of straining within a test piece can introduce further problems of calculation, depending to a considerable extent on the temperature at which the test is made. The shearing yield stress may vary not only with total strain and with strain rate, but its relationship with both may vary with temperature.

At normal and lower temperatures the yield stress will rise with increasing rate of straining (7,8.) making the stress distribution of Fig.2.(b) take a form such as that shown in Fig.2.(c). This would entail a change in the torque equation, quoted above, since the maximum shearing stress at the surface is now the sum of the minimum yield stress at the centre plus some function of the total strain, plus a function of the strain rate

$$\text{i.e. } \tau_2 = \tau_1 + f(N\tfrac{d}{2}) + f(\dot{N}\tfrac{d}{2}) \qquad (7)$$

In addition, the mean stress difference can no longer be assumed to act through the polar radius of gyration of the section, since the stress difference does not increase uniformly from the centre to the outside surface; therefore, the basic torque equation requires significant modification.

At elevated temperature the effects of total strain and rate of straining may not even be as straightforward as the above (particularly at temperatures approaching the melting temperature of the material under consideration). This occurs because, not only does the rate of increase of yield stress not vary uniformly with either total strain and with strain rate (9,10,11) but, with larger amounts of total strain and with higher rates of strain, the temperature may tend to rise and vary significantly during testing due to internal friction. A further factor is that structural changes, such as recrystallization, precipitation and solution, within the material, if they take place, may greatly modify stress/strain behaviour. Thus, there is a rate of straining at which natural cooling can no longer keep the specimen temperature at the control level and its internal temperature begins to rise and vary, the yield stress begins to fall, and accurate prediction of internal stress distribution becomes impractical. On the other hand a combination of these effects may result, for a particular material, in a strain behaviour approximate to that of the ideally plastic material, typified in equation (2), in which the yield stress remains constant (9,10,11,17).

If a reasonable idea can be obtained of the variation of stress across a particular material it becomes possible to make fairly accurate calculations of stress and strain behaviour, if the torque, the applied strain rate and the applied total strain can each be measured with reasonable accuracy.

In putting these ideas into effect good results have been obtained by Robbins et al (12) with an equation of the type:

$$\tau_s = 4\Gamma\frac{(3 + i + m)}{\pi d^3} \qquad (8)$$

where τ_s is the shearing stress at the surface of the test cylinder, i is the slope of the log Γ v. log γ curve at constant strain rate and m is the slope of the log Γ v. log $\dot{\gamma}$ curve at constant strain.

This they claim to be no more than $\pm 3\%$ in error when using suitably-derived values for steel, even if i is assumed to be zero for test temperatures greater than half of the absolute melting temperature of the material (i.e. for metallic materials, leading to an included error not greater than $\pm 1.5\%$, m being temperature dependent and varying appropriately between 0 and 0.2 for steels at temperatures up to 1200°C.

A recent suggestion, made by Barraclough and others (17), is to use the concept of an "effective radius", analogous to a radius of gyration, in conjunction with a similar type of equation. The "effective radius", r_m, is that radius of the test cylinder section at which the actual stress, τ_x, is equal to the mean stress τ_m which would have to prevail throughout the section to give equal torque and satisfy the equation (2) which becomes:-

$$\Gamma = \frac{2\pi\tau_m}{3} r^3 \qquad (9)$$

where r is the outside radius of the solid test cylinder.

The radius r_m has been shown (17) to be relatively insensitive to the effects of variations in work hardening etc. as affected by total strain and strain rate. Therefore, if this radius is identified on the specimen it gives a position in which material is being subjected to quite accurately known conditions of testing, even with quite widely differing ratios of gauge length to specimen diameter in the same test material. The error due to this assumption, when i + m is in the range of 0.2 to 0.4, is claimed to be less than 0.5%.

To satisfy the condition $\gamma_m = \gamma_x$ the value of r_m is obtained from an equation of the form

$$r_m = \left\{\frac{3}{3 + i + m} \times r^{i+m}\right\}^{\frac{1}{i+m}} \quad \text{------} \quad (10)$$

This identification of r_m has other advantages when metallographic examination is attempted after a test, see (iii) below.

It is possible to correlate one form of plastic strain behaviour with another form and then to develop a link between tests in plastic torsion with the other form of test, e.g. creep testing. A suitable relationship of this kind makes possible the cross-checking of results and the verification or derivation of appropriate constants (10). This has been done effectively by Sellars and Tegart (13) for some materials with a relationship of the form:-

$$\dot{\varepsilon} = A(\sinh\alpha\tau)^{n^1} \exp(-Q/RT) \quad \text{------} \quad (11)$$

A, α and n^1 being temperature independent constants, Q the activation energy, T the temperature, and α and n^1 are related by a constant $\beta = \alpha n^1$

These various relationships give torsional plastic strain testing a potential facility and clarity, with respect to stress and strain conditions in specific materials, not possessed by any other current plastic strain testing system.

The problem of temperature rise resulting from evolved heat of deformation is still a source of inaccuracy, since this change will alter the yield stress and the work hardening characteristics of the material; but the situation is no worse than with other strain-rate sensitive testing systems and, indeed can be a good deal better with suitably plastic materials in which stable conditions of temperature can be established with appropriately prolonged straining.

(ii) The outermost annular layer of a torsional test piece can be subjected to approximately constant conditions of temperature, strain and, probably, stress over the whole of its ~~volumes~~ volume during a test (equations 5, 6 and 7). If a suitable control, recording and observing system is provided it is possible to study behaviour in this layer with the certainty that it is reasonably representative of the behaviour of the material under the imposed conditions. Also, if provision for direct observation cannot be made, and it is ensured that the tested specimen is not damaged when extracting it from the test apparatus, the representative characteristics of the deformed surface are left unchanged from the test and remain for study at leisure. No other test system gives such ready facility for examining the effects of such accurately known conditions.

The limitation on this facility is that the flow conditions of this annular layer are not typical of those further into the test piece, since they are modified by the effect of the exposed free surface, both with respect to the local mode of straining and with respect to the effect of any surface reaction that may be taking place with the ambient atmosphere. Nevertheless, useful information may be derived from a study of such test surfaces with respect to the indications they may give on mode and direction of surface flow and on sub-surface turbulence.

Most workers do not consider it worth while to go to the complexity involved in making direct observations during straining; therefore this aspect of study is commonly limited to the post-straining state.

(iii) The volume of material, in each cylindrical annulus, which is under nearly identical conditions of temperature, strain, and stress is significantly large and its location is easy to identify during subsequent macroscopic or microscopical examination. Thus, if a cross-section is being examined the radial distance from the centre identifies, within reasonably consistent limits the local conditions of strain and stress and an immediate visual comparison of any significant structural variations is possible. Similar facility is available in longitudinal sections provided that the location of the section relative to the centre line of the specimen is known. This becomes particularly significant if the concept of an "effective radius" is used,(17) see p 14 above, since the location of this radial position is likely to be associated with quite precisely known conditions of testing,free from the uncertainties of behaviour likely in other locations.

Examination may readily be taken a stage further and use may be made of the electron microscope, if required, since even the smallest torsional test pieces usually contain an amply sufficient supply of material in each relevant condition.

The limitation on this aspect of the torsional testing system lies in the fact that the strain and stress conditions,typically obtainable in torsion,may not correspond sufficiently closely to those prevailing in an applied metal-forming operation to give directly comparable results if these are required. It may be necessary in this situation to derive fundamental features of behaviour, as far as this is possible, by torsional methods and reinterpret them in the light of what is known of the actual conditions in the particular applied operation (14,15,16).

(iv) Control of thermal conditions is usually practical in torsional testing, because it is possible to keep the heating system in position and operating during a test. Uniform external heating is usually applicable when steady, not too high, temperature conditions are required, and controlled rapid temperature changes are possible within the respective limits of the chosen specimen size, the heating system and the method of cooling. Immediate or delayed quenching is possible when required. Thus, with suitable equipment a wide range of controlled thermal conditions may be applied.

(v) Strain may be applied in controllable sequences with suitable equipment. Thus, with fairly simple equipment, time-controlled "bursts" of straining, or given total strain increments, may be applied, alternating with controllable time gaps and/or changes in temperature. Cycles of treatment may be repeated indefinitely to the limit that the material will withstand. Thus, many sequences of strain application that might be encountered in applied metal working may be reproduced accurately, at least with respect to amount and rate of strain and with respect to temperature.

The limitation on this approach is set by the restricted mode of deformation possible in torsional straining, but it is claimed that correlation is possible with respect to forging, rolling and extrusion (14,15,16).

1.2. Physical Advantages and Limitations.

The physical aspects of testing plasticity by means of torsion may be considered under several headings including specimen shape, mechanical aspects, measurement of stress, thermal control and power requirements.

1.2.1. Specimen Geometry.

The ideal shape for a specimen which is to be subjected to torsion, whether elastic or plastic, is cylindrical principally because the deformation of non-cylindrical shapes is difficult to analyse but also because the cylindrical form is the easiest to produce with accuracy and is therefore one of the cheapest. Turning and grinding facilities are generally available, making reproducibility of shape, size, and surface finish easy to obtain. Application of the torsion may be made through quite simple end-shapes which may take such basic forms as those shown in Fig. 3. Different end-shapes may be used to suit particular

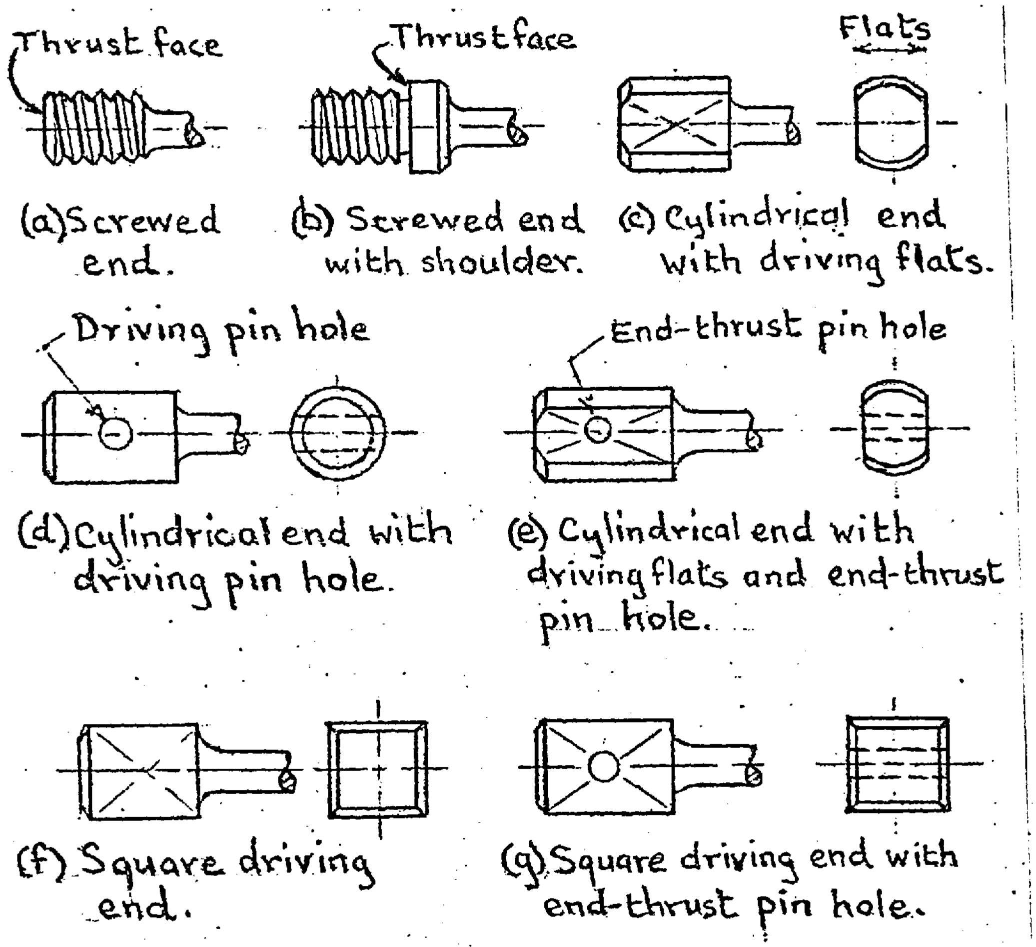


FIG.3. End shapes for torsional test pieces.

test situations. Types (a) and (b) locate in the threads and will transmit end thrust in either direction but can be used to apply torsion only in one direction, the same as that of the thread (usually right-handed). Type (b) is a little more expensive to produce than (a). Type (c) locates directly from its outside diameter and will transmit torsion in either direction but will not transmit end thrust. An extra machining set-up is required for making the flats which have to be quite accurately machined. Type (d) locates directly from its outside diameter and can transmit drive or end thrust, in any direction, through a shear pin. To permit the use of an adequate size of pin, the cylindrical end may have to be comparatively large in diameter. Type (e) is similar to type (c) with the addition of a pinhole for a shear pin to transmit end thrust; thus necessitating one more machining operation. The pinhole need not be as large as that in (d), because only end thrust has to be taken and the magnitude of this is likely to be small compared to the torsional driving forces. Type (f) appears simple but the square requires very careful, rather costly, machining since it both locates and drives the specimen. It will not take any end thrust. Type (g) is similar to (f) but incorporates a pin to take end thrust, also requiring an extra machining operation.

It is obvious that to define a precise gauge length, each one of these types must have significantly enlarged ends. This creates a difficulty in ensuring absolutely uniform transmission of torsion into the gauge length adjacent to the end. Some workers prefer to machine sharp corners between the ends and the gauge length (22) since this makes the torsion more immediately uniform in the gauge length but it also increases the risk of unbalanced flow or failure at the corner due to the stress concentration. Most workers prefer to have a fillet radius in each corner, either using the distance between the shoulders as the normal gauge length or making some allowance for the presence of the fillets (see below). A fillet radius of a size about equal to a quarter of the size of the gauge diameter seems a typical maximum but adjustment upwards might be required in the case of a particularly notch-sensitive material or to suit particular test conditions.

TABLE 1

Some Features of Test Pieces and Machines used in Plastic Torsion Testing.

Ref.	Basic Material	Gauge sizes: Gauge Dia d mm	Gauge sizes: L/d	Max. Temper^re. deg C	Max. Torque Nm	Test* Piece End Type	Max. Strain Rate /sec.
18	Al	6.0	8	600	10	(b)	15
19	Al	6.35	4	600	-	(c)	-
21	Al;Cu;Ni	12.7 max	5	up to 1000	45	(c)	-
20	Cu; Ni	10.0	4	1000	45	(c)	-
12	Fe	5.08	5	1400	-	(e)	18
22	Fe	6.0	1	700	20	(d)	180
23,24, 25	Fe	7.14	4	1200	11.5	(b)	10

* Referring to Fig.3.

\- Not quoted

Length between the end shoulders of test pieces, see Fig.4., varies a good deal with different workers, as shown in Table 1, depending to a certain extent on the limitations of their equipment and other facilities. A compromise usually has

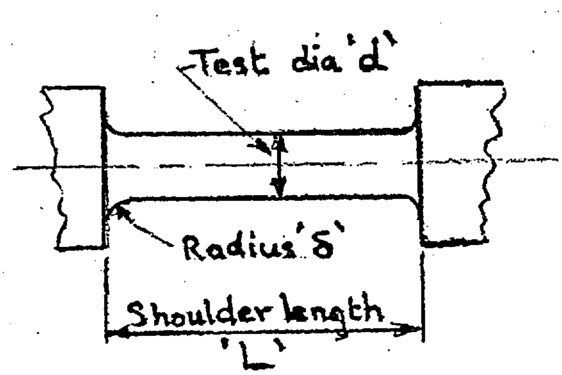
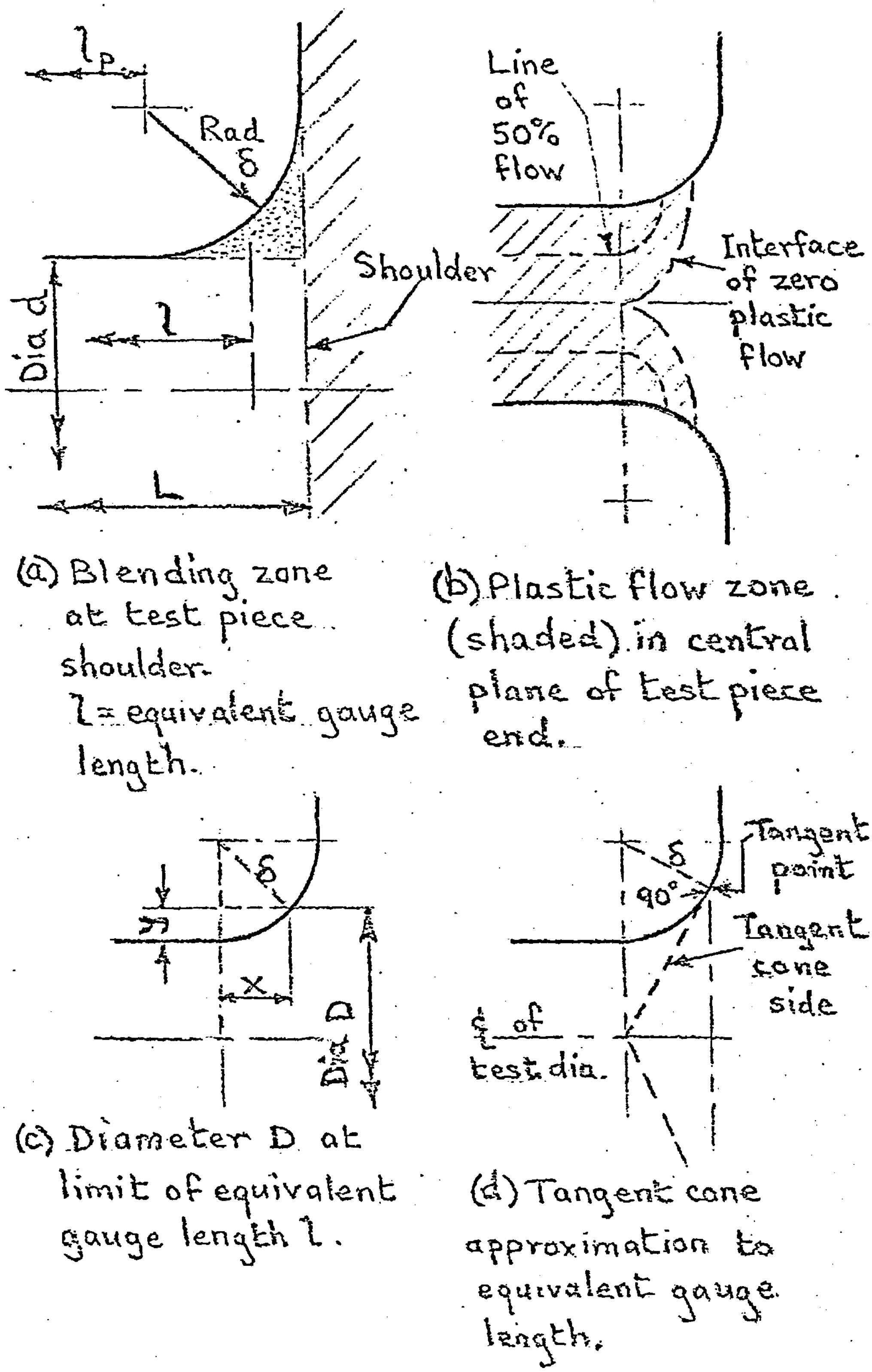


FIG.4. The typical features of the straining zone of a torsional test piece.

to be accepted in one way or another. If the cylindrical length is too great relative to its diameter then inherent variations in the consistency of the material may induce torsional instability, with consequent buckling during a test, in spite of extreme accuracy in locating and loading the specimen. About 8:1 seems to be the absolute maximum allowable for the ratio of gauge length to test section diameter. On the other hand, if the length between the shoulders is made too short the accuracy of location and alignment become much more critical factors and closer tolerances have to be maintained in the strain-applying equipment to avoid externally-induced misalignment of loading, with consequent inaccuracies in load measurement and/or externally-caused bending or buckling of the specimen. Of course, shortening the length between the shoulders, relative to a given maximum speed of twisting in a machine, is one way of increasing the maximum strain rate with respect to a fixed test diameter of the specimen. In some cases, the length between the shoulders is determined by the desire to make the effective torsional gauge length equal to π times the gauge diameter, see note to Equation 6 (after allowing for the blending fillets, see below).

If, as is usual in tests of plasticity in torsion, strain measurements are derived directly from the number of turns applied by the testing machine to the specimen's shoulder length, allowance must be made for the mode of geometric transition between the gauge diameter d (Fig.5.) and the respective shoulders. A sharp corner might be used but this must cause a very severe stress concentration likely to give (i) anomalous plastic flow behaviour and (ii) premature failure in or near the plane of a shoulder. It is probably for this reason that most workers use blending fillets, of some description, between the gauge diameter and the shoulders. A few workers (22) do use a sharp corner on test pieces with very low L/d ratios, but there must be some doubts as to the accuracy of their results.



(a) Blending zone at test piece shoulder. l = equivalent gauge length.

(b) Plastic flow zone (shaded) in central plane of test piece end.

(c) Diameter D at limit of equivalent gauge length l.

(d) Tangent cone approximation to equivalent gauge length.

FIG.5. Conditions in test piece ends during straining.

If blending fillets are used, the question arises concerning what allowance to make for their influence. Barraclough et al (17), in the context of use of hollow cylindrical specimens in which the transitional flow will not conform to exactly the same pattern as that in solid sections, have taken some account of the effects by making an arbitrary allowance on the test length equal to half of the fillet radius on each end (i.e. $L = l + \delta$). However, as far as can be ascertained, it looks as if most other workers have tended to ignore this aspect and base their strain measurements either on the length between the roots of fillets or the full length between the shoulders. It is apparent that both of these latter procedures cannot be correct and that strain measurement differences, proportionate to the related total axial lengths of fillet, must exist, amounting to at least $6\frac{1}{2}\%$ even with the greatest ratios of L/d and with modest fillet radii, when the error should be at its minimum.

Much more accurate assessment is possible if some account is taken of the mode of plastic flow in the blending fillets (17).

The problem then is to determine how much allowance to make. Blending is most logically and simply done by using a simple radiused fillet, see Fig.5.(a), so this system is considered here; but the same arguments may apply equally to other forms of blend if their use is considered necessary.

Normal plastic flow theory does not offer a reliable formula with which to tackle this problem, but certain limits can be set. It is quite obvious that torsional plastic flow will not end abruptly in any one special plane normal to the axis of the test piece. Within the gauge length, up to the beginning of the fillet, straining will be fully plastic. From the shoulders outwards deformation will be entirely elastic. Between the two, each cross-section will consist of a central elastic core surrounded by an annulus in plastic flow. The diameter of the former will increase progressively as a shoulder is approached. It is to be expected that, with a fillet of significant size, plastic flow will terminate somewhere within the fillet, see Fig.5.(b). The surface circumferential boundary line, that marks the end of plastic flow, must be the termination of a roughly conical type of boundary interface, which blends from a point lying on the axis, in the plane of the centre of curvature of the fillet radius, out to the surface at the appropriate position on the fillet. That is, the annular zone of varying plastic strain (zero at the centre and a maximum at the surface), normally present in a gauge cylinder undergoing plastic torsional strain, must blend round and outwards from its centre until it ends at the fillet surface. Thus, no single plane can truly represent the termination of total plastic flow; but the one which best represents it is probably that plane in which there is an annular area of plastic flow taking 50% of the total torque; because this plane should represent the average flow behaviour of the end zone of plastic flow. Since its exact location depends on the flow characteristics of the material, the relative position of this plane in relation to a particular size of fillet between the gauge diameter and the shoulder of the specimen is likely to vary with different materials and perhaps with temperature. However, certain limits can be set between which the 50% plane is likely to be located and its position Fig.5.(c), may be approximated to give a more accurate gauge length than is given by assuming that strain ends at the beginnings of the centres of curvature or at the shoulders or even by other, less extreme, arbitrary assumptions.

The extreme limit for the position of the termination of all plastic flow in a perfectly plastic material must lie on the diameter of cross-section d_E at which the area is just sufficient to resist the full torque by elastic strain alone. That is, this diameter must be that at which

$\Gamma = \frac{\pi}{16} d_E^3 \tau_s$ where Γ = total torque and τ_s = shearing yield stress of the material

but $\Gamma = \frac{\pi}{12} d^3 \tau_s$ for full plastic flow in the parallel length

$$\therefore \frac{\pi}{16} d_E^3 \tau_s = \frac{\pi}{12} d^3 \tau_s$$

Hence $d_E = 1.1006d$, giving an extreme upper limit for any assumption of representative plastic flow in the ends.

Since this is the extreme case; a closer approximation should be obtained from the equation :-

$\Gamma = \frac{\pi}{16} d_e^3 \tau_s + \frac{\pi}{12}(d_p^3 - d_e^3)\tau_s$ (in which d_p is the outside diameter of the section at which the torque is equally shared between plastic and elastic flow and d_e is the diameter of the central elastically strained area) for the condition when:-

$$\frac{\Gamma}{2} = \frac{\pi}{16} d_e^3 \tau_s$$

$$\therefore \frac{\pi}{16} d_e^3 \tau_s = \tfrac{1}{2} \times \left(\frac{\pi}{12} d^3 \tau_s\right)$$

$$\therefore \frac{d_e^3}{16} = \frac{d^3}{24}$$

hence $d_e = 0.8736d$

From $\frac{\pi}{16} d_e^3 \tau_s = \frac{\pi}{12} (d_p^3 - d_e^3)\tau_s$

$$\frac{d_e^3}{4} = \frac{d_p^3 - d_e^3}{3}$$

Substituting for d_e

$$\frac{(0.8736d)^3}{4} = \frac{d_p^3 - (0.8736d)^3}{3}$$

$$\therefore d_p = 1.053d$$

The accuracy of this value depends on the accuracy of the assumption that the amount of plastic strain in this section is proportionate to the relative amount of torque absorbed in causing plastic flow. This would be exactly true for the perfectly plastic material but it will be less accurate for real materials.

Thus, for a perfectly plastic material, D lies between 1.1006d and 1.053d, probably being much nearer 1.053d than 1.1006d, since 1.1006d represents zero plastic flow.

However, in a real material the elasticity of the material and, more particularly, its strain-hardening characteristics will raise the lower level significantly, but by an indeterminate amount. On the other hand, the minimum diameter of zero plastic strain will remain almost unchanged. A reasonably close approximation for a real material would probably be:-

$$D = \frac{1.1006 + 1.053d}{2} = 1.077d$$

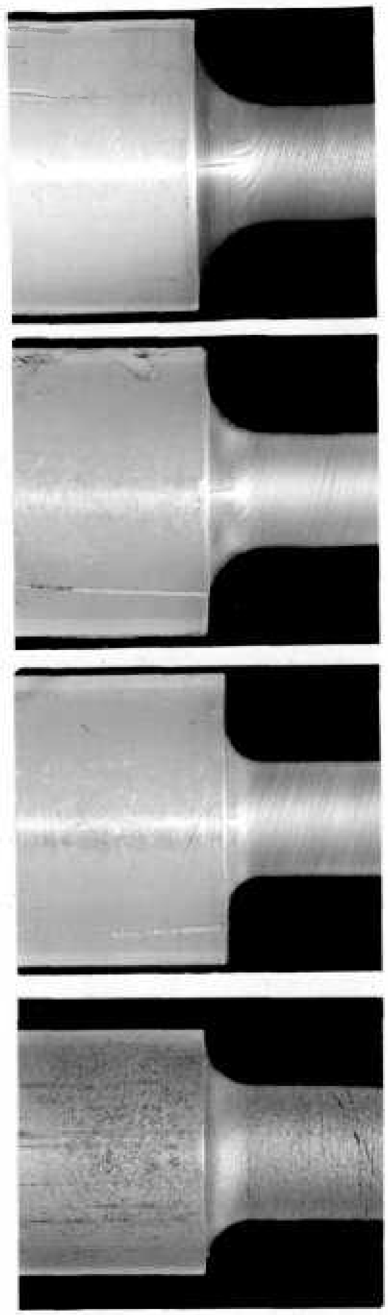
A few tests were made on aluminium -5% magnesium alloy, at room temperature, to assess the validities of these assumptions. Three sizes of blending radius were used, respectively $\frac{d}{2}$, $\frac{d}{4}$ and $\frac{d}{5}$, see Figs. 6 and 7. (For comparison, an aluminium bronze specimen tested at 600°C is shown at (d) in Figs.6 and 7. It can be seen, from the photographs, that the behaviour seems to conform reasonably closely to that predicted, both with respect to surface flow and internal flow. Each of the specimens was scribed longitudinally on the specimen surface, before testing, and the diameter on the radius at which the scribed line was deflected half-way between its original axial direction and the final helical angle of flow, see Fig.6, was measured as accurately as possible by means of a travelling microscope. The results are given in Table 2.

(a)

(b)

(c)

(d)



×2

FIG. 6.

External appearances of ends of specimens with differing ratios of δ/d tested in plastic torsion.

(a) Al.5%Mg.tested at room temperature $\delta/d = \frac{1}{2}$

(b) do $\delta/d = \frac{1}{3}$

(c) do $\delta/d = \frac{1}{4}$

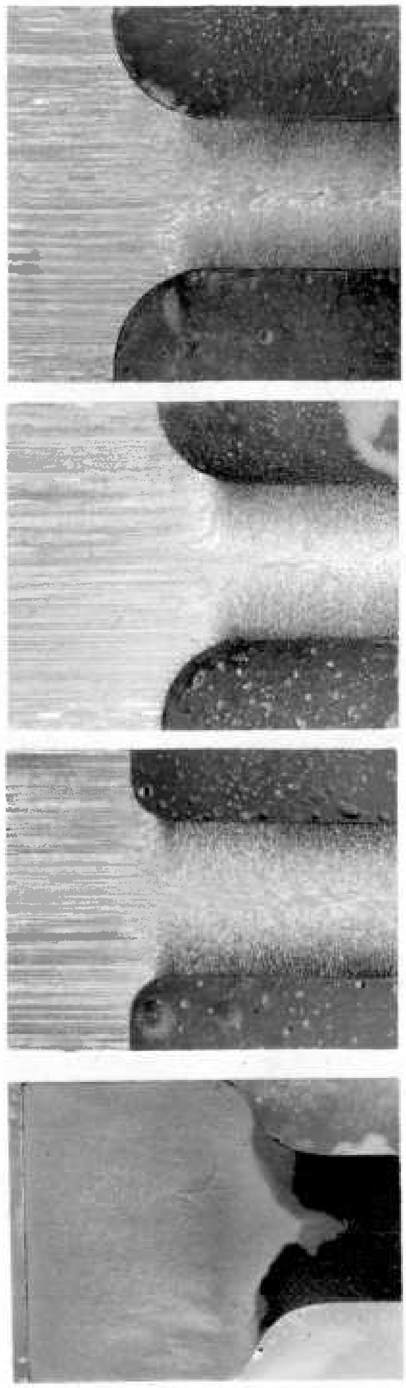
(d) Cu. 10% Al 5%Fe. 5%Ni. tested at 600° C $\delta/d = \frac{1}{3}$ (Note. There were no initial longitudinal reference scribe marks on the surfaces of this specimen; therefore the graduation in plastic flow at the end of the test length is not so clearly indicated.)

(a)

(b)

(c)

(d)



×3.

FIG. 7.

Macrosections on the central planes of the ends of specimens tested in plastic torsion.

(a) Al.5%Mg.tested at room temperature $\delta/d = \frac{1}{2}$

(b) do. $\delta/d = \frac{1}{3}$

(c) do. $\delta/d = \frac{1}{4}$

(d) Cu.10% Al.5%Fe.5%Ni.tested at 600°C $\delta/d = \frac{1}{3}$ (Note. It was very difficult in this material to etch up the structural modification caused by the plastic flow, although the effect was visible; so the extreme outline of the plastic flow zone has been artificially marked, as accurately as possible, without attempting to indicate variations in the flow structure.)

TABLE 2.

Approximate diameter D at 50% apparent plastic angular deflection in specimen ends of Al.5% Mg. tested at room temperature.

Fillet radius	Gauge diameter d mm	Average D mm	D/d
$d/2$	8	8.78	1.098
$d/4$	8	8.70	1.091
$d/5$	8	8.76	1.095

Average value of D/d = 1.094

From this it looks as if D/d = 1.077 is a reasonable guess for this material, bearing in mind that the actual value of D was not measured to a high degree of accuracy.

It is of interest to make some comparison of the probable limits of accuracy of calculation of strain with different methods for determining L, with differing fillet radii.

The value of D may be approached in several ways.

1. By direct calculation from the ratio using the assumed diameter.
2. By a system of approximation such as that based on the circumference at which a tangent cone, see Fig.5.(d) touches the fillet. (Note that this gives a decreasing value of D for decreasing fillet radius.)
3. By an arbitrary increment based on the size of the fillet radius (17).

In the following tables the relative accuracies are tabulated on the basis of the relevant equivalent gauge length being equal to 3.1416d and being used as the reference gauge length.

TABLE 3.

Basic length between shoulders for a gauge length $= 3.1416d$ at $D = 1.077d$. Symbols as in Fig.5.(c). Values all related to d.

$\frac{d}{\delta}$	x	$(\delta - x)$	L $l + 2(d-x)$ *	Error in gauge length relative to observed values Table 2 D = 1.094
2	0.19241	0.30759	3.75677 (3.71831)	+ 1%
3	0.15552	0.17781	3.49721 (3.46693)	+ 0.8%
4	0.13330	0.11670	3.37498 (3.34977)	+ 0.7%
5	0.11798	0.08202	3.30563 (3.28399)	+ 0.6%

* Equivalent corrected lengths for D = 1.094d shown in brackets.

TABLE 4.

Length between shoulders with $l = 3.1416d$ at tangent cone positions, see Fig.5.(d). Values all related to d. Compared with $l = 3.1416d$ at $D = 1.077d$.

$\frac{d}{\delta}$	Equivalent D	x	$(\delta - x)$	$L_{T.C.}$	% error in gauge length
2	1.50003	0.43302	0.06698	3.27556	- 13%
3	1.39976	0.30549	0.02781	3.1972	- 8½%
4	1.33416	0.23571	0.01429	3.17098	- 6%
5	1.28543	0.19166	0.00834	3.15827	- 4½%

TABLE 5.

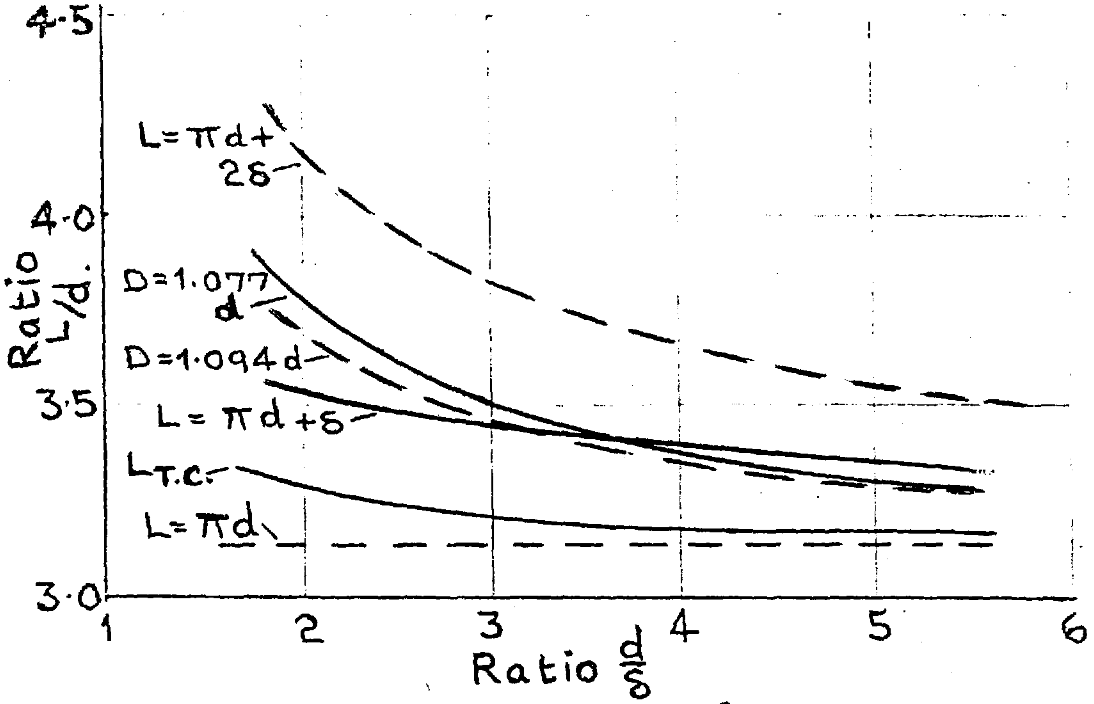
Errors with fixed lengths between shoulders of 3.1416d, + δ and 3.1416d + 2δ. Compared with l = 3.1416d at D = 1.077d

$\frac{d}{\delta}$	Error with L = 3.1416d	Error with L = 3.1416d + δ	Error with L = 3.1416d + 2δ
2	- 16½%	- 3%	+ 10%
3	- 10%	- 1%	+ 9%
4	- 7%	+ ½%	+ 8%
5	- 5%	+ 1%	+ 7%

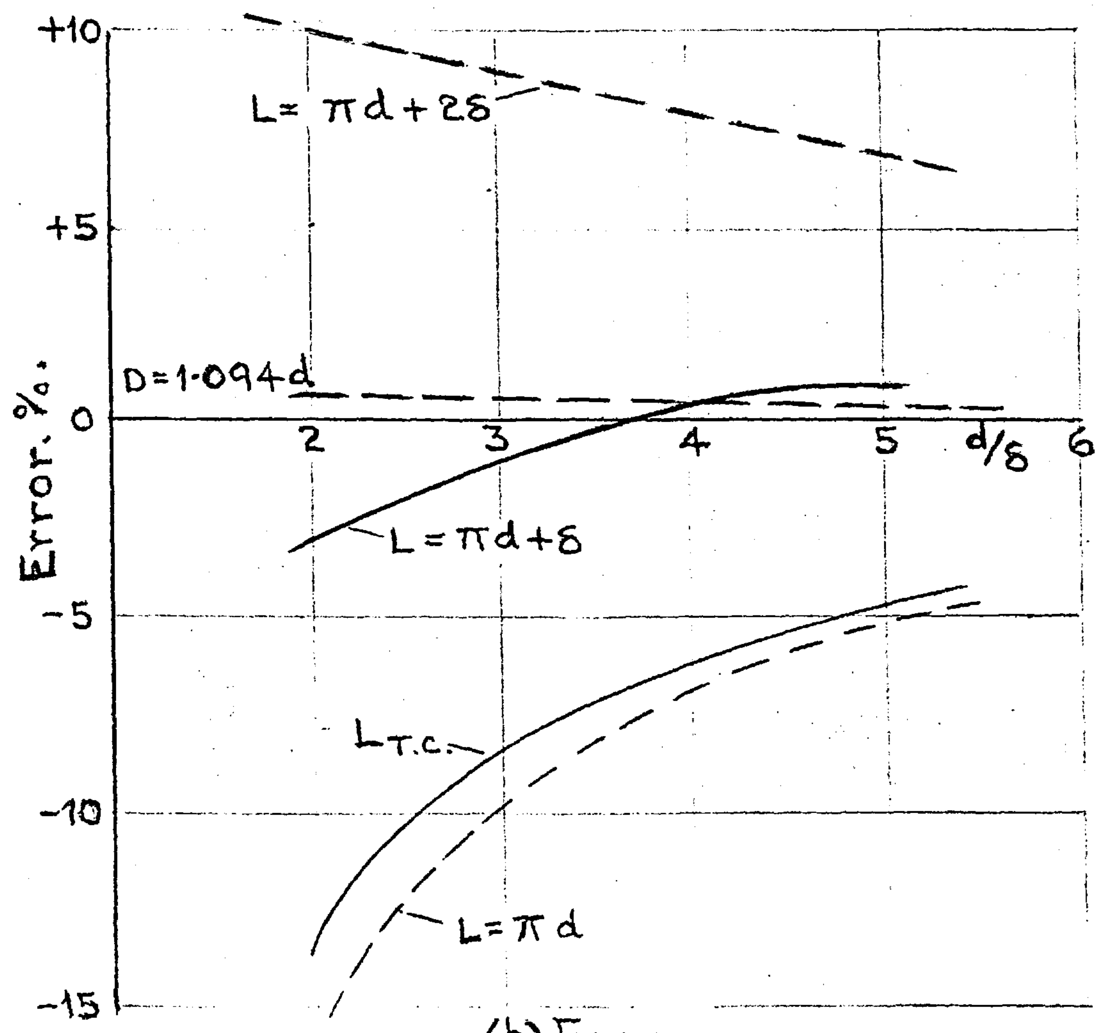
These results are recorded graphically in Fig.8. and from this it certainly seems worth while to make a realistic estimate of the equivalent gauge length, although L = 3.1416d + δ (17) gives a fairly close approximation for values of $\frac{d}{\delta}$ lying between 3 and 5.

With smaller ratios of $\frac{l}{d}$ the strain errors would be proportionately greater than those shown in the tables and with smaller radii proportionately less, provided that anomalous flow and/or premature failure does not occur in the fillet, under the influence of stress concentration. Examination of Fig.7 will show that plastic flow under the sharpest fillet does not conform to the pattern found in the other two and that anomalous behaviour is developing.

Since no analysis of these sources of error is given in any of the current literature on testing in plastic torsion, this leads to some doubt concerning the accuracies claimed for much of the experimental work. In practice, even when a realistic gauge length has been estimated, it is desirable to correlate observed strain markings on the specimen with the assumed gauge length, see Fig.6, and to make a suitable correction if that seems appropriate or necessary.



(a) Relative size of L.



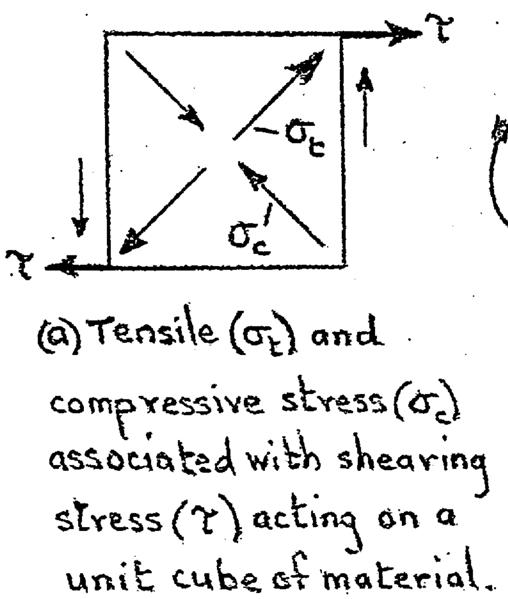
(b) Error.

FIG. 8. Various values of length between shoulders, L, and blending radius, δ, showing associated error in gauge length, related to l = 3.1416d at D = 1.077d

A reasonable compromise for the values of $\frac{l}{d}$ and δ would seem to be $\frac{l}{d} = \pi$ and $\delta = \frac{d}{5}$; giving a maximum error of about $+\frac{1}{2}\%$ without creating too severe a stress concentration at the ends.

Of course there is always a possibility of change in axial length during testing; sometimes a shortening and sometimes a lengthening.

Although not clearly understood, it seems probable that the cause of these phenomena is linked with the tensile and compressive components of stress always present when a shearing stress is applied, see Fig.9.(a). In the case of a cylinder subjected to coaxial torsion, see Fig.9.(b), a helically disposed tensile force, winding in the direction of torsion, is set up in conjunction with a helically disposed compressive force winding in the opposite direction. The resultant stresses are at their maxima at the surface and tail off to zero at the central longitudinal axis. In theory these stresses vary with and, in



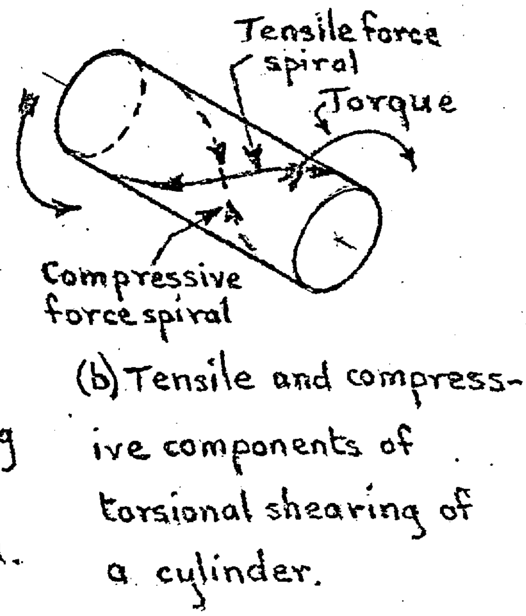


FIG.9. Tension and compression associated with shearing.

intensity, are equal to the shearing stress (i.e. $\tau = \sigma_t = \sigma_c$) and should not initiate plastic flow in these conditions. However, the resistance of a material to plastic flow need not be quite so simple once torsional shearing gets into conditions of yield. In fact, no one knows how a material's resistance to tension and compression varies during massive plastic flow conditions; but, in view of the observed phenomenon of changes in axial length of cylinders subjected to coaxial plastic torsion, it seems evident that the respective resistances to tension and compression must change, and not necessarily in step. It appears likely that such factors as crystallographic structure, crystallographic orientation, grain size and heterogeneity can each affect this behaviour in particular situations.

If tensile and compressive resistances fall, but do not vary in step so that their influences balance out, a stage might be reached where secondary slip could occur. If the equivalent yield in tension (Y_t) is greater than the equivalent yield in compression (Y_c), flow in compression will occur, analogous to "barrelling" in a compressive test, and, as a result, the cylinder will shorten. The "tensile length" tends to remain unchanged on its helical path and a helical ridge of compression deformation tends to rise between the turns of the tensile helix allowing them to draw closer together, see Fig.10.(b). Conversely, if Y_t is less than Y_c, tensile stretching may occur against unyielding compression, with the result that the cylinder will lengthen; because plastic contraction occurs in the path of the tensile component. That is, a helical tensile groove (~~analogous to~~ resembling the neck in a tensile test) sinks into the cylinder surface, see Fig.10.(c). In each case the ridges that appear on the specimen surface would be orientated in the same spiral direction.

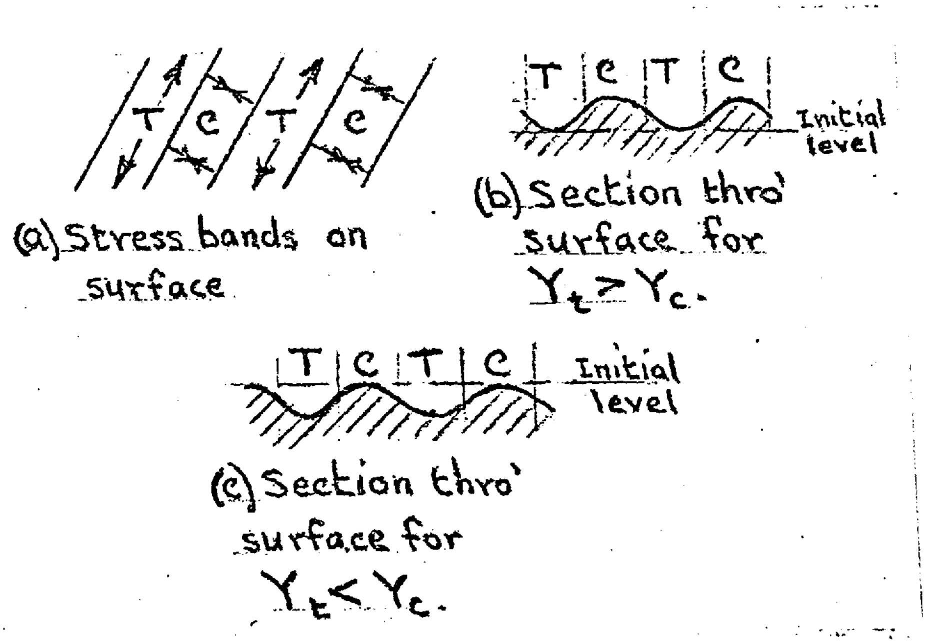


FIG. 10. Possible modes of deformation, in torsional plastic flow, due to direct stress components.

1.2.2. Mechanical Aspects.

Torsion is a relatively easy kind of load to apply, entailing only a simple drive from a standard rotatory power source, such as an electric motor. The drive may readily be transmitted through a standard variable-speed mechanism, such as either a selective gear box, with a number of fixed ratios, or a hydraulic drive with infinite variability of ratio within a given range. Stopping and starting may be done by means of a conventional plate clutch or dog clutch.

Basic machine frame construction can be simple and uncomplicated, because the load is free of undue shock and because stress need not be of high intensity except within the test piece.

Mechanized and automated interlinking of functional sequences is readily possible, with some sacrifice of simplicity.

As shown in Section 1.2.1., test piece shape is simple (Table 1). There are very few problems of insertion and removal of test pieces, provided that this is suitably taken into account in the design of the machine.

Control of vibration should not be difficult.

1.2.3. Measurement of Stress and Strain.

Owing to the uncertain mode of distribution of stress in a cylinder undergoing massive plastic strain (see Section 1.2.1.) it is impossible to measure stress directly in any region of a test piece by any means currently available. The best that can be done is to measure the deforming torque and correlate it to surface strain by a relationship such as $\tau_s = 4T\left(\frac{3 + i + m}{\pi d^3}\right)$, see page 14. Accuracy is then dependent more on accurate knowledge of the values of the appropriate constants than on accuracy of torque measurement. This accurate knowledge can be acquired, on a statistical basis, with increasing experience of torsional straining in conjunction with confirmatory results from experimental work on associated aspects in the field of plastic straining (16,26,27). Thus, as more experimental work is done, more accurate values for constants are obtained and more accurate equations are developed. At present, accuracy within $\pm 2\%$ (12,17,26.) should be possible, provided that torque measurement is itself reasonably accurate.

There should be no problem in making torque measurements accurate to within $\pm 1\%$ and $\pm \frac{1}{2}\%$ should be possible. Given the former accuracy the total error should not exceed the $\pm 3\%$ which is already claimed (12). This accuracy is appreciably better than that possible in other systems of testing of massive plastic flow conditions, since, in all such systems tried so far, strain or stress conditions do not remain constant long enough to obtain accurate values of load for a known cross-sectional area (15,16,28,29).

Provided that the total torque is kept to a low level, relative to the machine stiffness (i.e. provided that unstable elastic deflections in the measuring apparatus are avoided) and provided that the friction in the torque measuring system is maintained at a low level, a simple single-lever measuring system, perhaps acting on a load cell, should be adequate.

Strain measurement is a simpler matter and, at least with respect to the surface flow conditions on a cylinder, can be accurate within the limits of uniformity of behaviour of the material (29). Ideally, a torsional strain indicator (1) actuated directly from a uniformly deforming parallel length of test cylinder would give best results; but such indicators are bulky and likely to modify plastic flow by their presence, particularly if used on a small-diameter specimen. Furthermore, they are likely to interfere seriously with a heating system. Hence, it is usually more desirable to measure plastic flow from the shoulders of the test piece, as discussed in Section 1.2.1. and accept some risk of error. If the testing machine parts are kept relatively stiff and lightly loaded, then, compared with the strain in the test-piece gauge section, any deflection in the machine will be negligible and strain measurement may then be taken directly from the drive spindle rotation (29). That is, drive-rate can be directly convertible into strain rate at the specimen and total rotation can be directly related to total strain in the specimen. Spindle rotations are easy to measure accurately; but spindle speed is more difficult because (i)speed may vary significantly with load, (ii) times for measurement may be very short, and (iii) the times taken for acceleration and deceleration may introduce significant errors. However, care in the design of the machine and suitable choice of equipment should bring these sources of error to a satisfactory minimum.

1.2.4. Thermal Control.

Because torsional plastic straining involves only simple rotational movement of the drive input, and because the straining length remains more or less constant, it should be comparatively cheap and easy to arrange a controllable heating system, of the radiant resistance type, for isothermal testing. On the other hand, if the requirements for low-intensity loading and torsional rigidity (see previous Section) are fulfilled, there will be the relatively large mass of the machine-drive spindle-end and the similar mass of the holding spindle-end present either within the controlled temperature zone or in its close vicinity. It is inevitable that this mass must act as a heat sink, with the result that the heat input must be relatively high to compensate for heat loss and controlled rapid rises and falls in temperature may be difficult to achieve.

The heat-sink effect could be overcome by using either a high-frequency induction or a direct-resistance heating system, but these would inevitably involve greater complexity in design of the equipment and greater cost in the power source. Of the two, the direct resistance heating would be far simpler to apply to suitable materials and lower in cost. In fact, the massive spindles might then be an asset in conducting current to the test piece without much current loss or voltage drop. Current could be introduced by way of a carbon commutator on the drive side and a flexible braided copper connector on the holding spindle side. It would be best to use a.c. because direct-on switching could then be used with less trouble from arcing. A cheap a.c. welding set could provide a suitable low-voltage power source, e.g. a 200A continuous-rating welding generator is capable of heating a 6mm dia. steel cylinder to 1300°C in a draught-free, enclosed, environment with radiation-reflecting walls.

Another alternative would be to use a pair of atomic-hydrogen "flames" (30), or a pair of plasma "streams" (30), see Fig.11.; but it is certain that the capital costs involved in

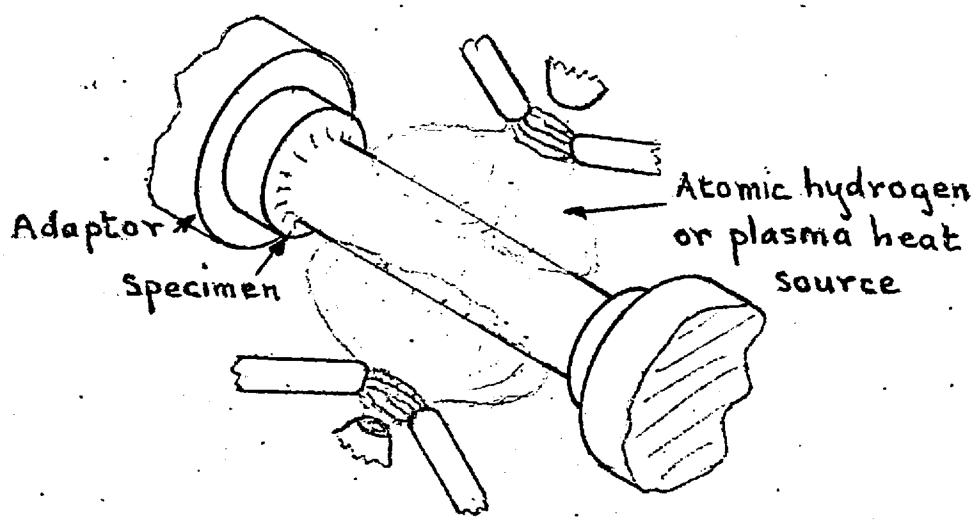


FIG.11. Possible system for heating test specimen to high temperature.

providing these would be too high and it is unlikely that suitable power packs would be available for borrowing. It would be possible, quite cheaply, to use multi-jet oxy-fuel gas flame heating in a similar manner, but oxidation from the flames would present insuperable difficulties with some materials.

Quenching should not be difficult to arrange except that the heat mass, mentioned above, would necessitate that the quenching flow would have to be maintained long enough to prevent reheating of the specimen by the soaking back of heat from the spindles. This problem would not be nearly so great with induction, or direct resistance heating, since the spindle ends need not then be heated to the same extent. Suitable design of equipment should prevent contamination of the machine by water during quenching.

Whichever heating system is used; it should not be difficult to find a suitable, automatic, temperature control system to operate from a thermocouple sited in contact with, or close to, the gauge section of the test specimen. Uniformity of temperature along the specimen gauge length and throughout its thickness would depend on suitable application of the selected energy source in conjunction with suitable control of thermal insulation relative to the main sources of heat loss (notably, conduction loss through the specimen grips).

1.2.5. Power Requirements.

The power requirements for hot torsional straining are likely to be modest as far as mechanical power is concerned, hence the largest machine in Table 1 has a maximum torque of 45Nm and, if the maximum strain rate required is assumed to be 20/sec on a specimen gauge length of about πd, the maximum mechanical power requirement is only 6kW (approx. 8h.p.) for an efficiency of 95%.

Heating power would depend on the temperature required and on the heat losses, say another 5kW altogether, as typically required to heat steel 12mm dia to 1300°C in a radiant resistor tube furnace, although the losses in h.f. induction heating could be much more if the coupling efficiency is low.

Other power requirements would be small compared to these and may be neglected at this stage. Hence, the maximum total power requirements are unlikely to exceed, say, 12kW, whereas a cam plastometer for testing a specimen of comparable size at similar rates of strain is likely to require at least double this power.

If modest demands are made with respect to specimen size, strain-rate range, and maximum temperature, the mechanical and electrical power requirements of a medium-sized plastic torsion testing machine are likely to be within the average range of power ratings of standard items of equipment such as motors, gears, clutches, etc., so the capital costs for such items are likely to be moderate.

2. RATIONALIZED REQUIREMENTS FOR A TORSION TESTING MACHINE.

Before making a decision to develop a plastic torsion testing machine it is necessary to be clear what is required from it in terms of type of service, range of results, adaptability for experiment, and local limitations. From this first rationalization a more precise specification may be prepared.

In the present instance there are four qualifying factors:-

(i) The machine must be capable of use by undergraduates as a means for demonstrating principles etc. (i.e. for teaching purposes).

(ii) The machine must be adaptable to a relatively unpredictable range of differing applications for post-graduate research studies.

(iii) The design must be such that as many parts of the mechanism as possible are relatively cheap, off-peg standards and the remaining parts are of a nature suitable for manufacturing within the Departmental facilities.

(iv) Both the initial capital cost and the average running costs must be low.

2.1. Teaching Purposes.

At present there is no standard of practice for plasticity testing in torsion. As can be seen from Table 1, page 21, there is neither an agreed uniformity in the proportions of, nor a common basic size for test pieces. On the other hand, it seems reasonable to expect to be able to demonstrate most features of behaviour in plastic torsion with lower-melting-temperature materials, such as light metals and alloys, at relatively slightly elevated temperatures on a compromise size and shape of test piece.

Two things would appear to be of over-riding importance (a) ease of use and (b) safety.

2.1.1. Test Piece Proportions.

The use of light metals and alloys as standard test materials does not necessarily imply any notable reduction in the power requirements of a machine relative to a given size of test piece. Indeed it could well be that the torque required to strain, say, a particular aluminium alloy at about 500°C could be greater than the torque required to strain an identical-sized test piece of low-carbon steel at 1200°C, because the strength of the aluminium alloy could be relatively the greater at the lower temperature of working, see Fig.12. (31,32). Hence, the criterion

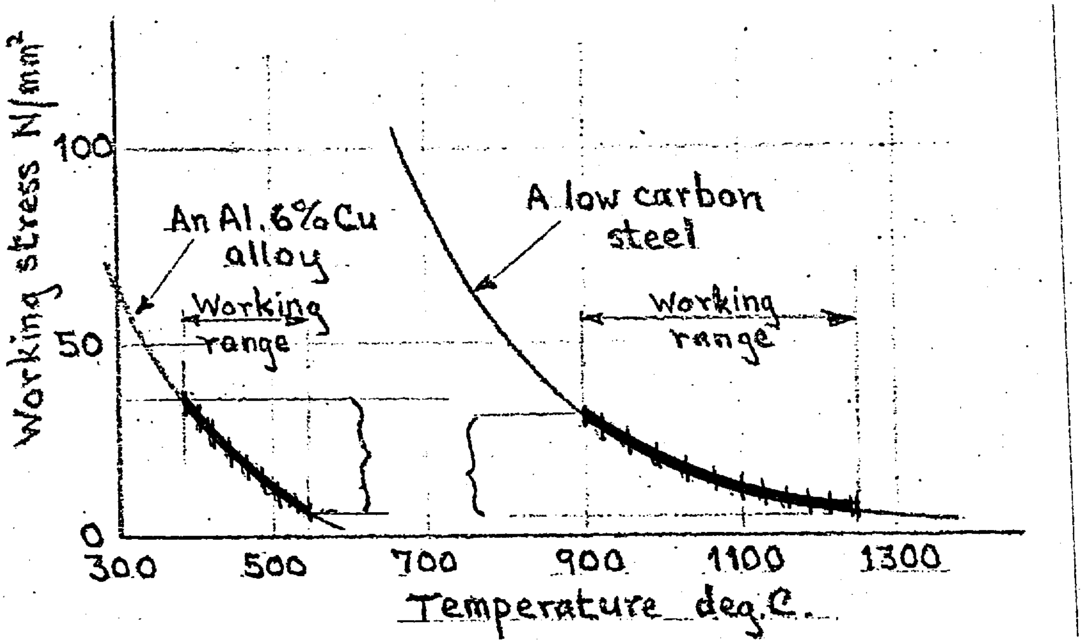


FIG.12. Stresses required for deforming two differing materials in their respective hot working ranges. (Abstracted 31,32.)

of power-requirements depends more on size than material, and equally valid results appear to have been obtained by each of the workers listed in Table 1, page 21, irrespective of the test piece diameter. It would seem that workers using larger-diameter specimens did so principally because they had extra power available, rather than from necessity. On the other hand, the smaller the diameter the more likely there is to be a notable effect from structural differences, such as grain size. A diameter around 6mm seems a reasonable compromise, leaving latitude for increase or decrease if it is considered desirable.

To ensure provision for flexibility it would be preferable to give adaptability for length:diameter ratios over the whole range of Table 1 while choosing a basic ratio of π to simplify strain reading.

2.1.2. Ease of Use.

For laboratory teaching it is essential that the equipment should be fairly simple to use. This requirement evolves partly from the need to control the time taken in setting up for testing and/or changing one specimen for another during testing and partly from the fact that relatively unskilled operators such as undergraduates are likely to be making the tests.

In fact, to achieve this simplicity, provided that reasonable flexibility is still maintained, it is desirable both to limit the complexity of the machine and to make specimen changing simple, even at the sacrifice of some accuracy.

An obvious simplification would be to drop any requirement for reversing the strain direction during a test. This change would have a twofold effect.

(a) It would simplify the whole mechanism, both with respect to mechanical complexity and with respect to skill required in setting up.

(b) Drive to the test piece would be positive and problems of backlash would be eliminated.

There appears to be no pressing need for strain reversing and, indeed, only one of the research projects listed in Table 1 records any results with reversed strain, and aspect (b) would permit test accuracy to be maintained whilst achieving simplicity of test piece design.

A reasonable compromise would be to make the machine basically for unidirectional straining but adaptable, subsequently, to reversing operation by means of some simple modification to the test piece ends.

Setting up is always likely to be somewhat time absorbing, particularly if widely differing test conditions are required from test to test. Even with unidirectional straining, it is likely therefore, that,if differing test conditions are required, the potential effective utilization time for the machine on actual testing would be very low. It seems logical to design the machine to make it adaptable, if desired at a later date, to dual operation; that is, to provide facilities for two test heads which can be operated alternately at will from one drive mechanism.

Load and strain measurement systems should be kept simple and preferably with facilities for autographic recording or data logging when required.

2.1.3. Safety in Laboratory Teaching.

Whenever a teaching demonstration is being given, but particularly in a teaching laboratory, safety is a paramount consideration.

The very nature of practical laboratory teaching lulls operators into a state of mind in which they are not best able to cope with the unexpected and in which they become careless of known risks. Risks that are acceptable when a qualified research worker is doing fundamental or exploratory work, in a relatively leisurely way, become criminal when present in laboratory teaching work. The latter is particularly true in a practical laboratory where the operators are students likely to be relatively unskilled, sometimes irresponsible, and often pressed for time, either because of limited laboratory hours or because they are in a hurry to rush off to some other interest.

Consequently, the maximum safety must be incorporated into a potentially dangerous piece of equipment such as a torsion machine.

The dangers can take several forms such as (i) mechanical, (ii) electrical, (iii) thermal, and (iv) ergonomic.

(i) Mechanical hazards arise whenever there are mechanically moving parts. In a torsion machine the basic motion is rotary but there will be some reciprocating movements associated with specimen insertion and removal, and, possibly, heating and cooling facilities.

Rotary motion is particularly dangerous when there are radial projections on the rotating parts and more especially if there is interlocking of such projections between adjacent rotating parts, e.g. as in gear wheels. In the latter case guards must be fitted. However, even smoothly-contoured rotating parts can cause damage by frictional catching on clothing, hair or loosely flexible material. Guards should be fitted or, if complete guarding is not practical for some reason, the rotating surfaces should be as smooth and frictionless as possible and as little exposed as possible.

Reciprocating motion may be either manually or power driven.

In the former case, provided that the ratio of manual movement to machine movement is kept high (i.e. effective speed of machine movement is kept low) and that the manual movement can only be performed in such a way that the operator's body is clear of any resultant movement of the machine, there need be very little danger. Specimen insertion and removal would be certain to require manual movements; therefore, as far as possible, these should be kept to safe limits.

With powered reciprocatory movement there is always danger whenever movement is rapid and wherever two parts of a machine move towards each other. In the latter case there are two opposite situations; (i) when movement is slow it tends to be forgotten until part of the operator's body is trapped and it may be too late to escape; or (ii) if the velocity of closure is rapid there may be insufficient time to escape. In the present instance it seems unlikely that powered reciprocating movement will be required for anything except, possibly, a quenching operation. Provision must then be made for effective guarding, in conjunction with some form of safety interlock that prevents the dangerous movement taking place until the guard is safely in position.

In general, all moving parts should be guarded as far as possible; but it is likely to be necessary to have ready access to some moving parts for additional instrumentation etc. In the latter case it may not be possible to fit guards; but, if this situation arises, the type of exposed movement must be of a relatively innocuous kind. If it is not, the instrumentation system or other relevant requirements must be changed to make it so.

(ii) Electrical hazards are likely to be present wherever electricity is used but particularly when water is also present. In the present instance electricity is likely to be required for four purposes (a) primary drive, (b) control purposes, (c) heating and (d) instrumentation. Water is likely to be present for quenching and possibly also for functional cooling (e.g. if induction heating is used).

It is highly desirable that water should be eliminated from the locality but this presents problems. Quenching rates must be as high as possible and water is the only convenient, cheap medium. Oil quenching might be used but the cooling capacities of oils and other fluids are limited compared to that of water and, with any recirculating fluid system, suitable cooling of the fluid would have to be provided at quite high extra cost in capital, in floor space, and in mechanical complexity. Taking all the factors into account, water is essential to the plastic torsion testing situation if metallographic study of test pieces is to have reasonable significance. Therefore, the presence of water has to be accepted and every precaution taken to ensure electrical safety. These precautions should take two forms:-

(1) As far as possible all electrical power and control points should be kept as remote as possible from the vicinity of free water such as spray and draining flow areas etc..

(2) When it is impossible to ensure spatial separation electrical insulation must be adequately waterproofed.

The primary drive is likely to be a three-phase motor (see Sections 1.2.2.) therefore the maximum voltage to be considered immediately becomes 415V (i.e. the voltage across two phases). On the other hand it is feasible for the motor to be kept fairly remote from other parts of the machine to which electric power is fed. Only at the control panel stage need single-phase supply (maximum voltage 230V) come near a second phase, and even this contiguity can be confined almost exclusively to the solenoid-operated power switch, if the three-phase supply is kept otherwise entirely separate. Another factor which makes this arrangement desirable is that the local provision for three-phase power supply in the laboratory in which the machine is most likely to be used is a 4-pin, (i.e. three-phase and earth system) without a neutral; hence 230V control power cannot be taken directly from the three-phase supply without running an independent neutral line. It is simpler to run an independent 3-wire single-phase and earth system from an adjacent outlet socket, several of which would be available.

An electrically-operated control system is essential because of its unbeatable versatility, adaptability, facility for interlocking and speed of operation.

As suggested above, a control system would not normally be operated at a potential above 230V. It can be argued that an appreciably lower voltage is desirable for safety, but against this is the risk of accidentally linking auxiliary equipment of 230V potential into the low-voltage line behind the fuse and so damaging the circuit. Most of the likely standard auxiliary equipment (temperature controllers, u.v. recorders, data loggers etc.) is 230V so the sensible decision appears to be to standardize on a uniform 230V circuit for control and instrumentation purposes and then to insist on good mechanical protection and good electrical insulation. The latter aspects should not be difficult to ensure, since a large proportion of standard industrial electrical controls etc., designed and tested for 230V operation, are readily and economically available. It is desirable that as many of the controls as possible should be centrally and accessibly sited, but preferably grouped together on a separate console. Heating is most likely to be electrical and, whatever basic system is used, it is desirable that the primary potential should be 230V. It is essential to ensure that, if a separate electrical power source is used for electrically-powered heating it is taken from the same phase as the control system. If this is not done an undesirably high potential difference of 415V would exist. Normally, single-phasing should not be difficult to arrange, because adjacent single-phase socket outlets in a laboratory must come from one phase; but, if a special supply has to be laid on, then phase mixing could occur if care is not taken to avoid it. Inevitably, the heating centre in torsion testing must be in or near the quenching zone, therefore extreme precautions must be taken to avoid dangers of electric shock arising from the presence of moisture and dampness. It is essential that the electrical heating power should be turned off before quenching and it is preferable that the electrical heat source should be moved out of the way before quenching water is turned on. The quenching water supply should be safely electrically interlocked with the heating power.

Instrumentation is most likely to be electronic in nature and, as mentioned incidentally above, is best kept uniformly suited to 230V single-phase supply for similar reasons to those applying with respect to controls. Standard equipment, of known built-in safety, should be used wherever possible and any special electrically-powered instrumentation must be given at least the same margin of safety.

Appropriate, safe fuses must be incorporated into each circuit.

(iii) <u>Thermal hazards</u> must exist when significantly-elevated temperatures are in use.

Fire is always a possibility and it is obvious that combustible material should be kept away from heated areas; but, apart from this, no more can be done than to take normal fire-safety precautions. The latter should take the form of the provision of a fire extinguisher, suitable for "electrical" fires (a powder or gas type), and a fire blanket or blankets. Of course the machine should not be sited where it creates a possible fire trap.

However, a possibly smaller, but almost certainly more likely, danger is that of skin burns through contact with hot surfaces (or open flames if they are used).

Wherever possible and practical a water-cooling and/or thermal insulation system should be used; but it is impossible to do the present kind of work without exposing hot surfaces at some stage in an operation, and therefore, some risk has to be taken. As far as possible, hot surfaces must be surrounded by a mechanical guard or guards. It is likely that a simple shroud-cover, enveloping the whole of the operating area, and interlocked with the electronic control circuit,will give best results, particularly in conjunction with some degree of automation of the heating-quenching cycle. The shroud should incorporate a fairly generous-sized viewing panel so that every normally-visible happening can be observed.

It is obviously desirable that the lowest suitable maximum test temperature should be used. Probably aluminium and its alloys are sufficiently good for most teaching purposes

(iv) Ergonomic hazards can arise from (a) uncomfortable operating attitudes, (b) badly placed controls, (c) obstructions to free movement and (c) physical overstraining and/or psychologically-induced carelessness.

Operating in an uncomfortable position can lead to rapid fatigue and slow reaction to suddenly-arising danger or sudden change in the state of an experiment. Therefore, it is desirable that the working height of the machine should be the most comfortable both for setting up an experiment and for observation during an experiment. This requires an operating height of about 900mm which is likely to be comfortable for manual manipulation, in a standing position (by an individual of average height), of moderate-sized and light parts such as are likely to be used on a machine of this kind when setting up an experiment. Ideally, a position just below eye level is most comfortable for the visual observation, in a standing position, of a single phenomenon or very small numbers of phenomena, so 900mm would seem to be low for comfortable specimen observation. However, this is not the only point of interest, and a number of closely-timed observations of the specimen, of control settings, of timing sequence etc. are likely to have to be made. Hence, it is reasonable to make the specimen the lower part of the circle that has to be visually scanned and to place important controls behind, close together, and a little above the specimen position, so that they can be scanned quickly without raising the eye level too much, and preferably without having to change the body position (i.e. they should all be within the normal, easy, cone-of-scanning of the average eye). Less critical controls, or controls needing less frequent scanning, should be placed above the others; but, if possible, still within the limits of unstrained upward eye movement. As far as possible, every point to be viewed should be kept nearly equi-distant from the eye position. The latter requirements can be very difficult to fulfil without entailing

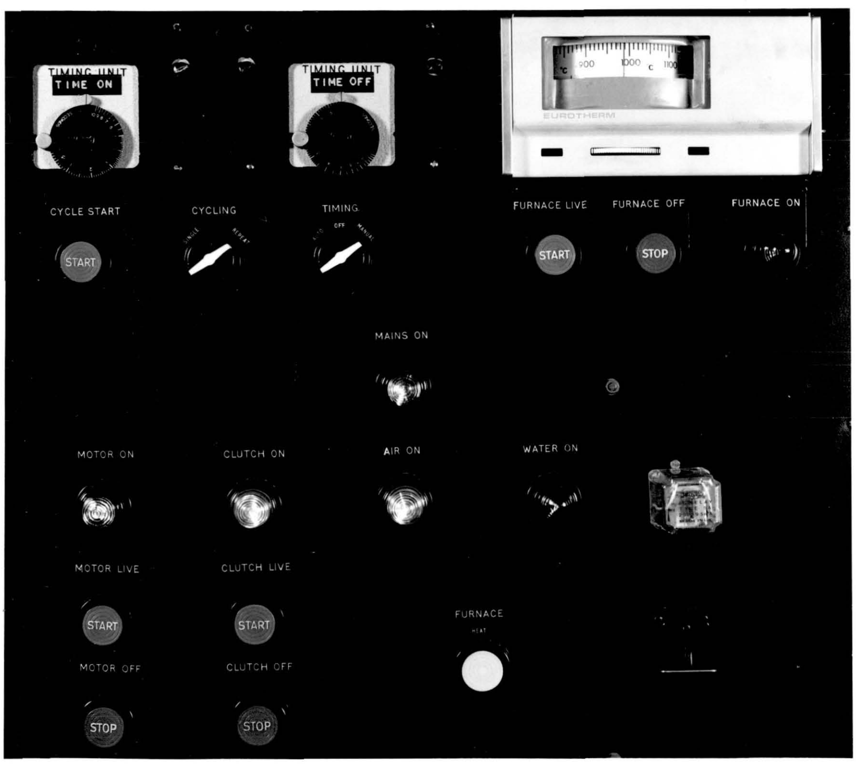


FIG.13. View of main control panel.

considerable complexity in the design shape (and consequent appreciably-increased cost). In the present instance, a reasonable compromise would be to use a vertical, flat, control panel above and behind the working level of the machine, see Fig.13.

The positioning, and nature, of controls are important factors in aiding the rapid, consistent operation of any system requiring external setting and/or adjustment. They are also important factors in ensuring safety. For easy visibility the principal control position is best placed above and behind the test level of the machine, but this recommendation must be considered also with respect to ease, speed and safety of manual access. The overall height of any guard or cover over the test position is unlikely to be greater than about 1050mm and, provided that the outer surface of the guard is kept smooth and free from projections likely to catch on clothing or any parts of the body, it is unlikely to offer any serious impediment to free movement of the hands which are, normally, likely to be held at a comfortable, ready, position about level with and perhaps even resting on the guard itself. There is no need for the controls to be more than about 200mm behind the test piece centre line or more than 200mm above it. Probably only four controls of critical importance to safety in operation are likely to be needed ("on" and "off" for each of motor power and clutch engagement) and these could probably be grouped closely together within an area 300mm square. Thus, after an initial hand and arm movement of about 400mm to the centre position, all critical controls would be within finger and wrist movement distance. The total forward reach over the machine need not be more than 350mm. It is desirable that the purposes and states of each control should be clearly indicated; therefore, each should be distinctively labelled, the buttons should be distinctively coloured and a suitably-coloured warning light signal should be provided on each system close to the buttons; but not so close to each other that respective button positions could be confused with each other, see Fig.14.

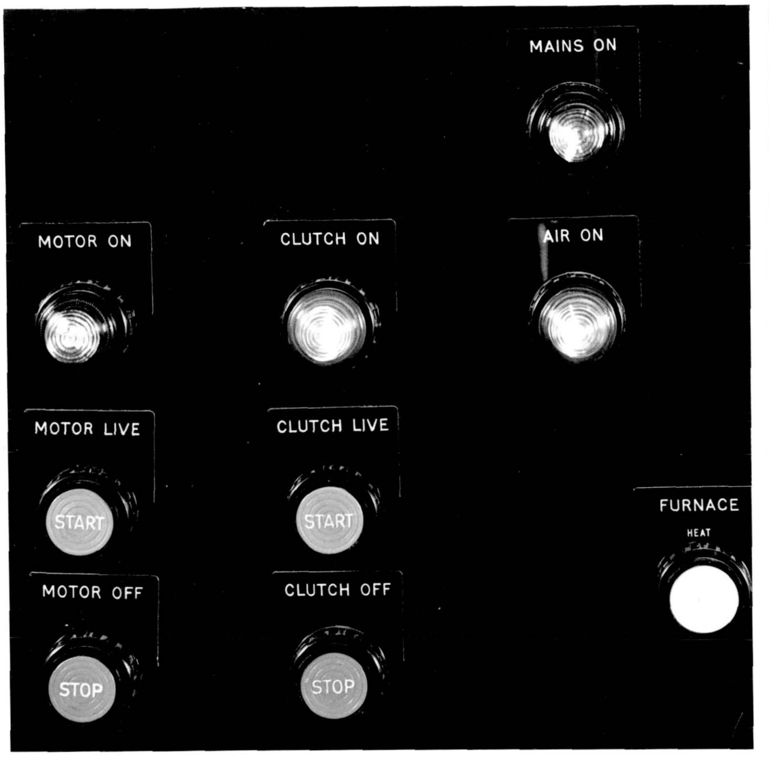


FIG.14. Close-up of view of button operators on main control panel.

There is one other likely control, not required for safe operation during a test, which could give rise to danger when used at the end of a test, namely, the quench-control button. This should be sited so that the act of operating it keeps the operator's hands away from the test position. Simultaneous two-button operation with the buttons at least 500mm apart and below the test position would be ideal; but, in the present instance, a research operator would almost certainly require one hand free for near-simultaneous switching of ancillary equipment, so the compromise is accepted of using one button and positioning it on the front and to the left of the centre of the under-frame cross-bar of the test bed, see Fig.15. It is essential that the quench-operating button circuit should be interlinked with both the heating-power supply and the clutch-power circuit to ensure that the heating circuit is safely cut off before the quenching water flows (the natural delay due to the inertia of the water in the quenching system is probably sufficient to give this condition

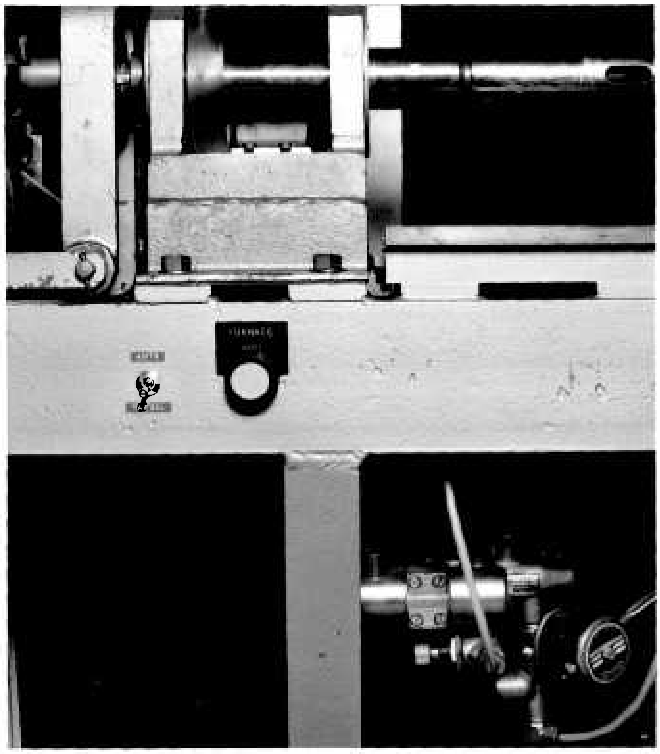


FIG.15. "Quench" control button operator on front of test bed frame below dog clutch.

without special precaution; but this should be verified and, if doubt exists, other precautions should be taken) and that the clutch is disengaged before quenching water flows. If the latter is not ensured there is danger to torque-measuring equipment and other parts of the system due to the sudden rise in torque likely to occur with falling temperature.

Obstruction to safe, free, movement of the operator is always a hazard with any power equipment. It is mentioned above with respect to the ease of reaching operating controls; but there should be concern also with movements of the operator when setting-up and when performing ancillary operations during tests. To keep easy movement possible the front face of the machine should be free from unnecessary projections, particularly of sharp or long projections, and foot movement should be unrestricted within the working area. To give the latter conditions the machine front line should run away from the operator as it falls to knee level and should run farther still towards foot level. A steady slope of the machine stand from the bedplate level back to about 150mm from the front face should be sufficient, see Fig.16. Although all main control positions

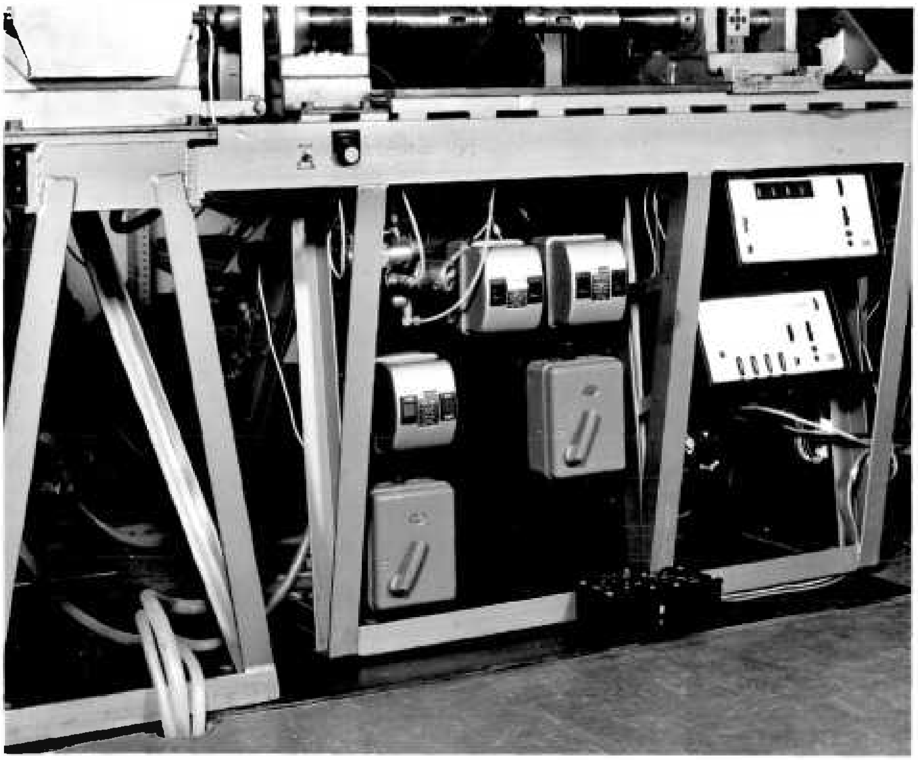


FIG.16. Clearance slope at front of machine with service and servo-control panel set back in frame.

should be within easy hand reach, without need for either strain or for excessive movement, it is permissible that the primary service controls should be rather less accessible and they should certainly be in a position in which they do not cause obstruction. Below the test-bedplate and set back under the machine gives quite good unobtrusive positioning for the latter, see Fig.16.

The general impression should inspire confidence in the operator without overawing him. Thus, a stationary machine should look rigid and stable, but not massive or overwhelming. Clean, light, colouring should be used wherever possible to create an air of cleanliness and cheerfulness. Economy prevents any extravagance in this present connection, but, even with limited money and restricted manufacturing facilities, it should be possible to create a reasonably good impression, particularly since the machine need not have a great maximum height and can be kept within fairly restricted limits of ground area. In this case, it is possible (and desirable for other reasons) to use rigid-looking members and overall proportions that make the machine look stable and reliable. Normal industrial paints in conjunction with tidy, bright, controls and a little chromium plating can be used to give quite a good impression.

2.2. Adaptability for Research Purposes.

In general, it is impossible to predict what might be required by future workers in new, or developing, associated fields. However, there are certain aspects for which flexibility for adaptation seems desirable.

These aspects include (i) tests on differing, or new, or unusual materials, (ii) variation in strain rates and /or strain durations, (iii) accommodation of differing load ranges and load variations, (iv) control of end-load measurement and end-freedom, and (v) provision for reversed straining.

2.2.1. Differing Materials.

It is not necessary to go to new or unusual materials to find widely divergent conditions for conventional testing. Aluminium alloys and steels, for example give contrasting requirements. Aluminium and its alloys are commonly hot-worked at temperatures in the region of 500°C, whereas steels may be similarly wrought at about 950-1200°C. In the case of hot-working of aluminium, surface oxidation is unlikely to cause much difficulty, but steels are very prone to scaling in their hot-working ranges. Thus, steels are likely to require both relatively high temperatures and simultaneous protection from the atmosphere, neither of which is easy to achieve. In fact, 1200°C must be near to the maximum temperature attainable by conventional means in the circumstances appropriate to testing in plastic torsion (see Section 1.2.4.). Any temperatures above this would require direct electric-resistance heating, electric induction heating or one of the special systems mentioned in Section 1.2.4. Some of the latter are unsuited to the provision of a protective atmosphere, in particular, the cheapest system, the oxy-fuel gas flame. Most high-temperature heating systems present difficulties with respect both to accurate control of temperature and to control of uniformity of temperature.

Altogether, the regular provision of relatively high temperatures, combined with effective protection, would be very costly, probably more than doubling the total cost of the equipment, and cost is important, see Section 2.4.

On the whole, since the immediate emphasis is on routine teaching experiments, it seems reasonable to limit the primary heating provision to a simple, cheap, system, say radiant-resistance heating, by means of a tube furnace, keeping the normal maximum temperature level down to 600°C with provision for some work at higher temperatures in conditions of relatively less efficient heating. However, the machine layout and components should be designed to permit the subsequent installation of more sophisticated heating, if required, and if money and/or equipment becomes available. Specimen adaptors etc. should be made as heat-resistant as possible right from the start.

In the latter context, because no machinable engineering material of sufficient strength and toughness is available for use at temperatures much above 1000°C, the heating system chosen specifically for high temperature work must be one which does not greatly elevate the temperature of the specimen grips or adaptors, i.e. the heating must be suitably concentrated on the specimen.

There are definite disadvantages in using a tubular, radiant, electric-resistance furnace:-

(i) It is bound to impart a large proportion of its heat to the specimen adaptors and machine spindles to overcome the temperature drops at the specimen ends.

(ii) Heating must be by "soaking-up" to a controlled furnace temperature, which entails appreciable delay in attaining test temperature, due to the presence of the heat sink effect of the specimen adaptors etc.

(iii) Quenching requires the removal of the furnace before quenching can begin.

(iv) Quenching must be prolonged to cool the large mass of the adaptors etc. before specimen handling becomes possible.

(v) Observation of the specimen during testing is difficult to arrange.

Taking all these points together, a radiant furnace is not particularly desirable; but its cost is lower than that of any other suitable system by a margin of about 1:50. It is also reasonably suited to the setting up of a controlled atmosphere.

On balance then, a suitably-proportioned electric, radiant-resistant furnace, say Kanthal-wound with a temperature range potentially up to 1200°C, would seem to give some flexibility with respect to testing materials with quite wide testing temperature differences.

Varying plastic strengths between different materials and/or differing test conditions need not cause great difficulty in normal circumstances, because the diameters of individual specimens may be adjusted within quite wide limits. Only with materials of abnormally high plastic strength, or when small-diameter specimens are unsuitable, would greater torque be needed. Greater torques could be achieved, however, by accepting a lower range of strain rates and installing say a 2:1 in-line drive-speed reduction-gear, when required, provided that the machine components are made of adequate strength in the first instance and that provision is made for measuring the increased torque (i.e. provision should be made for applying and measuring torques of up to 46Nm).

2.2.2. Variation in Strain Rate and Strain Duration Times.

According to Table 1, page 21, the maximum strain rate used in research varies up to 180/sec. with a 1:1 $\frac{l}{d}$ specimen, but most workers seem to set a limit round a maximum of 20/sec. A reasonable compromise would be to set a maximum of, say, about 50/sec. with a 1:1 gauge ratio or 16/sec. with a π:1 ratio at a maximum torque of about 23Nm which seems about average for Table 1. On the other hand if, say, a rate of 100/sec. is required it can be obtained from the 50/sec. system by installing a 1:2 in-line drive speed multiplier-gear and accepting a reduced maximum torque of 11.5Nm.

Variation of strain rate must be possible over the available range, at least in fairly small steps, but preferably with infinite variability. For research work a hydraulic, variable-speed reduction-gear seems a simple, obvious, choice presenting no problems.

In almost any research project strain duration time could take two forms: (a) a single "burst" of strain over a given time, or over a given total amount of strain, or to fracture, (b) repeated "bursts" of strain of controlled duration, or specific total amount, alternating with a controlled time interval, i.e. cyclic straining. Start and cut-off would have to be sharp, particularly with cyclic straining, to give the necessary selectivity with short times and small bursts of strain. This condition can be met only by an electro-magnetic clutch system, possibly in conjunction with an electrically-operated brake, although the latter seems, at first sight, likely to be an unnecessary refinement in this application. Electronic timing with wide flexibility of adjustment is essential, but there is a snag in that, if wide time adjustment is provided, the accuracy of short time setting is poor. A fair compromise would require the use of electronic timing devices (one would be needed for strain duration and one for time-off interval) with ranges chosen for what seems likely to be required in the first type of use; to be readily replacable, if necessary, or to have part of their interior readily changeable to give other desired ranges of control. The complete timing circuit would have to be easily adaptable to single cycle, multiple cycle, and manual control, to suit it both for setting-up and for test running, as required.

At high rates of strain, precise control of total strain may be difficult to obtain on a time basis. In this event it would seem that an electronic signal taken directly from the strain-measuring device would be required. This involves a significant increase in complexity of circuitry, but it should be quite feasible if a batch-control type cycle recorder is used for strain measurement and the batch-complete signal switch preset, is linked into the strain cut-off side of the strain-duration timer, which would have to be shorted out when the batch-timer is in use. The incorporation of a batch-timing circuit raises the cost of a cycle recorder quite significantly; so, in the present instance, it may be necessary to omit this particular facility.

Strain must be measured as accurately as possible and strain rate should be accurately self-indicating, if possible. An electronic system is the only sufficiently-rapid, sufficiently-accurate type, if a suitable source can be arranged. In the present case the machine is likely to be designed to be adaptable to two torsion heads operating, one at a time, from one common power source, see Section 2.1.2; therefore, the power source could, logically and conveniently, lie at right angles to the two heads in a horizontal T-configuration, with the power source as the stem. In this situation it is easy to extend the main drive spindle outside the machine and to provide a convenient signal-generating system to feed both an indicating strain-counting instrument and a strain-speed analysing and indicating instrument. (It is unlikely to be possible to get one, off-the-peg, instrument combining both these functions). This part of the system is so important, with respect both to speed of indication and to accuracy, that a mechanical system would be practically useless; therefore, electronic instrumentation has to be used at a considerable outlay in capital (about £500 seems likely after a preliminary look at what is available).

2.2.3. Accuracy of Torsional Load Measurement and Variability in Load Range.

In making any series of tests in plastic torsion the torsion load, generated in the tests, is a variable of primary importance. The load is bound to vary to some extent even during the performance of a single test, if only to rise from zero to its appropriate maximum and then to fall again to zero. Further variations must occur with imposed change of straining rate, with changes in the structure of a material occurring during testing, with different test materials, with change of testing temperature and with change of specimen diameter. Many of these changes occur between different tests but variations occurring actually during a test may be of critical importance in some types of research. Therefore, the measuring system must have not only static accuracy but dynamic accuracy as well. That is, it must be "hard", responding instantaneously and precisely to rapid changes, without extraneous oscillation.

There is only one reasonable solution to this kind of problem, namely, the use of a load cell of some type in conjunction with a torque lever-arm, the latter possessing an inherent rigidity to match the "stiffness" of the load cell. As a result of this latter requirement the lever, even when suitably shaped is likely to be relatively heavy, since a normal light-metal, with its low modulus of elasticity, would give a load deflection appreciably and undesirably greater than would a similar steel lever of equal total strength. However, this relatively greater mass should not give rise to any serious problems of inertial behaviour, because lever movement would be minimal with a "hard" load cell. At the worst, load cell deflection is not likely to exceed, say, 1·5mm at the end of a lever of appreciable length.

Many load cells are available with linear responses accurate to $\pm 1\%$, or better, over their recommended load range and this accuracy is likely to be quite good enough for the present application. Load cells using resistance strain gauges, sensing from thin-walled load cylinders or deflecting load beams, seem likely to be the most suitable types, particularly as instrumentation is easy to arrange with the aid of standard laboratory equipment. Deflecting-beam load cell systems are suited, more particularly, to very low load ranges and are the type most likely to have a large deflection such as that mentioned above.

If accuracy of torque measurement is to be maintained over a wide variation in total load, it is certain that more than one load range is desirable;so that, as far as possible, normal maximum loads are kept to about the third quarter of the range, to allow for the odd unexpected peak load and yet to keep measurement accuracy close enough for research purposes, say $\pm 2\%$ of the measured load. Some variation can be accommodated by adapting the specimen diameter, once the approximate strength of a material is known from preliminary trial runs in appropriately similar test conditions; but this is not a very desirable method. The choice of load ranges is important,because, although it might quite rightly be said that load cells could be interchanged (and indeed provision must be made for this), it is undesirable that changes should have to be made and recalibration done too frequently.

Taking into account all these factors, and keeping economy in mind, it seems reasonable to make provision for two load ranges covering 0-23Nm and 0-11.5Nm respectively but making the 0-23 adaptable for some testing at loads up to 46Nm, see Section 2.2.1.

2.2.4. End-Load Measurement and Freedom of End Movement.

Ideally, there should be little or no change in the length of a cylinder of material undergoing coaxial plastic torsion; but a change is frequently seen to occur (12). In some cases the cylinder shortens and in others it lengthens. Little appears to have been done to study the cause of the change and the question arises as to whether or not, in the present instance, it is possible and practicable to provide facilities.

Quite how these effects could be studied in a plastic torsion test, other than by microscopical examination and measurement of the profiles of sections taken normal to the spiral ridges, does not seem very clear at this stage, since anything done to measure the relative magnitude of the total axial forces must inevitably change their distribution. A series of tests with successively increased intensities of axial restraint, applied in the appropriate direction, seems one possibility, but the restraint would have to be "floating". That is having applied a particular axial force the end-restraining mechanism would have to"give" at the preset force, as the length of the specimen changes. Pneumatic and/or hydraulic application is a fairly simple possibility, but even this involves a very significant amount of extra complexity and expense, which would be difficult to justify at this stage. On the other hand, if the end of the test bedplate is made of adequate length and left free of attachments, there should be little difficulty in adding this facility later.

Taken all round, it seems inappropriate to provide floating end restraint, but the end of the torsion-measuring shaft should be provided with a suitable fastening, the bedplate should be given some increased length, and a load cell and anchor bracket should be provided so that rigid end-restraint, with or without full end-load measurement, becomes possible.

Associated both with end-load measurement and end-restraint and with end-freedom is the need for the torsional-load measuring system to be as free as possible to move axially, without either, (i) imposing uncontrolled end-restraint on a specimen, (ii) greatly modifying end-load measurement accuracy, or (iii) modifying torsional-load measuring accuracy. That is, the torque-measuring head must be made fully floating, with as little friction as possible, yet it must be kept as accurately in linear alignment as possible and as rigidly torsionally stable as possible.

2.2.5. <u>Provision for Reversal of Straining.</u>

It is difficult to provide facilities for reversal of straining during a test, not because of any mechanical difficulty in causing a reversal in the mechanism but because of the time factor involved in making the change. Essentially, change of direction of straining should be as nearly instantaneous as possible and the only way in which this can be ensured, from any conventional form of primary drive system, is to provide two contra-rotating primary drives and then to engage these as required, by using electronically-timed electro-magnetic clutch switching. A two-input single-output gear-train, free from both backlash and large inertial effects, is essential.

In spite of a likely use for a reverse-straining facility, it would seem that cost would prohibit its provision at this stage; but, since it looks as if a basic T-configuration layout will be provided, so that two heads could eventually be operated from one power source, see Section 2.1.2, it is readily possible, as an alternative, to change the purpose of the drive provided for the second head and to make it a feed-in for a second primary drive mechanism, complete with motor, gear and magnetic clutch. It is apparent that, if this is done, the electric circuitry would have to be adapted to the changed purpose, the specimen end-design would have to be made suitable for reversed loading, and the torque-load-measuring system would have to be made two-way. Of these only the last is of any particular difficulty (It would in fact rule out the use of a deflecting-beam load cell, since that type is not easily made for two-way operation).

2.3. Manufacturing Limitations.

Owing both to administrative convenience and cost limitations, see Section 2.4, the machine would have to be manufactured as far as possible within the facilities of the Department of Metallurgy. This requirement places certain limitations on construction within which a practical design has to be restricted. Such limitations should not, of themselves, greatly restrict any of the prospective functions of a machine, but would require the adaptation of the design with respect to the manner in which each function is attained.

The limitations would apply particularly in relation to the three factors:- (i) handling and siting within the Department, (ii) available machining and fitting facilities, and (iii) maintenance.

2.3.1. Siting and Handling within the Department.

Within the confines of the Department there are at least four possible laboratory sites on which the testing machine might be located, so that it would be within easy reach of the essential range of services, including (a) electric power (three phase and single phase a.c.), (b) cooling-water supply, (c) drainage, and (d) compressed air. Unfortunately, not only are these sites in different laboratories on different floor levels, none of which is on the same floor level as the machine shop, but at this stage, it is impossible to say which one might be used, or even if the site, once chosen, would be permanent. Therefore, the construction of the machine must be such that it can readily be transported from the place of manufacture (the machine shop in this case) to laboratory site and from site to site. That is, it must be made in sections, each of a size and weight suitable for movement up or down stairways and through doors some of which are limited to 920mm wide.

Certain likely routes would require manual handling in restricted spaces (notably, bends in a staircase); consequently, the weight of each individual part should not exceed, say, 100kg, which would also suit handling during manufacture, see Section 2.3.2.

A further limitation is that two of the sites have a floor weight restriction of 200lb/ft , but it is not visualized that this will present any difficulty, other than the provision of a "spreader" plate, or plates, under the feet.

Sectionalization need not be too disadvantageous, since, although it can add appreciably to the amount of machining, it may make other machining operations simpler, see Section 2.3.2.

2.3.2. <u>Manufacturing Facilities</u>.

Metallurgy Department's machine shop is well equipped with precision lathes, universal milling machines, surface grinding machines, a cylindrical grinding machine, a shaper, a power guillotine, a folding machine, welding facilities, including manual metal-arc, gas-shielded argon-arc and oxyacetylene processes, and the usual auxiliary equipment. The standard of craftsmanship is high and a variety of special skills are available (e.g. instrument-making, die-making, precision grinding and high grade welding). However, the main emphasis is on very light engineering, rather than on general engineering, so there are definite limitations both on the bulk of unit part that can be accommodated in the available machines and on the extent of the surface area that can be machined on any one part at one setting.

These limitations make it necessary to adopt modes of construction, for rather larger articles such as the present one, that do not permit either the use of an ideal design or the easy control of overall precision.

On the other hand, past experience, with constructions correctly designed to suit these concepts, has proved that the interest taken by the machine shop technicians in overcoming the challenge, presented by such a type of construction, has resulted in satisfactory products of a very high standard of finish and performance.

Provided that: (i) design details are properly controlled and (ii) suitable standard components, available at reasonable cost, are used wherever possible, there is no reason why an effectively useful and satisfactory hot torsion testing machine should not be produced in the machine shop without taking an excessively long time or at an unreasonably high cost.

2.3.3. Maintenance.

A very important aspect of any equipment is maintenance, which includes maintenance of operational accuracy as well as replacement of damaged parts.

In the present case, most of the individual parts are within the maintenance scope of the machine shop for making replacements, but many of the parts are likely to involve far more machine shop time and delay in individual replacement than is desirable . Preferably, parts should be so designed and constructed that rapid wear and risk of frequent damage is confined to details that can be replaced or repaired on site by laboratory technicians with their more limited facilities and without undue delay.

When making hot torsion tests the most likely zone of wear and damage in a torsion testing machine is at, or near, the actual test specimen where parts are subject not only to frequent, occasionally rough, handling, during setting up and dismantling, but must also go through frequent heating and cooling cycles, whilst subjected to stress and risk of spontaneous local welding between torsion drive-shaft and anchor-shaft ends, adaptors and specimen ends.

Logically, as many parts as possible should be of a rugged, firmly-supported, design made of high-quality material so that they are unlikely to fail or wear excessively for a very long time. This approach is neither so costly nor so wasteful as it might seem at first sight, because of the needs for minimizing undesirable torsional and lateral deflections and for eliminating undesirable vibrational modes (see Section 4), which also require fairly massive construction. Thus, several aims can be achieved simultaneously on these lines.

With respect to the critical zone, mentioned above, the common-sense approach is to break down the local construction into a number of relatively small component parts, easily separable from each other and easily replaced by suitable similar parts, perhaps kept in stock for the purpose, when wear or other damage occurs. Of course, when a potentially unit part is broken down into component parts in this way, there is grave risk of loss of accuracy from cumulative errors from the joints; therefore, great care is required in the design of these joints. On the other hand, there is greater facility for adapting the material of construction of each component part to suit the particular rigours of its service.

2.4. Cost Limitations.

Almost every research or development project in a university department suffers from a restriction of money. This is particularly true of a non-standard project, such as the present one, for which there is no specific capital allowance. The main problem is, usually, to get sufficient capital to provide the basic equipment, since running costs may often be quite readily provided, particularly if they are likely to be relatively small as they should be for a hot torsion machine.

Unfortunately, the capital costs involved in designing and developing a custom-built machine of this kind are high. A conservative estimate of the total cost for an outside body to design and make this machine is £8,000. This sum is out of the question; but £1000 is available which has to cover the purchase of any standard off-the-shelf components, any special instrumentation and any materials of construction. Departmental machine shop time, provided it does not interfere with the normal running of the Department, is not chargeable and, of course, there are no overhead charges. (It is only fair to say that the likely cost in technician time will probably be about £2000).

2.4.1. Purchase of Off-The-Shelf Component Equipment.

Certain components of equipment, which must be built into a machine of this kind,are much less costly to buy as standards than they are to make independently. In addition, the standard item, because of more experienced manufacture, is likely to give more efficient and reliable service than an individually made-up item. Included in these components are certain obvious ones,such as the electrically-powered drive motor, the hydraulic variable-speed gear, the magnetic clutch, any ball or roller bearings, pneumatic equipment and fittings, hydraulic valves and fittings, electrical switchgear, fuses, relays etc.

In some respects the outlay on these must be conditioned by the available cash. Thus, if a large proportion of the capital must be devoted to special instrumentation, see Section 2.4.2, and some to materials of construction, see Section 2.4.3, the expenditure on components must be confined to the balance, which may entail limiting not only the capacity of the machine with respect to power etc. but also the flexibility that is provided (i.e. by limiting the complexity). As a first guess it would seem that not more than £350, see Sections 2.4.2. and 2.4.3, will be available; but, within that limit, it should be possible to provide nearly the capacity and most of the facilities commended in Sections 2.2. and 2.3. In fact, the knowledge of this limit controls some of the more arbitrary decisions already outlined in those Sections.

2.4.2. Special Instrumentation.

Many types of measuring, recording and controlling instruments are available in a metallurgy department for mutual sharing between different research projects and teaching experiments. An obvious saving can be made in capital outlay by making use of such of these as are available so that they can be plugged in or linked up, as and when required. This is not a particularly hampering decision to make, since a machine of this kind is extremely unlikely to be operated for actual test purposes for more than an average of 10% of the normal College hours. That is, instrumentation is unlikely to be required for more than this time, therefore it is not justifiable to use capital exclusively for this, in spite of some inconvenience.

One exception to this might be temperature control, because heating times may well be far greater than test times, both because of possible delay in attaining temperature and because much trial and error may be involved in determining effective heating conditions. As a consequence, it is advisable to purchase at least one suitable temperature indicator-controller, costing say £50.

As mentioned in Section 2.2.2. an electronic revolution counter and a speed indicating analyser are each essential, but are not normal parts of metallurgy equipment, so these will have to be purchased for this application.

2.4.3. Materials of Construction.

Most of the parts to be manufactured in the Department are likely to be made from mild or low-alloy steel bars or sections which are, almost certainly, likely to be available at wholesale cost from the Departmental stock maintained in the machine shop. In addition, bolts, nuts, etc. may also be drawn from the same source at suitable cost. In these circumstances the total cost of materials for such parts is not expected to exceed £40.

However, heat-resisting material will be required for test piece adaptors, connecting sleeves and other elevated temperature fittings. The cost of this material can only be notionally estimated before critical sizes etc. are known, but £50 seems a reasonable allocation if some allowance is made for a few spare replacement adaptors and sleeves.

3. SPECIFICATION OF OUTLINE DESIGN.

Having discussed the pros and cons of the various possibilities for a hot torsion testing machine, including availability of capital, it is now possible to summarize the basic requirements on which an effective design must be based.

This summary must include the basic function, strain measurement system, torque measurement system, end-load measurement system, heating and cooling needs, safety aspects, degree of automation and general layout. The main general requirements are listed in Table 7. More specific comments on, and factors determining, the final specifications are considered in the remaining parts of this main section. Related details of the final design are then considered in the next main section.

TABLE 7.

Summary of main features of design specification.

Function	Limits	Means
Strain rate range	Continuously variable from 1 rev/min up to 970 rev/min	Hydraulic gear
Power	Max. torque 22.3Nm at 970 rev/min 2.25kW (3 h.p.)	Electric Motor
Straining control	Preset single straining times and up to 10 sec. on and 10 sec. off automic cycling	Electronic via magnetic plate clutch
Strain measurement	1/100 rev. upward	Electronic
Strain rate measurement	1/10 rev. sampling	Electronic
Torque measurement	0-11.5Nm and 0-22.3Nm occasional 0-46Nm	Single lever on electronic load cell
End load measurement	0-500N either way	Electronic load cell
Heating	Temperatures up to 600°C normal, 1200°C occasional	Electric resistance radiant tube
Heat control	0-1250°C	Pt-Pt.13% Rh thermocouple electronic control
Cooling	Max. rate practicable	Water spray
Reverse straining	No provision	-
Dual operation	Possibility only	Dual drive output

3.1. Basic Functions.

Included in basic functions, apart from the mode of applying the actual torsion to a specimen, are included (i) speed range and control, (ii) drive power source, and (iii) control of test cycle.

3.1.1. Speed range and control.

The basic range of strain rates (up to 20/sec), recommended in Section 2.2.2., amounts to a maximum speed of rotation, at the end of the specimen, of about 1000 rev/min, with the possibility of progressive reduction to a minimum speed of, say, 50 rev/min, without having to change the mechanical set-up of the machine. Those requirements can be almost perfectly satisfied by means of a stock-size Carter, hydraulic, variable-speed, gear box (of either the "F" type or the "M" type mentioned below), provided that a slight reduction is made in the required maximum drive speed from 1000 down to 970 rev/min. There is no firm reason against this reduction so it may be accepted.

The type "F" box uses a flange-coupled, direct-drive, electric motor and the "M" type uses a top-mounted electric motor, acting through a belt drive. Although mechanically less efficient and liable to transmit belt "thump", the latter requires a little less floor space, is less liable to transmit vibration from the motor, and is a little cheaper in first cost.

It is proposed that the whole design should be based on a Carter, Type "M", gear box fitted with a standard speed-indicating control knob; the actual size of box to be determined on the power requirements outlined in Section 3.1.2, correlated with acceptable cost.

3.1.2. Power Requirements.

If a Carter Type M14 gear box is used, a motor rating of 2¼ kW (3 h.p.) will give a maximum torque output of

22.3.Nm at speeds from 30 rev/min up to 970 rev/min. This torque output is a little lower than that recommended in Section 2.2.2. (23.Nm); but it is thought right to make use of this size of box, since, to use the next size up would raise the initial overall cost by the significant amount of 10%, after allowing for the cost of appropriate associated changes in the drive train, switching etc.

On its output shaft this drive should be equipped with a Croft, Type F.E.B, electro-magnetically operated, multi-plate clutch.

The electric motor should be a standard 3-phase a.c. induction type, which, at relatively low cost, will give an efficient drive at reasonably constant, reproducible, speed. The cost of a motor and associated equipment for more precise control of drive speed would be prohibitive in this application.

As the motor need never be started under load and as its power rating is quite low, direct-on starting, with cheap and simple switch-gear is acceptable; but a suitably-delayed overload cut-out should be included in the switchgear to reduce the risk of accidental damage to either motor or gear box. This degree of protection should be necessary only when the gear output is set for a high speed and either starting on load is attempted, or an unexpected on-load rise in torque demand occurs. At low speed settings, accidental starting on load, or unexpected rise in torque demand, should cause disconnection of the drive, say by failure of a safety shearing pin, before motor-load becomes excessive.

3.1.3. Control of Test Cycle.

To ensure reproducibility of test conditions, the control of the test cycle must be automated in a reasonably flexible way, as was discussed in Section 2.2.2. A load cycle is likely to be initiated by the manually controlled closing of an electric switch,acting on the electro-magnetically-operated clutch, whilst the drive motor is running and is likely to be most effectively completed by preset,automatic,timed opening of the same switch. It must be possible to vary the timing of these operations in a systematic manner and provision must also be made for repeated cycles with a systematically adjustable time interval between them. It must also be possible to override the automatic timing by manual control. Two standard Venner electronic timing units, each suitable for time settings up to 10 seconds, can provide these facilities when used in conjunction with a standard system of electronic control-button-operators, interlocks, switches etc. (such as those made by Allen West), interlocked,as far as possible,to give the degree of safe operation specified in Section 3.6. The first choice,of 10-seconds-range time controllers,is made on a purely arbitrary basis and provision must be made for easy changing of the maximum ranges of the timers.

It must be possible at any instant to see,at a glance,the state of the set-up and the stage of a cycle.

3.2. Strain Measurement.

There are two strain measurements to be made: (i) total torsional strain and (ii) speed of torsional straining Both are to be taken from a suitable extension of the primary drive shaft, as suggested in Section 2.2.2, for which an appropriate, toothed-type,pulse signal generator is to be devised.

3.2.1. Measurement of Total Strain.

Several batch-type,electronic,pulse-counting devices are available from different makers,none of which appears to have any particular disadvantages or merits compared with the others. The Smiths Industrial Ltd.,electronic,single-programme, 4-digit,batch counter, with 6-digit total indicator, is chosen because it is slightly cheaper than most of its competitors and because it matches with the same firm's rotational speed indicator, see below.

The counter needs a pulse-type signal generator, which is required to indicate to at least $\frac{1}{10}$th of a revolution of the drive shaft. The Smiths Industrial Ltd.,"Miniature", magnetic perception head is to be used in this connection, since it is designed for use with this equipment and can readily be actuated from a suitable pulse generator.

3.2.2. Measurement of Speed of Rotation.

At the time of specification, only one, electronic, rate-indicating instrument is readily available on the market, namely the Smiths Industrial 4-digit display, electronic, rotation-speed indicator, operating from a pulse sensor similar to that quoted for the total strain indicator. A separate sensor is needed for the speed indicator, but it may be actuated simultaneously with the total-strain sensor, from the same toothed signal-generator; thus, only one signal-generator is needed for both sensors.

The signal generator, which takes the form of a toothed, ferro-magnetic, disc, should have a sufficient number of teeth to allow indication to $\frac{1}{10}$th of a revolution of the drive shaft. This means that for each tenth of a revolution the disc must give as many pulses as possible for sampling and integration by the speed counter. Space does not permit the use of a very great number of teeth on the generator; probably, about 60 can be accommodated, but this should be sufficient.

(Note:- Subsequent to placing the order for this equipment and beginning manufacture, other instruments, which might have been more suitable, came on the market too late for consideration, e.g. the "Orbital" indicating system with a phased pole, rotary, pulse generator, which can measure accurately to 1/10 rev and indicates speed by measuring the phasing sequence relative to mains frequency).

3.3. Load Measurement.

It is desired to be able to measure two types of load: (i) torsional load and (ii) experimental end-loads.

3.3.1. Measurement of Torsional Load.

Torque is to be measured in two load ranges: (a) up to 11.5Nm maximum and (b) up to 22.3Nm (nominally 23Nm) normal maximum, with the possibility of use up to 46Nm occasional maximum, by means of a single-lever system with two lever-ratios, derived from two suitable attachment pivots on the one lever arm. The lever-ratios should be at about 100mm and 250mm radius, respectively, from the centre of moment, to give (i) simple conversion from indicated force to torque and (ii) sufficient, but not excessive, clearance for incorporating the two load cells.

Low-range torque measurement is to be done with a Coutant, deflecting-beam, load-cell, Type 128, suitable for loads up to 45N, acting at about 250mm radius, bearing in mind that, if reversed torque is ever to be applied, this cell would be unsuitable.

Maximum-range torque measurement is to be done with a Coutant, reversible, direct-loaded-cylinder load-cell, Type ER60, suitable for loads up to 670N, at 100mm radius. This instrument gives a rather higher possible torque maximum than is essential; but, of the various makes and types which would also be suitably interchangeable with the end-load cell, as recommended below in Section 3.3.2, this is the nearest in range.

Coutant instruments are selected because their standard sizes come nearest to the present needs and their repair and maintenance facilities appear to be the most effective when considered relative to the location of the College.

Within the limits of the machine design, the load measuring accuracy is to be kept within $\pm 2\%$ of the respective, normal, maximum load ranges.

Existing standard Departmental equipment is to be used to indicate and record the load cell outputs. Although only three tracks are likely to be needed in this instance (1 for torque, 1 for end-load and 1 for temperature) the instrument most likely to be used is a Southern Instrument Co., ultra-violet, multi-beam, reflecting galvanometer type, set to record six channels, simultaneously, on 250mm wide sensitized paper moving at a calibrated speed of 100mm/sec.

The saddle, which carries the torque-measuring equipment, is to be made free to float in the axial direction, to keep restraining end-loads to a minimum.

3.3.2. Measurement of End-Load.

End-load measurement is a speculative venture, because the likely magnitudes of end-loads are not known. However, in verbal discussions with workers in this field, values up to 500N were mentioned as having been obtained in work, subsequently discontinued, on rigidly-held test specimens of sizes and in materials similar to those likely to be used in the present application; therefore, provision is to be made for rigid end-load measurement up to one quarter in excess of this value, namely, to 625N.

A standard Coutant, reversible, direct-load cell, the type ER60, maximum load 670N, fits this range quite well and is sufficiently near to the maximum-torque-range cell requirements, already indicated in Section 3.3.1, to justify the use of this same size and type of cell for that application; thus providing, at no extra cost, an emergency, spare, torque cell for use when end-load measurement is not being made.

3.3.3. Protection of Load Cells.

It is desirable, both to maintain accuracy and to avoid unnecessary cost, to protect load cells from the danger of serious overloading.

Because end-load measurement is speculative, it is not possible, at this stage, to provide overload protection for that load cell; but the risk should be small since end-loads are likely to build up gradually and give time for emergency stopping of an experiment before a safe limit is exceeded (Coutant cells should be able to take 50% overload without damage).

However, torque loads can build up suddenly and protection must, therefore, be provided in the form of adjustable limit stops acting appropriately on the rotational movement of the cell-loading lever.

3.4. Thermal Conditions.

Cost prevents the installation, at this first stage, of more than the cheapest forms of heating and cooling; but provision, in terms of space and accessibility, is to be made in the design for subsequent adaptation to more efficient systems. A temperature of 600°C is to be made the normal maximum for the initial operations; but, with provision for temperatures up to 1250°C.

3.4.1. Heating.

As considerations of cost are of overriding importance, the heating system must be of a simple, radiant, electric-resistance-heating type, using a refractory-tube furnace. Nickel-chromium windings round a silica tube should be adequate for the specified, normal maximum, temperature range and suitable adjustment of the winding turns will minimize the end-effects.

Control is to be by means of a Eurotherm temperature controller, type PID, built into the instrument panel, suitable for operation from a platinum - platinum .13% rhodium thermocouple, to give a provisional control range up to 1600°C and capable of passing heating currents of up to 20A. A Eurotherm instrument is selected because,(a) it is reliable, (b) this type of instrument is well known and understood in the Department, and (c) it requires minimal provision of associated control equipment, such as relays, rheostats etc.

The furnace must be mobile in the axial direction,to allow for easy setting-up and removal of specimens and to allow for water quenching of the specimen. For the latter purpose it is to be automatically moved, at high speed, by a pneumatic piston. Thermal insulation of the furnace is to be made as effective as possible, to avoid heat loss and to prevent overheating and distortion of adjacent parts of the equipment; but without causing an undesirable increase in the dimensions of the machine, such as making the bedplate-to-axis distance too great. (The latter change would make the operation of the machine less stable and, hence, more difficult to control if other related dimensions were not increased in proportion). Thermal insulation need not be such a serious problem with more sophisticated heating systems, even at much higher temperatures, because then the heating can then be kept much more localized (e.g with electric induction or direct-resistance heating).

The possibility of using a higher-temperature radiant electric-resistance-heating system, for occasional use at temperatures higher than 600°C, is to be investigated and the system incorporated if the costs permit.

Oxidation is to be lessened by making provision for the use of a controlled atmosphere, to give as much protection as possible without high cost.

3.4.2. Cooling.

For metallurgical reasons, rapid cooling is often required at the close of a test. Many materials will require water quenching; and water cooling is therefore, to be made standard.

Cooling water must not be allowed to contaminate any part of the equipment, so it is essential that a suitably designed quenching chamber, that will not hamper other operations on the machine, must be provided. A recirculating system is unnecessary since the amount of water required is small and will probably not exceed 5 litres/min for, perhaps, 5 minutes at a time with intervals of an hour or more between.

3.5. Automation.

A high degree of automation is beyond the cost allocation of the present project, but it is essential to be able to automate the timing of the test cycle, or cycles, and it is essential to automate the quenching sequence to give optimum efficiency of quenching. It is necessary, also, to interlock both the automated and the manual operation systems with the electric power, the water, and the pressure air supplies both for physical safety, as outlined below in Section 3.6.1, and for the protection of the equipment.

3.5.1. Timing Sequence.

Electronic operation, based on the use of trigger pulses, is to be used to give a choice of each of the following sequences.

(1) Automatic cycling operation. This is to be initiated by manual closing of the electro-magnetic clutch "start" operator and give the following sequence. (a) The "start" operator actuates the strain timer and, simultaneously engages the electromagnetic drive-clutch switch. (b) At the end of the

preset straining time, the strain timer actuates the interval timer and, simultaneously, disengages the drive-clutch switch. (c) At the end of the preset time interval, the interval timer actuates the strain timer and, simultaneously, closes the drive-clutch switch. Sequences (b) (c) repeat until either the clutch "stop" operator switch is operated manually or a "stop" signal pulse is fed in independently from an external source such as the total strain counter.

The system is to be suitable for connecting-in a batch-timing signal, from the total strain indicator (if the latter is fitted with a batch timer), in place of the strain timer, so that stage (b) becomes strain-increment dependent instead of strain-time dependent.

(2) Automatic single-cycle operation. This is initiated by manual closing of the electro-magnetic clutch "start" operator. The "start" operator, simultaneously, actuates the strain timer and engages the electro-magnetic drive-clutch switch. After the preset strain time the timer disengages the clutch switch and the sequence ends until the drive-clutch "start" switch is closed again, manually.

(3) Manual operation. Manual closing of the drive-clutch "start" operator initiates straining which continues until the clutch "stop" operator is opened manually.

The respective time lags on starting and stopping operations of the electro-magnetic clutch are not to exceed $\frac{1}{10}$th second.

If the main drive motor is not running, it should be made impossible to engage the electro-magnetic clutch.

3.5.2. Quenching Sequence.

Manual closure of the "quench" operator switch should, with a single pulse, simultaneously, switch off the furnace and operate an air valve to move the furnace out of the way and replace it with the quenching chamber; completion of furnace movement should close an electric circuit which will open the cooling water valve.

Absence of water or air pressure should deenergize the system, so as to make it inoperative and leave it in a safe state.

Return movement of the furnace should be controlled to a suitably slow speed by a preset air-flow speed-control valve, actuated by a pulse from the manually operated "heat" operator switch. The latter should be interlinked to ensure that the furnace is not switched on while it is in a position in which it is heating the torque measuring shaft and not the specimen.

3.6. Safety.

The apparatus is to be made as safe to operate as possible, without hampering essential functions.

This requires attention to the safety of electrical equipment, heated parts and dangerous moving parts. As much as possible of the manual manipulation required in operation is to be centred on a height between 900mm and 1m.

3.6.1. Safety From Electrical Hazards.

All electrical equipment is to be properly protected against the risk of anyone accidentally touching "live" parts, against the risk of any accidental shorting likely to cause fire, and against the ingress of water where that might be reasonably likely to occur.

Electrical interlocking is to be used, wherever possible, to avoid the risk of the accidental creation of a dangerous situation (e.g. see last paragraph of Section 3.5.2.). The machine should "fail safe" in the event of any unexpected happening.

3.6.2. Safety From Heated Parts.

Complete safety from heated parts is not possible, since some work must be done in or near exposed hot zones when setting up an experiment or removing a broken specimen; but, for routine operation, an electric interlock is to be incorporated into the design in conjunction with the protective cover mentioned below.

3.6.3. Safety From Moving Parts.

Normal fixed guards are to be fitted over gears, drives and other dangerously rotating surfaces. Particular protection is to be given by a protective cover, closed manually, for use during quenching operations. This cover is to be electrically interlocked with the "quench" operator switch and mechanically locked by means of an electrically interlocked latch so that it cannot be opened during a quenching operation.

3.7. General Requirements.

Apart from general convenience and safety of layout, provision is to be made for possible future development with respect to reversed straining, end-load experiments, more effective heating and dual operation.

3.7.1. Provision for Reversed Straining.

Reversed straining seems, at present, to be a less urgent need and, at this stage, costly provision is not justified. However, if provision is made for dual operations it will be possible to adapt this to reverse drive purposes, after providing modified specimen adaptors and attaching a suitable duplicate drive mechanism in place of the second test head.

If suitably modified specimen adaptors are provided, reverse straining is possible from the Carter gear box control but the reversals would have to be manually set each time; entailing an interval of at least 15 seconds between reversals.

It is not proposed that modified adaptors be incorporated at this stage.

3.7.2. Provision For End-Load Experiments.

Provision is to be made for rigid end-restraint with optional end-load measurement, and for almost complete end freedom; but, more complex approaches may be required if that kind of work is taken up more seriously than at present. Whilst cost prevents any greater provision for this contingency, it is logical and practical to provide, in the basic design, an extra 100-150mm on the length of bedplate, extending beyond the minimum length provided for basic end-load measurement and control.

3.7.3. Provision For More Effective Heating.

At this stage it is not possible to provide a more effective heating system; but provision is to be made for reasonably generous space in the vicinity of the test position so that alternative heating equipment may be incorporated, subsequently, when necessary and financially possible.

3.7.4. Provision For Dual Operation.

There is insufficient money for a second test head at this stage, but the design is to include provision for the future easy attachment of a second head to the main drive frame, at 90° to it and at 180° to the first test head on the same axis.

4. DESIGN OF MACHINE.

Determining Factors in Design.

In a text such as this, when outlining the determining factors of a design, it is not necessary to reproduce routine, conventional, standard (2,33) design factors, except where these influence a decision to be made on some less standard aspect of the design. On the other hand, it is necessary to outline those factors, peculiar to the particular situation, which influence decisions related to:- (i) systems of operation, (ii) systems of construction, and (iii) maximum working stresses.

Included in these particular factors are the following: (a) local availability of electric power, (b) limitations of machining capacity, (c) effects of automation, (d) influences of stiffnesses and vibrations and (e) frictional limitations.

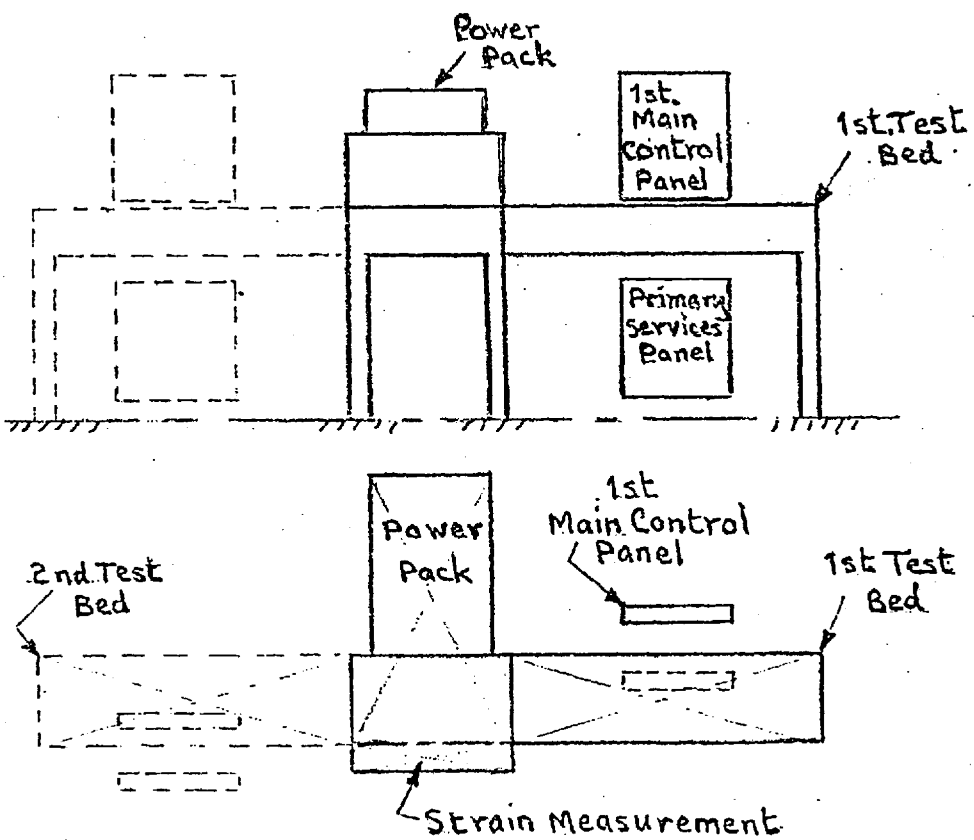


FIG. 17. Preliminary block layout of machine.

After outlining these in the present Section, consideration is given in later Sections to particular aspects of the design.

A block diagram of the prospective basic layout of the machine is given in Fig.17. and appropriate sketches are included in the subsequent text; but the final assembly and details are recorded in photographs of drawings TTM 1-13, included in Appendix A, page

4.1.1. Availability of Electric Power.

In at least two of the possible sites for the machine, electrical power supply is limited to one 13A three-phase, four-pin, power point (i.e. there is no neutral line), one 30A single-phase, three-pin, power point and two (or more) 13A single-phase, three-pin, power points. Each of the relevant single-phase power points comes from the same common phase. In these circumstances, at first sight, the logical approach would seem to be to use the three-phase supply for the motor drive and the 30A single-phase supply for both heating power and all control circuits. However, it is desirable to separate the heating power circuit from the control circuit to allow alternative heating systems to be employed, if and when required, without having to make any alterations to the control circuits. The latter system is adopted and, as can be seen in the circuit drawing TTM 1., Appendix A, the heating power circuit is electrically isolated, from the other circuits.

Thus, three power inputs are used for the main machine; one 3-phase and earth, one 30A single-phase and earth and one 13A single-phase and earth. Ancillary equipment is separately powered from the spare 13A three-pin point.

4.1.2. Machining Capacity.

Owing to the limited machining facilities available in the machine shop, unit parts are kept reasonably small so that, as far as possible each individual machining operation can be done at one setting. Where this is not possible, the design is modified to permit the use of accurate stock material, such as bright-drawn bar, in such a way that it may be hand-fitted to the necessary degree of accuracy over lengths, or larger areas, than can be covered, effectively, by available machining equipment.

4.1.3. Influence of Automation etc.

The essential degree of automation as outlined in Section 3.1, is not very great, but, as indicated in the circuit diagram TTM 1, the controls, in general, are made electronic as far as is reasonably practical. This means (i) that all the selected equipment, such as electrical switchgear, etc., is of a type suitable for remote control and (ii) mechanisms such as furnace-moving devices etc., are fitted with appropriate mechanically activated microswitching to ensure interconnected operation.

4.1.4. Stiffness and Vibration.

To ensure accurate operation, free from backlash and free of undesirable effects from inherent or resonant vibrations, the construction, particularly of parts subjected to stress arising directly from the torsion-applying action, is made much more robust than simple strength limitations require. Shafts are of significantly larger diameter, with minimum diameters catering for a stress safety factor greater, by a factor of at least 10:1, than the normal minimum safety of 2 on the yield strength (i.e. 4 on the tensile strengths of normal annealed mild and low alloy steels). This higher total factor of 40:1 is an arbitrary compromise between the minimum factor and a much higher factor that would:-

(i) give an excessively massive appearance to the construction,

(ii) add greatly to the inertial forces in the moving parts, and

(iii) increase the required machining capacity beyond the reasonable range of the machine shop equipment.

The larger sizes given by the compromise high safety factor have these effects:

(a) torsional rigidity is high and backlash low (see Appendix C),

(b) frequencies of torsional vibrations are high and the chances of exciting natural, or resonant, vibrations are small (see Section 4.5),

(c) transverse vibrational frequencies are high, even while maintaining bearing centres at reasonably workable distances, thus minimizing the chance of transverse bending vibration, either from out-of-balance torsional forces or incidental vibration from other sources, such as the electric motor (see Section 4.5.).

4.1.5. Friction.

To maintain the degree of load-measuring accuracy outlined in Sections 2.2.3. and 3.3.2. and to keep power-loss low, friction is kept to a minimum. This is ensured by the liberal use of ball and/or roller bearings on all relevant rotating or other moving parts; but this liberality is tempered by the need to keep mechanical construction as simple as possible.

4.2. Power Pack.

In approaching the final layout of the design it is apparent that the power pack is going to have a dominating influence on the construction.

There are several reasons for this:-

(i) The power pack is likely to be the bulkiest and most awkwardly shaped part of the structure.

(ii) The placing and layout of the power pack will determine the general ease of accessibility, for normal working of the equipment, and its adaptability for subsequent variations in use.

(iii) It is likely to be the most difficult part of the construction to adapt to the available machining facilities.

(iv) Vibration is most likely to originate in the power pack; therefore its design requires particular attention from the beginning.

According to the particular aspect of construction and/or prospective operation of the equipment that is under consideration at any particular time, the relative importance of these reasons will vary; but, it is logical to base many decisions on general layout and/or details of design on the characteristics of the power pack. Therefore it is considered here before going into details of the design of the test bed etc.

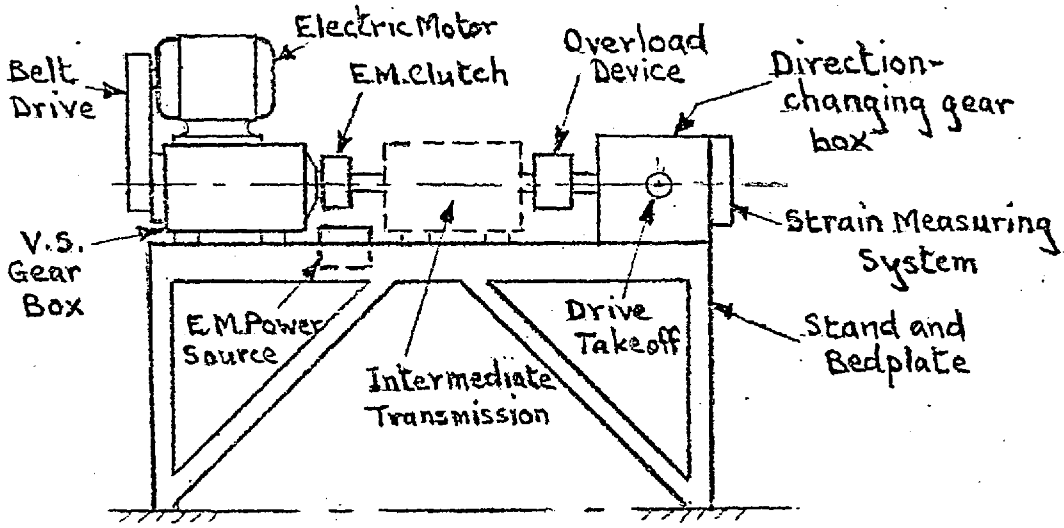


FIG. 18. Power pack system,
tentative layout.

The power pack, see Fig.18 and drawing TTM 2, consists of the motor, the hydraulic gear box, the electro-magnetic clutch, the intermediate transmission system between the clutch and the gear box (including a safety device against overload), the drive direction-changing gear box, the engaging mechanism for the final drive, the support frame and the strain-measuring system. The last is included in the power-pack aspect because of convenience of take-off on the lines discussed in Section 3.2.

4.2.1. Power Pack Stand.

The pack, Fig.18, is integrated by the stand (Item 1 Part 1, drawing TTM 4) upon which it is built and which accommodates the various parts so that the drive alignment is maintained, by setting the parts at the required working heights. The stand is rigid enough to be set upon four legs, stiff enough and suitably placed to be stable, adapted to rigid bolting attachment of the test frames at right angles to the drive frame, as indicated in Fig.17 and specified in drawings TTM 4 and 5, and free of low-frequency transverse vibrations.

To achieve these ends, the whole stand is made of welded construction,based on a rectangular bedplate made of 102mmx 51mm mild steel channel section welded with its open side facing inwards. On top of the stand, three sets of mounting pads are welded in positions suitable for bolted attachment of the hydraulic gear box, intermediate transmission and the drive direction-changing gear box ~~is~~ discussed in Section 4.2.5. From the limited information available about the electro-magnetic clutch,it appears that no special mounting is required for this. (In fact, the output shaft from the clutch turned out to be unsupported and necessitated the introduction of a small support bearing for use when using the direct-drive intermediate transmission shaft,Item 18,drawing TTM 8). Each set of mounting pads is surface-machined, accurately, (3 individual setting operations) to maintain the coaxiality of the drive train.

The legs are made of 51mm x 51mm x 6.5mm mild steel angle, set to splay, sideways, outwards towards the bottom and suitably braced with diagonals running between the respective bottom ends and positions near the midlength of the underside of the adjacent long side of the bedplate. The end pairs of legs lie in the vertical planes of the bedplate ends to take a transverse angle, welded flange-down between the bottom of each end pair, for bolting down to the floor, or levelling, if required.

4.2.2. Electro-Magnetic Clutch.

The electro-magnetic clutch, which transmits the drive directly from the hydraulic variable-speed gear-box output shaft, see Fig.18, has its magnetic current commutator firmly anchored to the gear box end. Its position in the train of drive is chosen primarily for convenience and cheapness, since the gear box can be supplied with the clutch in this position as a standard fitting. This arrangement may cause some difficulty in releasing the torsional load, if a speed reducing gear is in use for speeds less than 30 rev/min, but this is not considered to be a serious handicap at this stage.

Associated with the clutch is its low-voltage energy-pack which is placed conveniently close to the clutch, firmly bolted or screwed on the control-panel side of the stand bedplate. (If attached to the other side it could, subsequently, be in the way of the operator of a second test frame when one is installed).

4.2.3. Intermediate Drive and Safety Device.

The drive train is adaptable, within limits, to change in the basic ranges of drive speed, or power, subsequently, in the event of requiring certain test conditions outside the basic ranges. (There is a minimum speed limit of 30 rev/min on the output speed of the Carter gear box as well as a top speed limit of 970 rev/min and maximum torque limit of 22.3Nm.)

If a speed change, up or down, is required it is achieved within the coaxial alignment of the drive train in an axial space left for this purpose. It is possible to custom-build an adaptor gear; but a standard, in-line ("helio-centric") vibration-free, gear drive is readily available, made by Sanderson. A reducing gear is the most likely type to be needed, to give output speeds down to 1 rev/min; to avoid damage to its output shaft, the latter must be of sufficient size to take the full amplified torque from the drive side. Inevitably, this means a relatively large output-shaft diameter and allowance is made for this in the design. Ideally, the 1 rev/min minimum speed requires a 30:1 reduction; but the nearest available standard ratio, in the Sanderson Heliocentric Gear, Type G range, is 50:1 reduction. This ratio permits the use of drive speeds down to 0.66 rev/min (0.011 rev/sec) and makes strain reading and calculation quicker (speed input setting has to be coordinated with desired output), so this ratio is adopted. Provision is made for accurately mounting this particular box on the bedplate, as shown on drawing TTM 4, but the decision to purchase the reduction gear is to be made subsequently, when it is clear that it can be afforded. The speed reducing gear is to be connected into the transmission system by means of the same couplings described below in relation to the direct drive coupling shaft.

A simple removable section of transmission shaft is provided (Item 18,drawing TTM 8) for normal direct operation without the heliocentric gear. This shaft is made of low carbon steel (En 3-shearing strength 285N/mm^2) with its input end the same diameter (22mm) as that of the input shaft of the heliocentric gear and its output end of the same diameter as that of the output shaft of the gear. The former size makes the input end safety factor at 22.3Nm torque, only 27:1 but this is acceptable at this location since it does not affect the overall stiffness of the machine to a significant extent. On the input end is fitted a standard removable zinc-base die-cast, spider coupling flange, which engages with the matching flange of the coupling, permanently fitted to the electro-magnetic clutch output shaft. The coupling connection is not of the direct metal-to-metal type but incorporates a neoprene shock-absorbing transmission insert. (Note. It was found, subsequently, that the weight of the transmission shaft insert, acting through the coupling on to the clutch output shaft, was sufficient to cause misalignment of the latter and consequent inaccuracy in the operation of the clutch. This was overcome by providing a small steadying bearing, close to the clutch, to support the outside diameter of the flange of the coupling on the clutch output shaft, see note in Section 4.2.1.)

Between the intermediate drive and the direction-changing gear box input shaft is fitted an overload safety device, which is described in the next sub-section.

The whole intermediate system is arranged so that free interchange of shaft insert and reduction gear box is possible within, say, 15 minutes. This is managed by making the input coupling-flange free enough to slide right on to the input shaft end, until it clears its matching flange, and by making the overload safety device coupling, Item 19, drawing TTM 8, slidable, on its key, right on to the input shaft (Item 22, drawing TTM 8) of the direction-changing gear box so that an axial clearance of a few millimetres is left between the clutch-output coupling flange and the direction-changing gear box input shaft-end.

4.2.4. Overload Safety Device.

The overload safety device is needed to serve three purposes.

(i) In the event either of seizure of the machine drive in the direction-changing gear box and its subsequent train, or in the event of failure of the heating causing a sudden rise in torque-demand, it will protect the drive motor, gear box etc.

(ii) In the event of a sudden rise in the torque demand, perhaps due to heating failure, it will prevent serious damage to the strain measuring system.

(iii) In the event of seizure, either in the direction-changing gear box or in the subsequent train, when a speed-reducing intermediate gear is in use, it will prevent overloading of any part of the former system under the excess torque available from the speed-reduction gear output, as described in the preceding sub-section.

The form of the safety device is very simple and is no more than a straightforward, removable, double-shearing pin, see Fig.19, suitably mounted in a slidable coupling sleeve.

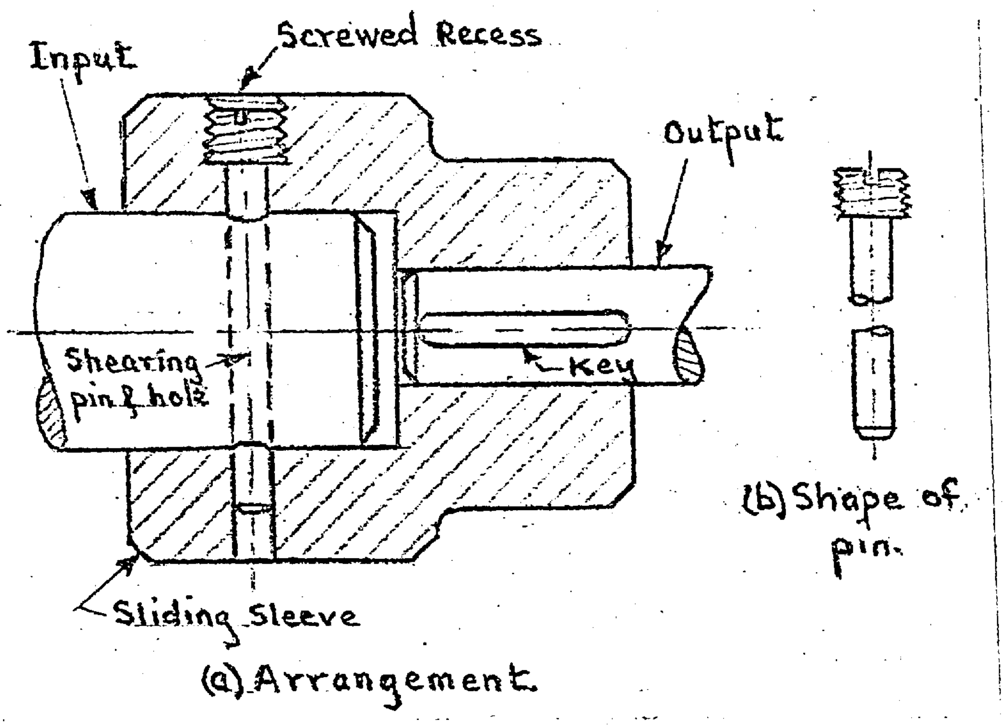


FIG. 19. Overload safety device.

The shearing is set up between the sliding sleeve (Item 19, drawing TTM 8) and the transmission-shaft-insert output end or the intermediate reduction gear output shaft, as appropriate, through the pin. The other end of the coupling slides freely, but not loosely, on the direction-changing gear input shaft and key (Items 18 and 21). A drilled and reamed hole run accurately through the sleeve and relevant drive shaft end carries a shearing pin (Item 20, drawing TTM 8). Both sleeve and drive shaft are of En24 steel, heat-treated to 1GN/m^2 tensile strength, hard enough to provide good shearing edges against the pin, which is made of a relatively weak material, such as a brass, in a suitable structural condition (e.g. annealed). The condition of the pin is important in ensuring a reasonably safe shearing load; therefore, to ensure that unsatisfactory material, casually picked up, is not used, the pin has a special shape to draw attention to this factor. It has a screw thread on an enlarged end, see Fig.19(b), which should not only draw attention to the special nature of the pin but provides a means for holding the pin in position and yet makes it easy to remove or replace it, without risk of damage to pin or sleeve. A slot is provided, in the pin end, for a screw driver. Because shearing failure in these conditions is somewhat unpredictable, due to the associated elements of bending and compression deformation, it is not possible to calculate accurately the exact nature of the material required for the pin. Hence, some experimentation is required to find the correct material. First estimates indicate that an annealed brass should be about right for a torque of about 23Nm. If, at a later date, a higher torque is to be deliberately applied, via the intermediate reduction gear, or with a modified drive, then the pin must be graded up to a high-tensile brass, a phosphor bronze, or other suitably-stronger material.

The maximum shearing stress in the key (Item 20), at 46Nm torque, on the direction-changing gear box input shaft is 15N/mm^2, giving a minimum safety factor of 27:1, which is adequate in this location.

4.2.5. Direction-Changing Gear Box.

Because the machine is designed with the facility for adaptation to two test frames, see Fig.17, and easy adaptation to strain measurement, it has its test-frame axes at right angles to the primary drive axis, and the final drive is through a direction-changing gear box.

The choice of gear systems for an application such as this is limited to three:-
(i) helical spur, (ii) plain bevel, and (iii) spiral bevel.

Helical gears are relatively simple in concept, smooth in operation and not excessively costly to make, but they do require a significantly large offset between their centres, either up or down, from one axis to the other. There can also be an appreciable amount of friction associated with their operation and, whilst the friction itself need not be intolerable, it will vary quite widely and undesirably with variation in lubrication. To offset the adverse effects of such gears, the complexity and cost of the construction would be raised beyond acceptable limits; hence this system is ruled out.

Plain bevel gears are simple and straightforward in operation and construction, and are economical in space. However, drive transmission through straight bevels is not perfectly smooth and there is some risk of chatter from resonant vibration. If the latter can be controlled or rendered insignificant, without involving a lot of extra cost, then this system could be satisfactory in the present instance.

Spiral bevel gears are considerably more costly to make than plain bevel gears but greatly reduce the risk of chatter.

On the whole it is cheaper to use plain bevel gears, as shown in Fig.20, rather than spiral bevel gears, even after making allowance for chatter, therefore this system is incorporated (Item 27, drawing TTM 9).

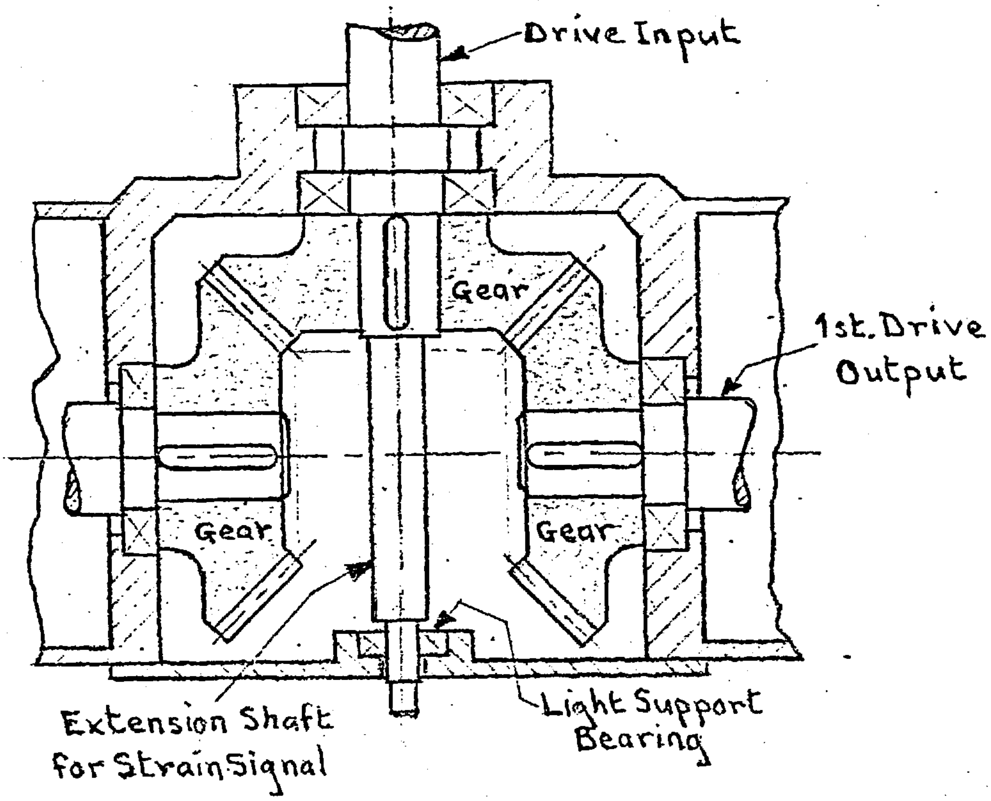


FIG. 20. Direction-changing gear box. Drive shafts running in taper roller bearings. Section on horizontal axis centre.

It is not practical to produce bevel gears in the departmental workshop so they have to be bought from outside. The overall gear size has to be kept as small as possible, consonant with adequate strength and a minimum number of teeth, but there may not be less than 25 teeth; below this number, the risk of chatter becomes progressively greater with fewer teeth. In addition, the tooth pitch should not be excessively large, say about 3 or 4 module. A commercial gear manufacturer is able to offer bevel gears, of 3.15 module, with 28 teeth, made in heat-treated En9 steel, giving a safety factor of about 7. The latter is on the low side but is considered satisfactory for the immediate requirements. If, subsequently, a much higher torque is required frequently, these gears would have to be replaced with others made in stronger material. (A maraging steel will give the maximum possible strength without having to make changes in the main construction).

A relatively low safety factor (high working stress) is acceptable in this particular component, since the effects of high working stress, on both back lash and vibrational frequency, are small; that is the maximum amplitude of elastic deflection of tooth is small and the natural frequency of vibration is very high.

On the other hand, the most likely main cause of vibration in the whole machine is the vibration due to slightly uneven transmission of load as the latter transfers from one tooth to the next during gear rotation. This effect is given more consideration in Sub-Section 4.5.2.

To avoid undesirable vibration, due to looseness in the bearings, the gears must be very firmly and accurately mounted on their shafts and the respective shafts must run very accurately and smoothly in their bearings after taking into account the noticable outward thrust from the gear teeth. A nearly ideal solution is adopted for this by using on each gear shaft a pair of taper-roller bearings, adjusted to give minimum end-clearance by opposing the tapers and presetting their position on the shaft, as shown for the input shaft in Fig.20. Skefco taper-roller bearings, for a shaft diameter of 30mm (maximum safe load at low speed 20kN) would provide a suitable minimum size, combining rigidity with adequate low-speed strength and low friction losses. Adjustment to local design sizes necessitates using slightly larger (and safer) sizes of bearings (Item 14 and 15, drawing TTM 3). The drive keys (Item 23) are stressed to the same level as the input drive key mentioned in

Section 4.2.4.

Lubrication of gears and inner bearings is provided by making the lower gear teeth dip into a bath of lubricating oil. The taper-roller bearings are packed with grease on assembly and are provided with ready means for uncovering and repacking them after a suitable length of service.

An extension shaft is provided on the end of the input shaft as shown in Fig.20, to carry the rotation forward, out of the gear box enclosure, for taking off strain measurements in the manner described in the next sub-section.

Provision of an alternative drive output is made in the direction-changing gear box at this stage to avoid the likely high cost that would be involved in a subsequent modification to adopt a single-output gear box to drive a second test head.

4.2.6. Strain Measurement Drive and Associated Equipment.

To operate the electronic strain-sensing devices, specified in Section 3.2, pulsed signals, timed from the main drive shaft, are supplied. For this purpose a light shaft extension is added to the direction-changing gear box input shaft, (Item 22, drawing TTM 8,) to carry it forward through the front of the gear box and there to drive the generator disc Fig.21 (Item 78, drawing TTM 11) for activating the pulse sensors. To prevent unwanted vibration and ensure smooth accurate running the extension shaft is steadied on a light ball-bearing (Item 77) set in the gear box cover, item 28, drawing TTM 9.

Total strain twisting is to be measurable to an accuracy of $\frac{1}{10}$th of a revolution (i.e. to about 0.63 of a radian) and strain speed is to be sampled over a similar and certainly not much greater amount of strain. To satisfy both these conditions, without requiring an excessively large pulse-generating device, it is decided to generate 60 pulses/rev and to count up one decimal point for each 6 pulses (maker's recommendation). With this pulse frequency, speed sampling can be set for every six pulses, so that strain rate is indicated either once every $\frac{1}{10}$th rev or once every $\frac{1}{5}$th rev. (In initial practice, the maker's claims for speed sampling do not appear to be met on either six or twelve pulse sampling, but this is probably due to inexperienced use of the equipment).

Separate signals are required for total strain and for strain rate and these are derived from two separate sensors picking up from the one generator wheel, on the lines laid down in Section 3.2.1.

This generator wheel (Item 78, drawing TTM 11) takes the form of a radially slotted steel disc rotating in the vertical plane, see Fig.21, with 60"square" teeth equally spaced around its perifery. The two sensors are mounted on opposite

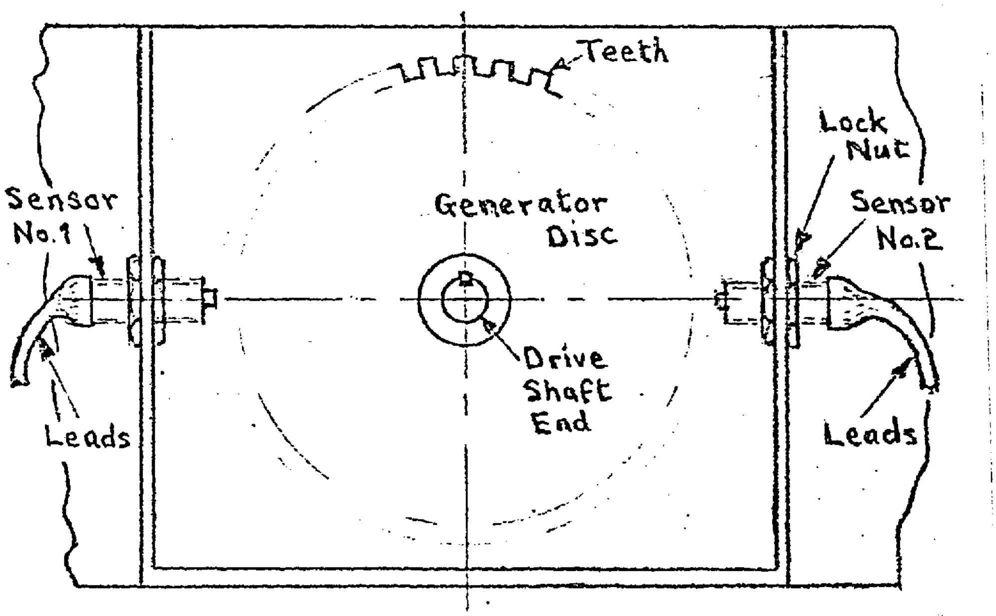


FIG. 21. Strain signal generator.

sides of the disc, in line with it on its horizontal central plane, and are each activated as a tooth moves through its inductive field. The sensor positions are chosen both to prevent the sensors projecting beyond the limits of the front outline of the gear box, so that they are given some protection (they are of very light construction not suited to rough usage), and to make it easier to put a cover over the arrangement.

4.2.7. Dog-Clutch Mechanism.

It is desirable to provide for positive disengagement of the drive when making preparations for a test, so that accidental starting of the motor, or accidental engagement of the magnetic clutch, when pretesting the speed output of the variable-speed gear box, does not cause damage or injury. Also, if, subsequently, a second test-frame is installed, it is essential to ensure that the drive does not transmit simultaneously to both sides.

These ends are served by a simple dual dog-clutch arrangement controlled by a linked lever system, as shown, respectively, in Fig. 22 and drawing TTM 2, in such a way that: (i) only one dog-clutch engages at a time and (ii) there is a positive intermediate lever position in which neither dog-clutch is engaged. Although provision is made, on drawing TTM 1, for electrical interlocking by means of a microswitch, actuated by the lever, this interlocking is not considered to be essential, at this stage, for single-head operations. It is not necessary to use a complicated spring-locking device on the lever, since it is not intended for frequent operations, so a simple locating pin, inserted and removed by hand, is used and should be quite satisfactory for all except extremely careless users.

The dual clutch system is designed into the initial scheme, in conjunction with the alternative drive incorporated into the direction-changing gear box of Section 4.2.5, to save excessively heavy subsequent cost if a second test-head is added, since the system is then already in a state suited to easy fitting of the latter.

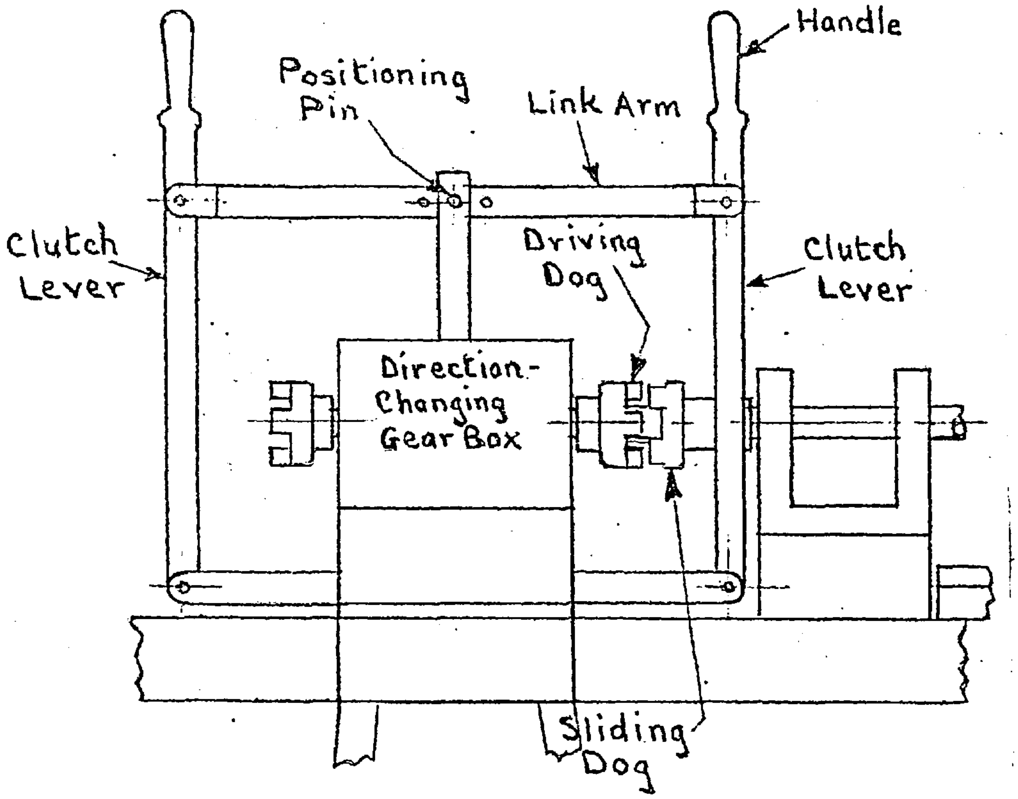


FIG.22. Dual dog clutch. Lever locked on and off by means of a hand-inserted positioning pin.

The sliding bobbin (Item 30,drawing TTM 9), which is the most complex part of each clutch,is located on the test-frame side of the drive, but one is not incorporated at this stage for the second test-head. This omission helps to keep the present cost of the provision for subsequent adaptation down to the lowest possible figure.

The dog-clutches are made of mild steel since they are not likely to be operated very frequently. Because reverse straining is not envisaged for the present, wide clearance is left between the meshing faces of the dogs and the latter are cut with square driving faces making manufacture simpler and clutch engagement easy. (If, at a later date, reversed straining is to be done the dog faces will have to be tapered slightly and sized to interlock firmly,in order to stop takeup and backlash during reversing.)

Because of the nature of the service of the dog clutches double key drives are used and the sliding bobbin keys (Item 31) are stressed only to 10 N/mm^2 at 46Nm torque, giving a normal minimum safety factor of 40:1.

4.3. Application of Torsion.

Having determined the main design layout with respect to the restrictions from the power pack, it is now possible to decide the details both of the mode of transference of torsion load to the end of a test specimen and of the system for anchoring the specimen at the opposite end.

To make an effective torsion test it is necessary that:

(i) the torsion ~~to~~ be applied coaxially, to a suitably standard-shaped specimen (test piece), in a suitably-controlled manner through a drive spindle,

(ii) the resistance to torsion at the restrained end of the test piece be rigidly coaxial with the test piece and drive spindle,

(iii) the torsional resistance be transferred, in a suitably-controlled manner, from the specimen back to an effective torque-measuring system,

(iv) any end-thrust on the specimen be reduced to a suitably-low minimum for testing in an axially-unrestrained state or else measured against a relatively rigid anchor system,

(v) insertion and removal of a test piece be simple.

Since the tests envisaged here are to be made at elevated temperature there are two more requirements:

(vi) there must be sufficient space to accommodate the heating system, without hampering the insertion or removal of test pieces,

(vii) the design of the heating system must not affect the precision of operation of the machine or the accuracy of measurement of the torque.

Each one of these requirements influences the design and construction of the test-frame of the machine since this is the part of the equipment in which the test is actually made.

There are five aspects of this part of the design that are of immediate interest, namely (a) the design of the frame in the test area, (b) the means for mounting the dog-clutch and drive shaft, (c) the method of gripping the specimen, (d) the system for setting up and dismantling a test and (e) the system of torque measurement etc. The first four are considered in the rest of this section and the fifth is given the next main Section to itself.

4.3.1. Testing Frame and Stand.

It is necessary in this part of the equipment to make provision for:

(a) rigid accurate connection of the test frame to the power-pack frame,

(b) rigid mounting of the torsion-drive dog-clutch and spindle mechanism,

(c) accommodation of a system of floating specimen-anchorage combined with torque measurement,

(d) accommodation for alternative systems of end-anchorage,

(e) supplementary equipment for heating etc.,

(f) incorporation of the main service-supply panel,

(g) production and assembly within the limited resources of the Departmental machine shop.

The testing frame must be of a very rigid construction, both for strength and to eliminate, as far as possible, any significant twisting or bending deflection. The likely-level of torsion from the performance of a test cannot possibly exceed 46Nm, acting over a length not greater than 1m. Bending moment, in the unlikely extreme case of the test frame hanging free on the power-pack frame, will not exceed 2260Nm from the combined dead weight of the frame, and all its superimposed equipment, plus the possible weight of a heavy man standing on the outer end whilst the frame is supported only at the inner end. Using the same size of mild steel channel section, 102mm x 51mm, used for the power-pack frame, welded open-side in, to form a rectangular frame, gives an absolute maximum stress in bending of 27.5N/mm^2 for the extreme case of cantilever loading and an absolute maximum bending deflection under extreme conditions of operation (assuming very poor adjustment of the machine's mounting) of 0.032mm. The maximum possible torsional deflection over the length of the bed is 0.0000114 radians. These values are all satisfactory for the present use.

Connection to the main frame is by means of four fitted bolts, but the end of the test frame (Item 1, Part 2 drawing TTM 5) also registers on a machined support shoulder on the power-pack frame (Item 1, Part 1 drawing TTM 4) so that, under normal conditions, the bolts have to take only incidental, horizontally-transverse, shearing loads since the shoulder takes all vertical shearing from the weight and any other downward loading on that end of the test frame. The arrangement also greatly simplifies manufacture by providing a positive, self-supporting, location between the frames. All the bolts have to do is to maintain positive uniform contact between the respective frame contact faces and sustain any small tensile forces that may be set up in addition to transverse shearing. In this connection it is important that the test frame should be firmly supported at the outward end to prevent the setting up of cantilever loading on the bolts. However, should the full possible cantilever loading, as outlined above, be accidentally applied, say in moving the machine, the maximum stress in the mild steel bolts due to the cantilevering should not exceed $25N/mm^2$, which is amply safe.

The shaft bracket, or countershaft block, (Item 3, drawing TTM 6) is positively and rigidly located, on its accurately-machined base, against accurately-machined pad faces on the test frame, by means of four bolts.

The whole system is adapted to provide accurate guide-ways for the floating, specimen-anchoring, platform etc. within the available machining facilities.

Fairly accurate, guided, movement is required over about 1.125m of the test-bed length and very accurate guidance is required, for about 0.4m in the middle of the right hand half of the bed, for the torsion block, item 4, drawing TTM 7. The less accurate guidance is required for movement of the furnace base and quenching chamber base (Item 8, drawing TTM 6, and item 9, drawing TTM 8). The required accuracy is obtained by using two bright-drawn, rectangular, steel bars as guide rails (Item 7, drawing TTM 7) each mounted on a bright-drawn, rectangular, steel support base (Item 6) which in turn is set upon suitable pads on the test bed frame, as shown in Fig.23. The pads are machined, as accurately as possible, in three machine settings and the respective support bars are then set on their pads and packed up exactly level by means of shims, item 71 (not drawn), inserted where a pad support falls short. The guide rails are now accurately clamped down in position on top of the steel supports, within the limits of accuracy stated in Section 4.3.3, the fitted-screw holes are accurately reamed through the rail assembly and into the frame, after which the fastening holes are tapped; then the rails are fastened down with fitted screws, item 72, drawing

TTM 11. Vertical setting is governed by the support-bar setting and the horizontal setting is maintained by the fitted screws. Of course, both rails and supports are accurately trued longitudinally, preferably within ±0.01mm before setting up.

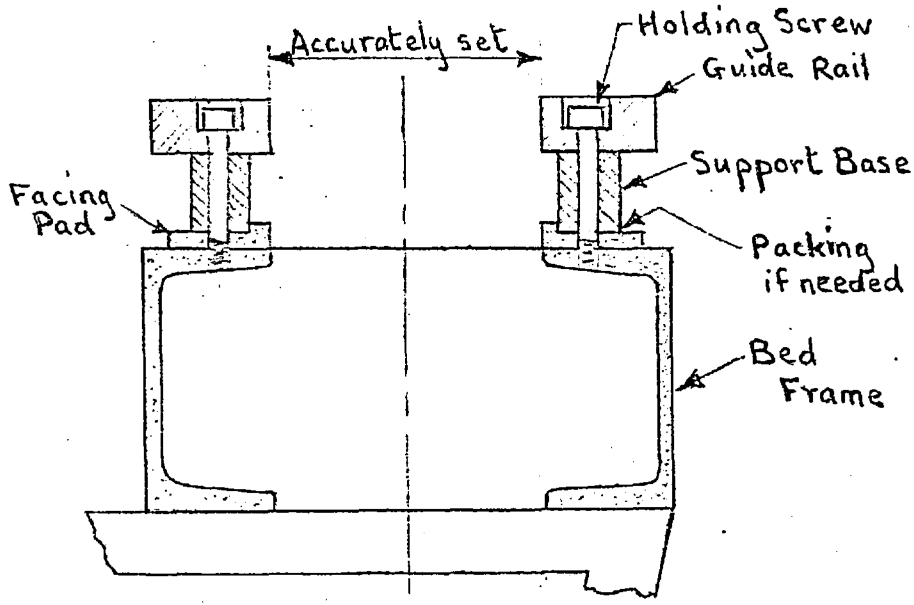


FIG. 23. System for obtaining accurately true guide rails with limited machining facilities.

End anchorage of the specimen anchor torsion bar, item 40, drawing TTM 9, is accommodated by using an anchor thrust block, item 5, drawing TTM 7, which locates on the guide rails and may be friction clamped in position.

The main service-supply and servo-control panel is fitted under the testing bed, to the rear of and between the two inner support leg frames, as already shown in Fig.16, p.57, which, at their front faces, slope back from the front towards the bottom to give knee and foot clearance as required in Section 2.1.3.(iv).

4.3.2. <u>Dog-Clutch, Drive Shaft and Specimen Adaptation</u>.

The dog-clutch mechanism is outlined in

Section 4.2.7. and the relevant bracket on the test-bed frame is mentioned in Section 4.2.1. On the test-bed side of the clutch the main driven shaft, item 35, drawing TTM 9, is carried in the bracket item 3, with the sliding clutch bobbin, item 30, drawing TTM 9, suitably keyed to its projecting input end and with the test piece adapting system, items 37 and 39, drawing TTM 9, fitting on its projecting output end. The shaft is accurately aligned and held on two opposed, Skefco, taper roller bearings (Items 14 and 15) of more than adequate strength and set to keep friction low, without permitting more than about 0.025mm end play relative to the test-bed.

The drive shaft is 46mm in maximum diameter and 30mm minimum effective diameter, giving a maximum shearing stress of 8.5N/mm^2 at a torque of 23Nm. This stress has an overall safety factor of 54:1 and is rather higher than the minimum recommended in Section 4.1.1. because (i) sizes have been kept up near to the minimum ball-bearing sizes, indicated in Section 4.2.5, and (ii) some extra allowance is made on the sizes at the output end to take care of any accidental overheating whilst under load.

The means used for connecting a test specimen into the system is an important aspect of the design which must take account of: (a) the need for accurate alignment between the driving adaptors and the "anchoring" adaptors, (b) the likely elevated temperature of the equipment near to the specimen and (c) the high cost of heat resisting materials. Points (b) and (c) are important for tests made at over 600°C and particularly important for tests made at temperatures exceeding 1000°C.

To accommodate these requirements, within the budget as far as possible, the system is divided into three parts on each end of the test specimen, namely, a shaft, a good quality heat-resisting adaptor for the shaft and a very good quality heat-resisting adaptor nut between the adaptor and the specimen; thus allowing progressively more suitable materials to be used as the elevated temperature zone is approached. Alignment is important; therefore, as can be seen for example on the output end of the driven shaft, item 35, drawing TTM 9, a No.4 Morse taper is provided, on each shaft end adjacent to the specimen, to give rigid, accurate, alignment to each respective adaptor, item 37, drawing TTM 9. The Morse taper not only allows for accurate rigid alignment but also provides a ready means for safe, simple, fitting and removal of its relevant adaptor, without risk of damage to adaptor or machine. Removal of the adaptor is effected, without shock or impact, by means of a suitable wedge actuated by a standard portable hydraulic jack acting against the resistance of a specifically designed and manufactured adaptor claw. The claw is not detailed in the torsion machine drawings since its design relates primarily to the jack, which is a standard workshop type with a screwed attachment system for retaining fittings.

4.3.3. Specimen and Specimens Grip System.

Although there is no one standard size or form of torsion test piece (see Table 1) and although likely approximations are suggested in Section 2.1.1, it is desirable, at this stage, to formulate more precise requirements upon which to base specimen adaptation.

The system considered most suitable is based on a maximum size of test piece having a gauge diameter of 13mm, shoulder length (see Fig.4, p.21.) of approximately 40mm, blending radius of 3mm and with 1in diameter B.S.F. screwed ends, of type (a) Fig.3, p.19. This design is selected because both of the threaded ends can be machined, in one machine setting, in conjunction with the gauge cylinder, thus ensuring perfect coaxiality in the specimen.

In addition, the specimen ends can readily be machined accurately-square with the body so that they will drive uniformly against the adaptor drive end face, without causing any deflection of the specimen. Thus, the rigid requirements for the specimen accuracy, in keeping with the need for accurate torsional loading, see below, can be met without too much difficulty. Adaptor nuts, item 39, drawing TTM 9, provide for the maximum-sized specimen but other adaptors will have to be made if other sizes are to be tested.

It is essential, (i) for the maintenance of accuracy of application of torsional straining, (ii) to avoid side thrust at the test piece position, where the spindles (Items 35 and 40) are most susceptible to deflection because of their long overhang, and (iii) to stop unwanted vibration, that eccentricity of loading and/or axial misalignment should be avoided. Eccentricity of loading is likely to be the worst individual cause of trouble since the side thrust at the specimen could be about 82N for 0.05mm eccentricity and about 320N for 0.5mm eccentricity, each at 23Nm torque. The former eccentricity error would be well within the absorption capacity of the machine and would have negligible influence on the indicated torque (0.112Nm max.). Hence, concentricity at the specimen must be kept within ± 0.05mm and preferably better. This necessitates provision of means for adjustment of one of the spindles relative to the other and item 40 is obviously more suited to give this adjustment, since it is already mounted on a "floating" carriage (Item 4, drawing TTM 7). Overall axial alignment is not quite so critical and is easier to control; but,if concentricity is to be maintained on the floating spindle, it is essential that the axial alignment of its guide rails should be good. The guide rails are to be parallel to the drive spindle axis and to each other within ± 0.03mm, or better, over 0.5m of the outer length of the rails (The tighter limit is needed to allow for leverage effect between the torsion block slide, item 4, and the end of the spindle, item 40, see Section 4.3.1. for the means for attaining this accuracy).

The specimen is to be gripped at each end by means of two adaptors (Items 37 and 39, drawing TTM 9) acting through adaptor nuts (Item 39) as shown in Fig.24. Each adaptor is

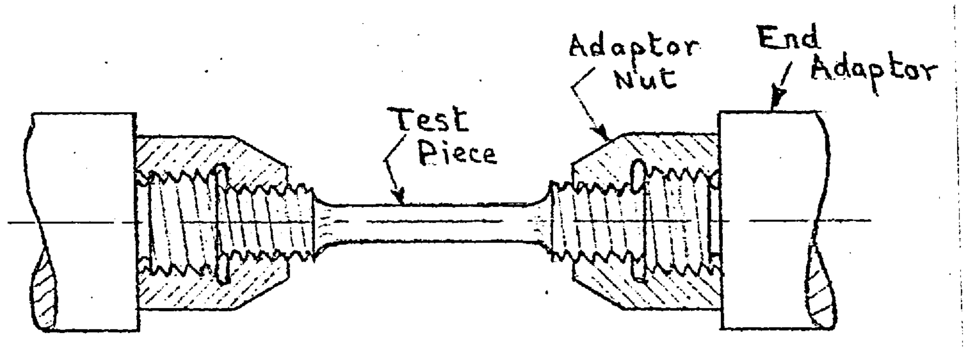


FIG. 24. Specimen grip system. Plain screwed end test piece.

esentially, a Morse taper sleeve with an accurately squared outer end, against which the specimen end thrusts, and, adjacent to the end, an accurately-threaded nipple (1in B.S.F. thread) for the adaptor nut which serves to adapt the sleeve nipple thread to the specimen thread. This system is adopted so that (i) the larger sleeve may be made of a cheaper heat resisting alloy than the adaptor nut and (ii) a change in specimen end size does not involve replacement of a massively costly part. The adaptor is made of Nimonic 75 and the nut of Nimonic 105, which are the highest-service-temperature, strongest, machinable, heat resisting materials readily available on the market (Incidentally these materials cost £70, £20 more than estimated (see Section 2.4.3.) owing to the firm's (Henry Wiggins) minimum charge levy for small quantities). The junctions between the adaptor nuts and the adaptor sleeves and the relevant spindles will act as minor heat barriers. It is appreciated that a small measure of accuracy is lost by using the dual adaptor system; but, machining is simplified, cost should have been reduced, cost of maintenance is lowered and, cost of change of specimen size is reduced.

It is not possible to keep high factors of safety in the adaptors at high testing temperatures in the close vicinity of the specimen. Indeed, there is great uncertainty regarding the real strengths of Nimonic 75 and 105 at temperatures over 1000°C, since even the 105 is not recommended for use at temperatures much over this. The shearing stress in the nipple end of item 35 is 20.9N/mm^2, at 46Nm torque, against a quoted shearing strength of 56N/mm^2, at 1000°C, so this is very near the minimum safe limit at this temperature. However, it should be reasonably safe if 23Nm torque is not exceeded. On the same basis the Nimonic 105, the stronger material (it's quoted strength at 1000°C is 112N/mm^2), should <u>just</u> be safe at 1250°C if 23Nm torque is not exceeded. Owing to the heat sink effect of the spindles, see Section 1.2.4, it is expected that the temperature of the Nimonic 75 will not, in general, exceed 1000°C when the Nimonic 105 is at, or near 1300°C.

One danger, which must be guarded against at elevated temperatures, is the risk of solid-phase welding between specimen and adaptor, or between adaptors. To prevent this, it is recommended that the adaptor contact surfaces should be at least partially oxidized before use and also coated with a suitable high temperature parting compound, such as Rocol "Anti-Stick" fluid spray.

Conversely, to prevent the Morse taper coming loose under any axial tension during a test, a spring retaining pin, item 38, is fitted near the end of each adaptor remote from the test specimen.

4.3.4. Setting-Up For a Test.

Preparation of the apparatus for a test is simple. The furnace is moved to its extreme right hand position, it is then uncoupled from the quenching chamber (by removing the locking pin (Item 10, drawing TTM 7) and the chamber is pushed as far as it will go to the left; the appropriate screwed adaptors can now be fitted to the spindle ends. The end-anchor bracket, item 5, is unclamped (if it was previously engaged) and the torsion-block assembly is pushed outwards, to permit insertion of the specimen, which may then be screwed right into whichever adaptor is most convenient. After this the torque measuring head may be carefully pushed back until contact is made between the free end of the specimen and the remaining adaptor. Gentle pressure on the measuring head combined with appropriate manual rotation of the drive spindle (a clear space is left on the spindle, between the bearings, for this purpose) can now be used to gain full engagement of the specimen to the point at which it is tight between the adaptor thrust faces. It only remains to return the quenching chamber to its original position, to lock it on to the furnace and then to clamp the end-anchor if it is to be used.

After setting the torque measuring equipment, setting the furnace in its heat position, and setting the furnace temperature control, the specimen may be heated preparatory to testing.

Removal of an unbroken specimen is achieved by reversing this procedure. The parts of a broken specimen are just as simple to remove, unless one end has jammed into the holder. If an end is firmly jammed it may be released by partially unscrewing the threaded adaptor, on which flats are provided for the purpose.

More details of sequence are given in

Appendix B.

4.4. Torque Resistance and Load Measurement.

To enable torque to be applied to a specimen the latter must be firmly anchored at its "outer" end; but, also, in this case, (a) the torque resistance must be measured with reasonable accuracy, (b) it must be possible to leave the specimen end axially unrestrained (That is, the torque-measuring system itself must be free to "float" in the axial direction) and (c) it must be possible to anchor the specimen's end axially and, if required measure the direction and magnitude of any end load that may be built up.

These aims are achieved (i) by mounting the anchor or torque shaft, item 40, with its specimen adaptor system, on a sliding saddle (torsion block, item 4) which is designed both to run on and be guided by ball bearings acting against the guide rails, ~~and~~ (ii) by mounting the torque-measuring lever, load cells etc. entirely on this saddle and (iii) by making provision for an independent end clamp stand with a load-cell link.

The general arrangement is as shown on drawing TTM 2.

4.4.1. Anchor Spindle and Specimen Fastening.

The adaptation of the specimen to the anchor shaft spindle is by the means outlined in Section 4.3.3. using items 37, 38 and 39. The anchor spindle or torsion shaft, item 40, drawing TTM 9, is of the same robust construction and material as the driven shaft, item 35, and is mounted in a similar way in a similar pair of taper roller bearings; but, primarily because of its purpose, there are differences.

Between the bearings is mounted the torque lever, item 41, drawing TTM 10. The torque from the spindle is taken through a single key, stressed in shearing to a maximum intensity of $15N/mm^2$ at 46Nm torque, acting within a closely fitting sleeve incorporated as the body of the lever. The lever is made up of low carbon steel (En3) sections, integrated by metal-arc welding, with two effective lever lengths of 100 and 250mm, respectively, and incorporating an accurately-balanced counterpoise weight to eliminate any out-of-balance torque from the lever's own weight. On the outer end of the shaft there is a small fork-end and a pivot pin for attaching, at will, the end-load measuring cell (for type and size see Section 3.3.2.) which, at its other end is pinned to item 5.

The shaft, item 40, is drilled out 6mm diameter co-axially from its inner end to a point beyond the torque lever where a radially drilled, 6mm diameter, hole and associated hose nipple meet the central hole to form an entry passage for a suitable gas to be passed into the heating chamber when the latter is in position. The adaptor, item 37 and adaptor nut, item 39, at the anchor end of the specimen are appropriately grooved and/or drilled to assist the passage of the gas. If a light flexible plastic gas supply pipe is used to feed in the gas, the pipe has an almost undetectable effect on the torque load measurement.

The mass of the torque shaft and lever, plus attachments, is about 7kg and the maximum frictional effect of this weight on the bearings, in terms of torque should be 0.175Nm, allowance for which can be made in the measurements of torque load, but the actual value should be checked experimentally at intervals, since its value depends to some extent on the accuracy of setting of the taper roller bearings, (Experimental value, at first setting, found to be 0.15Nm).

A groove is provided along the top side of the shaft on its projecting inner end, and along the top side of the relevant adaptor, to carry a pair of thermocouple wires, in their insulating sleeves, up to the specimen position. Such wires do not affect torque measurement if they are kept free to flex where they join the spindle near the bearing.

4.4.2. Torsional Load Measuring Saddle.

Item 4, drawing TTM 7, the torsion block, or torque-measuring saddle, is fabricated from low carbon steel plate by arc welding. It is made more strongly than simple strength requires to ensure that it remains undistorted when any normal working load is applied to it. Its main function is to act as a firm frame between (i) the bearing housings, (ii) the support and guide ball-bearings, and (iii) the load cell mountings. The principle of its operation is shown in Fig.25. At rest the

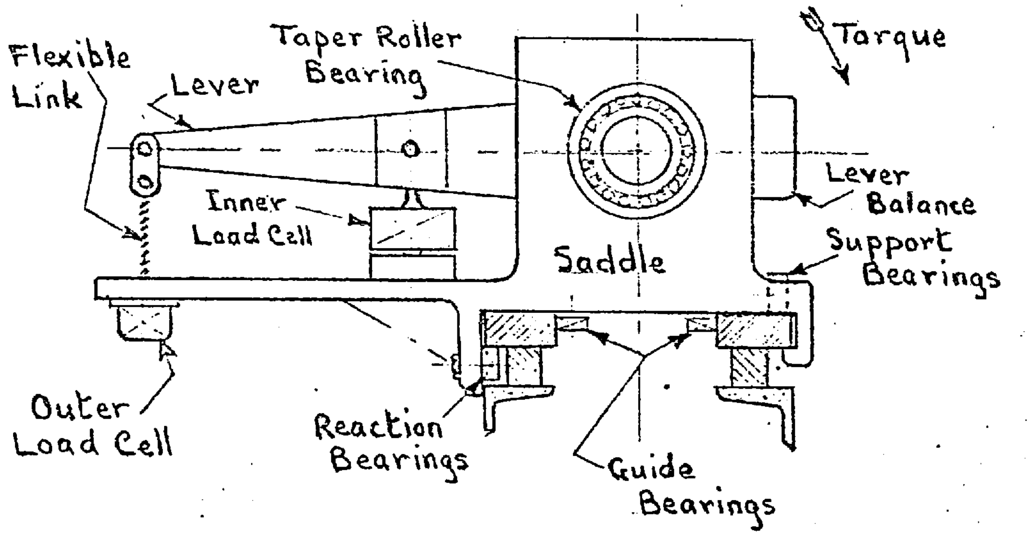


FIG. 25. Torque saddle block viewed from specimen side.

saddle sits down in face-to-face contact on the top of the rail in the left in the figure and at the right it rides on the two support ball bearings (Item 12) which themselves rest on the right hand rail. In this position the three reaction ball bearings (Item 12) are set by adjustment of the eccentric spindle (Item 47a) to be only just in contact with the under side of the left hand rail. The four guide ball bearings (Item 13) are adjusted relative to the inner sides of the rails,by means of eccentric pins (Item 47b), so that (a) the saddle is accurately and co-axially aligned with the driven spindle and rails, and (b) slight but equal positive contact is made between each bearing and its adjacent rail side face. This latter setting, in conjunction with the accurate set-up of the rails, discussed in Section 4.3.1, ensures accurate and quite free axial movement of the saddle. During a test the applied torque will tend to lift the left hand side of the saddle against the reaction ball bearings on its under side and press it a little more heavily down on the right hand support ball bearings,thus freeing the saddle from contact friction on the top of the left hand rail. (Note. This system was chosen to minimize the number of ball bearings and to keep manufacturing complexity to a minimum; but, while it proved satisfactory with slow heating up of the specimen, it applied too much off-load restraint to a rapidly-heating specimen, causing the latter to bend slightly. This restraining effect was overcome by supporting the left-hand side,in Fig. 25,on the rail-top through two additional,suitably placed,low friction adjustable PTFE pads, so reducing the axial no-load restraint to about 5N).

At a full 46 Nm torque the torque thrust on each rail (up on the left hand side and down on the right) is 218N plus the downward weight thrust of the saddle of 130N on the right side only. In the fully floating position, under this maximum torsion, the total frictional end load is not expected to exceed 30N.

The saddle is adapted to take a Coutant 128 and a Coutant ER60 load cell, respectively at the maximum and minimum leverage positions, and is fitted with individual adjustment and connection for each adaptation. There is also an adjustable limiting and/or locking attachment, items 45 and 46, drawing TTM 10, at a suitable position over the lever arm for use to prevent overloading of a cell during operation of a test and to prevent accidental damage to a cell during setting-up etc., with the cells connected up in position. Electrical connection to the cells is by loose-hanging screened cables which do not interfere with load measurement.

4.4.3. Torsional Load Measuring System.

As outlined, in Section 3.3.1. and immediately above, two load cells are incorporated for measuring torsional forces; namely a Coutant 128 for loads up to 45N at 250mm radius and a Coutant ER60 for loads up to 670N at 100mm radius. That is, torque ranges of 0-11.25Nm and 0-67Nm are covered, although it is not intended that the latter should be used beyond 46Nm.

The cells remain "permanently" in position, anchored to the saddle; but, at any given time, only one is connected to the lever by inserting its connecting pin into the linkage.

The smaller cell is connected rigidly to the underside of the saddle by suitable screws and is connected to the lever through a flexible cable and linking fork, item 42, drawing TTM 9. The cable lies free and clear of the lever when it is disconnected. The larger cell sits on an adjustable resting collar, item 51, drawing TTM 10, when not in use (set to prevent it tipping over and jamming the lever) and remains linked to the saddle below by a swivelling fork, item 49, drawing TTM 10, and pin, item 43b, drawing TTM 9; it is connected into a slot in the lever through its eye and a connecting pin, item 43a.

4.4.4. End Load Control and Measurement.

At the present stage only two things may be done on the machine with respect to end load; (a) the end may be left free, and (b) the end may be fixed. The first requires no special provision, but the second needs an adjustable swivelling connection between the torque shaft, item 40, and the anchor block, item 5. This connection is provided by a fork, item 70, drawing TTM 11, which may be twisted to a suitable position in item 5 and then clamped in place. Connection is completed through a Coutant ER60 load cell and two pins, item 43a.

To disconnect the end fixing, the pin between the cell and the shaft is removed and the cell is left resting on the support bracket, supplied for the purpose on item 5.

When the end load cell is in use the torsional flexibility in its connecting links and pins is sufficient to accommodate the very small angular displacement of the torsion lever, as it deflects the operative torsional load cell, without spoiling the accuracy of either torsional load measurement or end load measurement.

4.5. Deflection and Vibration.

To ensure accuracy and consistency in applying torsional strains and to obtain accurate load values, after making the system itself accurate, it is important to eliminate all external potential sources of variation and error. Probably, the most likely sources of error are deflection and vibration in the machine.

Wherever force is applied deflection must take place and all that can be done is, (i) to keep the deflections to the lowest sensibly-possible magnitude, and (ii) control their nature to maintain concentricity of behaviour and alignment. Because some critical sources of trouble are outside the designer's control there is also a limit to what can be done to ensure absence of vibration.

In the latter context, control may be exercised over the sources likely to operate from the actual mechanism; but, if the stimulation comes from irregular structural behaviour of the specimen, say from "sawtooth" or "see-saw" plastic yielding, then all that the designer can hope to do is to try and allow for its occurence and to absorb it. Two approaches may be used: (a) to keep the natural resonating frequencies in the machine very high so that resonance from the low-frequency sources of the machine and from the, probably, low-frequency excitation of uneven yielding of the specimen is unlikely, and (b) keep the moving parts of the machine that act directly on the specimen relatively massive so that they will absorb high-frequency changes in movement and damp them out.

Fortunately, both ends can be approached simultaneously, without either too much difficulty or with too adverse effects on the final appearance and the cost of construction, by designing the component parts to be inherently rigid and relatively massive and this approach is used in the present application.

4.5.1. Likely Causes of Deflection.

There are five likely causes of deflection in a mechanism such as a torsion testing machine.

(i) The torsional forces.

(ii) The dead-weight of structure and equipment and, in particular, of mobile parts.

(iii) Axial forces set up by torsional straining of specimens.

(iv) Inefficient integration of the machine assembly.

(i) Torsional forces begin in the main frame at the power source, an electric motor in this instance, and transmit right through the drive train, thence to the specimen and finally back to the frame through the torsional-load measuring system. At the motor there is the basic torsion, arising from reaction between the motor's magnetic coils and the squirrel cage armature, which transmits itself through the variable-speed gearbox and forward into the drive train by reaction against the power-pack frame through the gear case. The reaction resolves itself into a torsional couple plus a direct thrust reaction, see Fig.26. In this case the

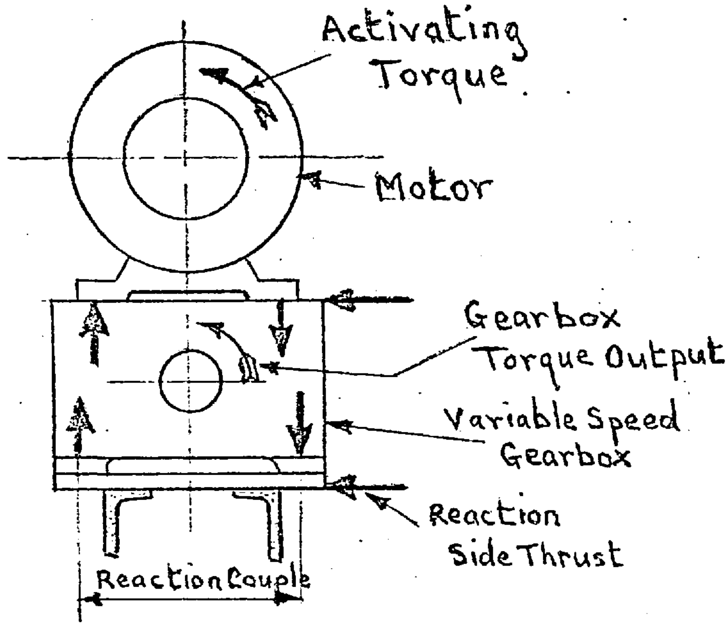


FIG. 26. Reaction couple and side reaction from the motor drive end of the power pack frame.

couple, which has a maximum moment of 23Nm at the gearbox output, reacts on the channelled side members of the power-pack frame. That is, there is a compressive downward force of 109N on the left-hand channel, an equal vertical pull on the right-hand channel and an equally-shared total side thrust of 135N acting towards the right in the top flanges of the channels.

The torsional force in the drive system has no further effect on the frame until a speed or direction change is attempted (In this context the final "anchoring" of the specimen is a speed change to zero). There is a possible speed change situation just before the direction-changing gearbox and, of course, a direction change at the direction-changing gearbox, each of which may react on the power-pack frame. Neglecting speed change for the moment, the direction change will set up a couple and side thrust in the end of the frame through the direction-changing gearbox the directions of which are determined by the particular direction of subsequent transmission. In the present case with the single torsion head, there is a downward thrust on the input shaft bearing and an upward thrust on the output shaft bearing. The force in each case being 524N at 23Nm torque and 1047N at 46Nm torque, respectively. The side thrusts will be horizontally to the left at the input side, at the frame surface, and their magnitudes will be 135N and 278N, respectively. This will be associated with an equal side thrust on the output bearing, pushing back parallel to the input shaft.

The difference in magnitude between the primary side thrust at 23Nm torque and that at the direction-changing input shaft at 46Nm represents the side thrust that would be set up at a speed-reducing gear used to increase the torque from 23Nm.

In a similar manner the torque and reactions acting in the test bed frame, item 1, Part 2, will operate as indicated in Fig.27 with the drive end rigidly held by the power pack frame and the maximum equivalent length depending on the extreme position of the torque-measuring head, item 4 etc. The maximum equivalent length (L) is unlikely ever to exceed 1100mm. These forces, in particular, are the ones most likely to cause harmful deflection in the torque-applying mechanism; therefore their effects are considered here in a little more detail.

The torsional rigidity of the test bed frame is very difficult to calculate because it depends on a number of variable or ill-defined effects, such as the stiffening influence

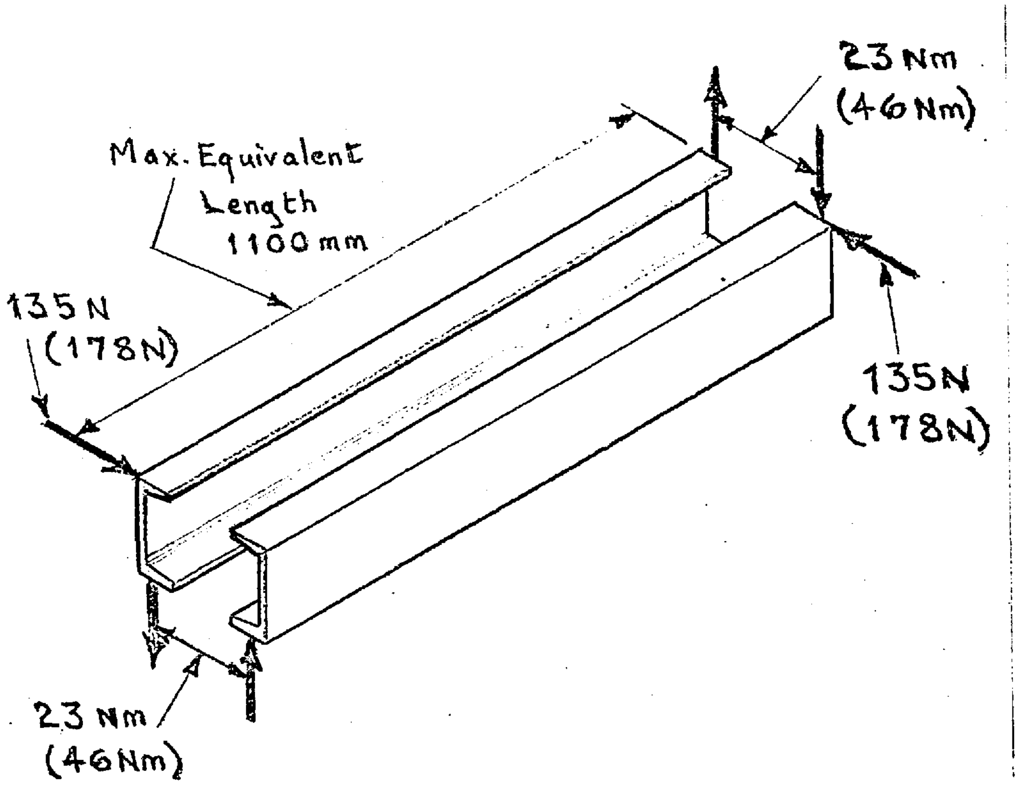


FIG. 27. The forces acting on the test bed frame. Values in brackets are those set up when max. high torque loading is applied.

of the superimposed guide rails (Items 6 and 7), the influence of cross-bracing from the webs, support members etc., and incidental bracing derived from the design and mode of fastening of the frame stand. However, if the extreme case is represented by considering the channel members as if they are acting in isolation from these other influences and if the deflection situation, then given, is acceptable, then the true state of affairs will obviously be much better.

It will be noted that the side thrust is acting on the tops of the channel members and tending to twist them in the opposite direction to the basic couple; thus the normal maximum resultant torque is:-

$$23 - (135 \times 0.05)\text{Nm}$$

$$= 15.55\text{Nm, clockwise}$$

and the occasional maximum resultant torque is:-

$$46 - (306 \times 0.05)\text{Nm}$$

$$= 30.7\text{Nm, anti-clockwise}$$

The polar second moment of area (J) of the pair of channels is:-

$$J = 47.04 \times 10^6 \text{ mm}^4$$

Hence (from $\theta = \frac{TL}{GJ}$, where θ is the angle of twist in radians, G is the shear modulus - $84.9 \times 10^3 N/mm^2$, L is the axial length - 1100mm, and T is the torque) the respective maximum angles of elastic twist are 4.28×10^{-6} and 8.46×10^{-6} radians.

These angles give relative lateral horizontal axial displacements of 0.00096 and 0.00~~0~~19mm, respectively, at the drive axis over the 1100mm length but these are reduced slightly to 0.00095 and 0.001893mm by the simultaneous bending moments (24.03Nm and 54.5Nm) caused by the respective side thrusts.

Even the maximum of these deflections is safely acceptable, therefore the design is completely safe from this aspect.

(ii) Dead-weight forces, under normal circumstances, should cause no significant deflections in critical parts of the machine structure.

If it is assumed that both manufacture and installation are correctly effected, the only dead-weight problem will arise from the mass of the torque-measuring head acting downwards on the test frame rails. However, the test frame item 1, Part 2, is adequately and firmly supported at four points, with a relatively short unsupported span of 525mm, located below the normal position of the torque measuring head, giving an absolute maximum deflection of 0.03mm between the drive and anchor axes. In general, due to the overhang of the anchor spindle, item 36, and the likely, slight, uptilt of that overhang due to the sag from the bending deflection, see below, which tends to cancel the direct displacement, the true deflection at the specimen position will be less, see Fig.28. (i.e. the total deflection effect is likely to result in a very slight, negligible, angular bending in

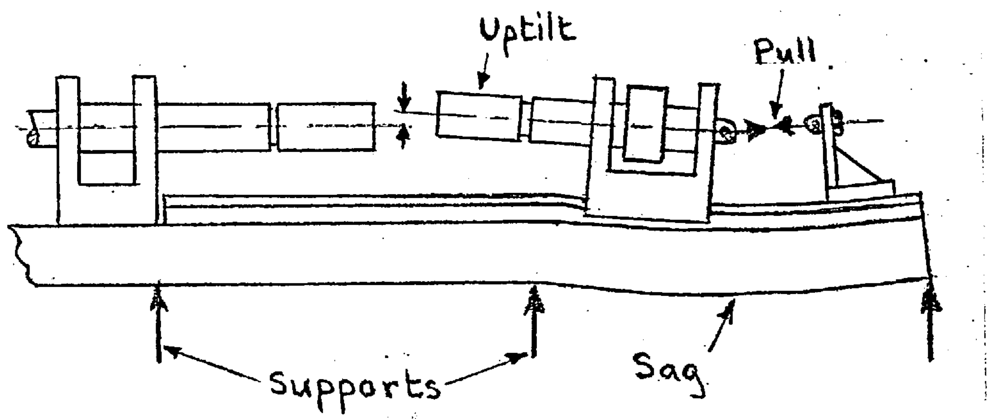


FIG. 28. Deflection in the test bed frame caused by the dead weight of the torque-measuring system and the axial pull from a contracting specimen.

the specimen rather than a more severe local eccentricity of alignment and this effect will be aided by the essential, although small, clearances that have to be left on the torque-measuring saddle guides). If specimen length increases during a test, deflecting efforts could be additive, as mentioned below in relation to axial forces.

(iii) Axial forces, set up by plastic torsional strain in a test specimen, will not normally be transmitted to the machine structure if the torque-measuring saddle is axially unrestrained. On the other hand, if end-restraint is applied, a cantilever moment in the appropriate direction will transmit to the frame, see Fig.28. The likely magnitude of the activating force is completely unknown but may be as much as 500N, as suggested in Section 3.3.2, and this may impose a significant bending moment of 162Nm in the central vertical plane of the frame. Using item 5 as designed, this moment could give a deflection of 0.0014mm in the end segment of the frame, additive to the dead-weight deflection (see above) if the axial force is contractive and subtractive if the force is expansive. These amounts of deflection might just be sufficient to cause some uncertainty concerning the exact plastic behaviour of the specimens which generate them; torsional load-measuring accuracy may fall off by a relatively small unpredictable amount when this situation arises and some allowance may have to be made for it in assessing results. After making some completely arbitrary assumptions with respect to the likely flexibility within the equipment, it is the writer's opinion that this particular error could not possibly exceed $\pm 1\%$, but it could be additive to other sources of error. If work of similar accuracy to the planned normal torsional work is to be done, whilst restraining the axial end movement of the specimen, the end anchorage system might have to be modified a little to reduce the amount of bending deflection it causes.

(iv) Inefficient integration of the machine assembly is always a possibility, particularly if grips etc. have to be changed frequently; but any distortion that is caused can readily be detected if a procedure of frequent, simple, alignment testing is performed. Provided that test pieces are accurately made, they may themselves make a useful test of alignment. Any such test piece that, for no apparent reason, proves difficult to insert in the machine should be treated as an indicator of misalignment and a proper alignment test should be made immediately. If these simple precautions are taken it is unlikely that deflection caused by inefficient assembly will escape undetected.

4.5.2. Likely Causes of Vibration.

There are not many sources of vibration in the system and those that are present tend to be of low frequency.

The prime source of vibration is the electric motor. Very few electric motors are perfectly free of vibration, which is most commonly transverse in nature, caused by very slight out-of-balance masses in the rotor. This vibration at the frequency of rotation, about 23.6Hz, transmits to the frame of the motor through the bearings and thence to adjacent parts with which the frames may be connected (in this case the hydraulic variable-speed gear and thence to the power-pack frame). This source should not result in significant vibration in the torque drive because such vibration should not transmit through a V-belt drive even if it is significantly present in the form of torque variation. Provided that it does not set up any serious resonant vibrations in critical parts of the machine, vibration from the motor should do no harm and could be positively helpful in reducing or eliminating uneven frictional behaviour in sliding and rotating parts (e.g. in the load measuring head) during a test by reducing "stiction". (Note. It was found subsequently that motor vibration did tend to interfere with the recording of load readings from the low-range load cell and a change in the drive system as indicated in drawing TTM 13, had to be made to offset this).

Another possible source of vibration exists in the transmission system, in the direction-changing gearbox, due to vibration from the gear teeth, as already mentioned in Section 4.2.5. This source, although of low energy, can be more directly serious than the motor vibration, with respect to actual test conditions, if it is allowed to set up resonant vibration. The frequency of excitation varies with the speed of rotation and, since there are 28 teeth it would be 28 times the revolutions per second. Hence in normal testing the frequency could vary from 0 up to 454Hz.

This excitation could react in two ways:-

(i) It could resonate with the natural torsional vibrations of the transmission system.

(ii) By its reaction through the bearings it could resonate with the natural transverse bending vibrations of machine frame and thence of the drive train.

The results of calculations of the basic values of natural frequencies are given in Appendix C and the following conclusions can be drawn from these.

(a) Resonant torsional vibration is unlikely because the maximum excitation frequency is well below the minimum resonating frequency.

(b) Transverse vibration in the drive train is unlikely because its minimum frequency is high and damping from the integration system is high (e.g. in the system of interlocking of items 37, 39 and 40).

(c) Transverse vibration in many of the machine frame members is unlikely because the minimum resonant frequencies are too high. In the other members, where resonance might be possible, either the part is too remote from the source of excitation to be likely to resonate or it is supported in the construction in such a manner that either heavy damping is present or the natural frequency is modified to a safe level.

There are two other possible causes of vibration, each of which arises from the deformation of the test specimen.

If the alignment between the two adaptors, which grip the specimen, is not reasonably correct, a side thrust will be generated in the specimen and, if there is any excessive freedom of lateral movement, in items 35, 37, 39 or 40, the gauge length of the specimen could start to "hunt", by buckling and twisting on itself, at a frequency equal to the frequency of rotation; thus transmitting to the system transverse vibration at the same frequency. This is not likely to resonate any part of the machine system but it could seriously upset both the consistency of straining and the accuracy of torsional load measurement. Alignment should be kept better than ±0.05mm, for which the maximum side thrust would be 82N at 23Nm torque, see Section 4.3.3. This side load would react on each of the relevant guide roller bearings, item 13, with a maximum force of 328N, therefore, these guide roller bearings must be set with no freedom for either side movement or vertical movement so that any oscillations from this source are effectively absorbed.

The last possible source of vibration may arise from the inherent behaviour of the material under test. If the material work-hardens or yields in "bursts", in "see-saw" straining, it may generate a vibration which might cause resonance. On the whole it is unlikely that serious resonance could occur from this cause, since the exciting frequency would almost certainly be too low and the specimen, in itself, should provide a damping mechanism sufficient to prevent transmission into machine vibration. This source is, therefore, disregarded.

4.6. Heating and Cooling System.

If metallurgical studies of internal structure are to be made of material after it has undergone plastic deformation at elevated temperature, it is essential that the temperature should be brought down to a metastable level as rapidly as possible. That is, immediately a straining test is completed the specimen should be quenched. These basic requirements and the cost limitations upon them are outlined in the S~~e~~ection 3.4. and Sub-section 3.5.2.

A radiant-resistor tube furnace is to be used and is to have facilities for rapid axial movement of the furnace over such a distance that its place can be taken by a quenching system suitable for effective quenching of the specimen. It is a logical step to couple the furnace to the quenching chamber in such a way that they may be separable and moveable independently or may be made to move simultaneously during the automatic quenching sequence.

There are, therefore, three interdependent aspects to this part of the design:- (a) furnace and quenching chamber movement, (b) furnace design and (c) quenching chamber design.

4.6.1. Furnace and Quenching Chamber Movement.

Several alternative systems might be used to cause movement of the furnace and quenching chamber assembly, including (i) manual power, (ii) gravitational force from a counterpoise weight, (iii) hydraulic power, (iv) electrical power and (v) pneumatic power.

Of these the first two are too slow and uncertain, both in beginning to operate and in performance; the third is too messy and slow in operation and the fourth is too complex and inflexible, if cost is to be limited. Taken overall, pneumatic power is likely to be the most efficient and adaptable, within the allowable expenditure. Other workers have also found this kind of system satisfactory (21).

Pneumatic power is simple to instal and an air supply at 45p.s.i. is available. (The latter point is not very important because the volume of air used is not likely to be great and an air bottle supply could have been used through a regulating,reducing,valve).

Activation of the system is to be electronic, therefore,there must be an air pressure switch interlock on this part of the system. This is fitted in the side of the airline that comes into operation to move the furnace into the heating position, see drawing TTM 1.

A standard 50mm diameter air cylinder and piston, with a stroke of 205mm and cushioned stroke ends (A. Schrader's Son Ltd.,), is used since this gives a potential thrust of 610N at 45p.s.i. sufficient to give the furnace and quenching chamber assembly (mass approximately 10kg) an acceleration of about 6.2m/sec^2. Each end of the air cylinder is fitted with an air speed control valve, by which speed of movement in either direction may be adjusted; so that the power stroke speed may be adjusted to a suitable value (6.2m/sec^2 acceleration is high for most purposes, since the full stroke is then likely to be completed in 1$\frac{1}{4}$s.) and the return stroke, for safety reasons, must be at a suitably slow speed. This arrangement may entail a compromise between the speeds, because, for cheapness and simplicity, a 4-way, soleniod-operated, balanced-poppet valve is the air activation source and the respective two cylinder end openings serve alternatively as entry and exhaust port according to direction of stroke. Direction of movement is controlled from the balanced-poppet valve by an electrical pulse to the appropriate solenoid which moves the poppet into the required position and leaves it there. There is, probably, a delay of about $\frac{1}{4}$s before full air pressure builds up in the cylinder end, thus giving a possible minimum complete working stroke time of about 1$\frac{1}{2}$s. The working stroke is activated through the "quench" button operator (for location see item 1, Part 2, drawing TTM 5) and the return stroke through the "furnace posit$^{n.}$ heat" button operator, see drawing TTM 3.

The cylinder is bolted in suspension below the guide rails, on a flange extension provided on the countershaft block, item 3, drawing TTM 6, and the piston rod end is pinned to the bottom drive arm, provided on the furnace base, item 8, drawing TTM 6.

4.6.2. Furnace Design.

A basic, tubular, radiant-resistor construction is specified for the furnace, see Section 3.4.1. The clearance of bore of the tube must be 50mm minimum, to clear the outside diameter (46mm) of the shafts and specimen adaptors items 35, 37 and 39. A silica tube, suitably wound on the outside with nickel-chromium resistor wire, would suffice for many heating purposes, but the maximum temperature attainable would probably not exceed 1000°C and it would need a long time to reach a stable temperature, both because the nickel-chromium elements cannot be heated much above 1100°C and because the heat losses are high, as commented on in Section 2.2.1. However, Kanthal-wound furnace tubes are available in certain standard sizes. One, in particular, of 3kW rating with a minimum inside diameter of 70mm and a tube length of about 200mm, would offer the possibility of attaining temperatures up to 1300°C, although admittedly with some difficulty. These Kanthal furnace tubes have their windings suspended inside the bore; therefore, because direct radiant heating is possible, they are more efficient than the normal externally-wound nickel-chromium type. Hence, heating efficiency should be higher and heat losses lower than with Ni. Cr. windings, even although the radiation gap is increased by 10mm on each side, beyond that width of gap which a conventionally wound tube bore would give. This improvement will be achieved, provided that the extra exposed areas at the tube ends are suitably enclosed, in spite of the fact that the windings cannot be locally concentrated at the tube ends to offset the local temperature drops caused by the rapid loss of heat outwards through the shaft ends.

One disadvantage is that the cross-sectional area of the element strip is large and its resistance is low, therefore currents up to 30A are needed. This may entail the use of a suitable 3kW, low voltage output, transformer to step down the mains voltage; but it should be possible to provide this from Departmental equipment.

Taking these factors into account, a Kanthal type REH-S-B 7-30 tube element, giving a heated zone of length 175mm, is adopted (see item 56(a), drawing TTM 10). To seal the end opening as far as possible, this tube is used in conjunction with two end spacing and centreing rings (Items 56b), suitably made up from suitable alumina cement, cast in a specially-made wooden former. The rings are also designed to accommodate expansion and contraction of the tube element. The element is encased in a 6mm thick, 205mm diameter bore, asbestos tube, item 55, clamped between suitably braced Sindanyo ends, item 53, with the heat insulating space (52mm on each side) inside the asbestos tube packed with shaped, heat resisting heat-insulating, foamed-slag bricks. A junction box is attached, outside the case, and a thimble tube, item 58, is provided so that the furnace-control thermocouple may be located close to the windings. This assembly is mounted on the guided furnace base, item 8, drawing TTM 6, leaving a 12mm airgap between the casing side and the base. The furnace base slides on its plane, under side, face against the tops of the guide rails, item 7, drawing TTM 7, and is retained on them and guided against their two outer sides by the guide ribs, item 11a. There is an axial projecting tongue on the end of the base, appropriate for interlocking with a matching recess in the quenching chamber base, item 9, drawing TTM 8, and a driving arm projects below the base through the gap between the guide rails with its end aligned with, and pin-jointed to the piston rod end as it projects from the air cylinder.

Power and thermocouple connections are made through insulated flexible wire connecting links hanging in a loop, in the clear space provided for the purpose, behind the guide rails, whence they are led up to the control panel, which is part of the back of item 1, Part 2, drawings TTM 5 and 6.

A controlled atmosphere may be maintained in the furnace by feeding in a suitable gas, in appropriate quantities, by way of the gas passage in item 40.

4.6.3. Quenching Chamber Design.

The quenching chamber is made up from a Perspex tube (Item 63), which incorporates a suitable, tangential, drain exit, clamped between brass ends, items 61 and 62, drawing TTM 11, into which it is sealed by O-rings, all mounted on the sliding base, item 9. The base slides on, and is guided against, the guide rails, item 7, by means of its flat base and guide ribs, item 11b. A matching socket is provided to receive the interlocking tongue on the furnace base end, which is retained in position by means of the locking pin, item 10. A microswitch, in the quenching chamber base, senses the presence of the furnace tongue in the correct position and completes the return line of the quenching circuit, which cannot operate unless this switch is closed.

Water can be introduced through a pipe at the back of the chamber-end, item 61, through a free hanging loop of 12mm dia. bore flexible hose. The water distributes to two parallel jet blocks, item 67, drawing TTM 11, which are set one above the specimen, and one equidistant below the specimen each directed towards it from a position parallel to it, lying along the vertical plane of its axis. The end jets overlap the adaptor nuts, which have to be cooled at the same time as the specimen, if the specimen is not to reheat from them. A guide plate, item 68, is fitted transversely in the chamber to steady the jet blocks and to guide the surplus water into the drain hole, from which the water may then be led, through a large-bore, freely-hanging, flexible tube into any suitably adjacent drain.

The cooling water is admitted through an Ether, solenoid-operated, valve, which may be activated by a suitably positioned microswitch when the chamber is in the quench position, after having been moved there by the furnace. This same microswitch, simultaneously, cuts off the furnace heating current so that the furnace does not overheat the torsion shaft, over which it is then sitting.

Escape of water past the specimen adaptors, during quenching, is prevented (i) by the controlled directionality of the jets, (ii) by the smooth rapid escape of water, out to the drain, preventing a rise in level in the chamber, (iii) by the axially-long, small, annular spaces through which such escape has to take place and (iv) by the entrapping groove inset into the chamber sleeves (Item 66, drawing TTM 11), within the annular space.

The minimum effectiveness of the proposed jet system was verified by means of an experimental mock-up, before finalizing the design.

4.6.4. Safety Cover and Interlock.

An important part of the design, from the aspect of operating safety, is the safety cover, item 79, drawing TTM 12, and its interlocking safety catch system (Items 81, 82 etc.).

When the quenching sequence, outlined in Section 4.6.3, is operated, the furnace assembly (see Section 4.6.2.) and quenching chamber assembly move axially to the right, in unison, over a distance of about 205mm, in approximately $1\frac{1}{4}$ seconds, with considerable force; thus creating a situation of some danger to the operator if he gets in the way.

In addition, the furnace case will get very hot when higher temperatures of test are in use, possibly reaching 200°C at the maximum test temperature of 1250°C, creating a very real risk of skin-burning to the operator.

Taking these dangers together, a simple way out is to shroud the system in such a way that the operator is kept clear of the danger zone in normal working conditions and endangers himself only if he does something extremely foolish. On the other hand, it is desirable that the operator should (i) be able to see what is going on, ~~and~~ (ii) be able to remove the cover easily, for access when setting and dismantling, and (iii) be unable to remove the cover when the dangers are present.

These ends are attained by using a metal-framed cover, item 79, so designed with a transparent plastic window, item 80, that there is a large clear area between the operator's position and the test area, permitting an unimpeded view of the furnace, quenching chamber etc. The frame is so fitted that a simple forward pull, and slight lift, will unhook it from its front pivot sockets, item 86, and free it from the machine. Because item 81 incorporates a microswitch (see drawing TTM 1) interlinked with the quench system, through the interlock microswitch mentioned in Section 4.6.3, the two projecting arms at the back of the frame must engage the safety catches, items 81 and 82, and be locked properly in position by operating the clamp axle, item 83, before the quench system can be activated.

When other systems of heating, which do not require a separate quenching chamber (e.g. h.f. induction heating), are in use then the safety cover need not be used, but the safety cover microswitch will then need to be shorted out if, the turning on of quenching water is to be controlled electrically.

4.7. Control and Operation.

Individual details of certain stages of manipulation and operation of the machine have been given, incidentally, in the text when considered necessary ~~by~~ in describing the design approach to the relevant parts. Most other details of operation are a matter of common sense in the light of common practice; hence, it is thought unnecessary to provide a detailed commentary on operational procedure in the present instance.

However, an operator's guide to operational sequence has been prepared and a copy is reproduced in Appendix B.

------ oOo ------

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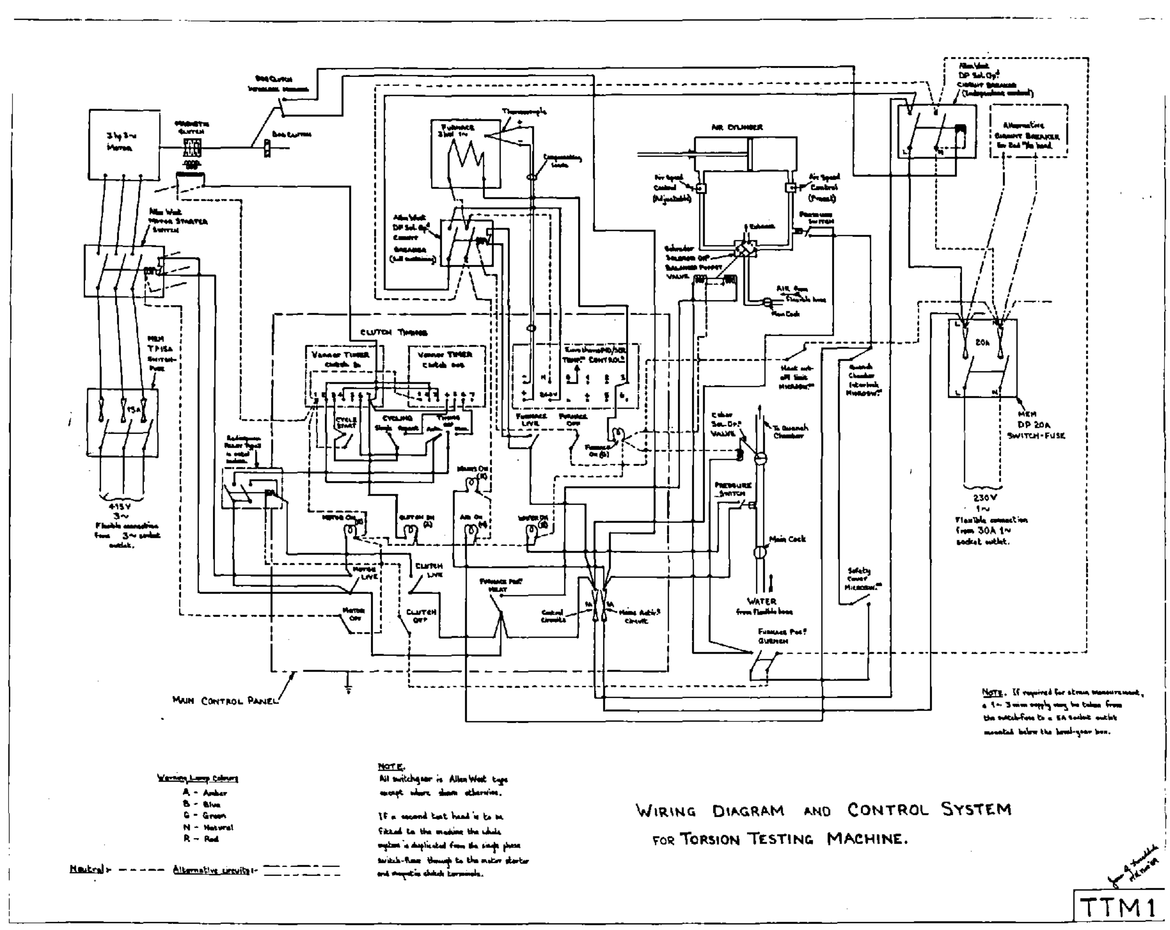
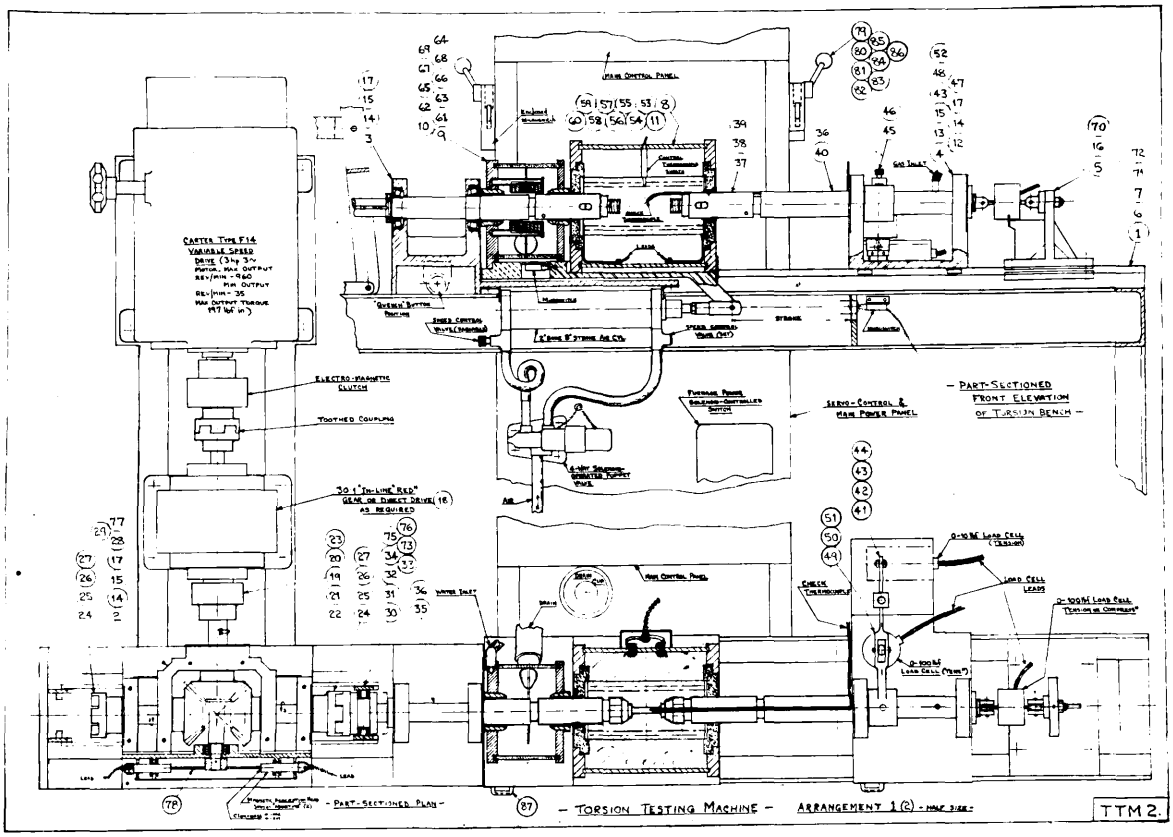
APPENDIX A.

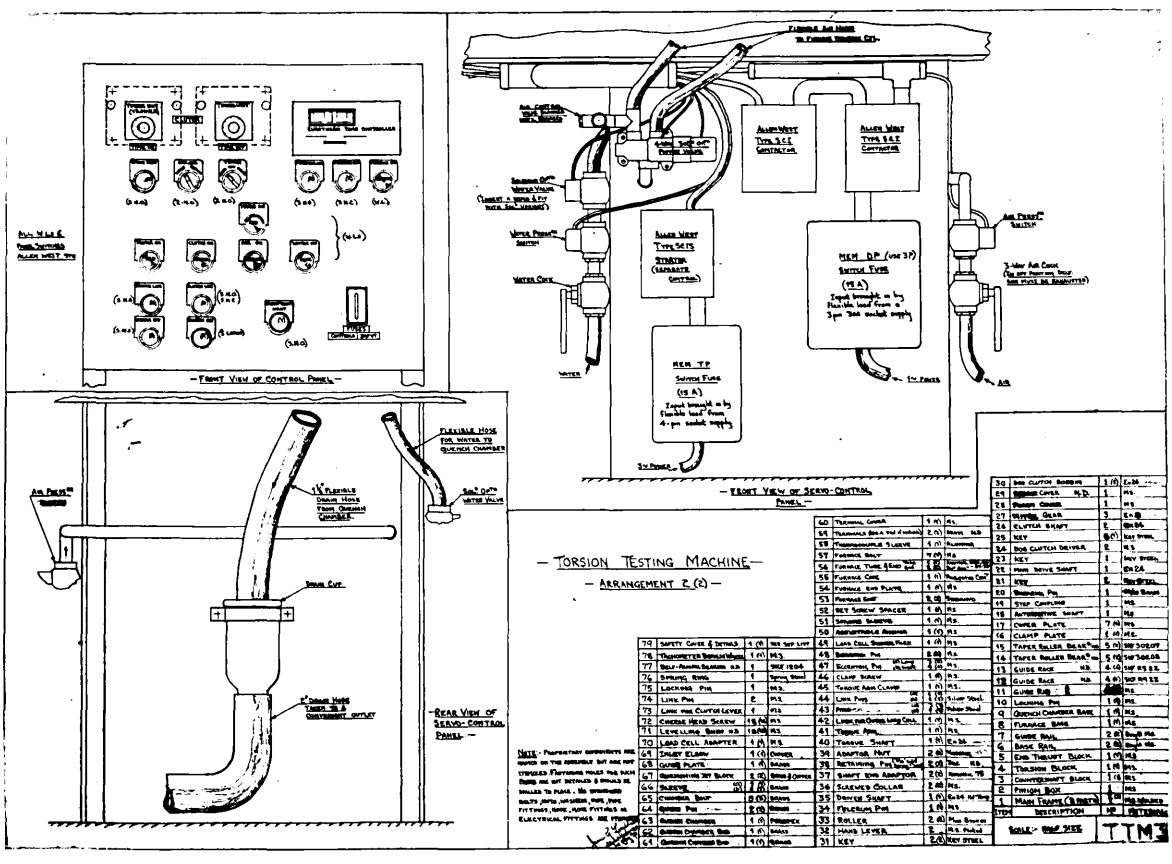
ASSEMBLY AND DETAIL DRAWINGS OF TORSION TESTING MACHINE.

Drawings TTM 1 - 13 photographically reproduced to 1/5th size.

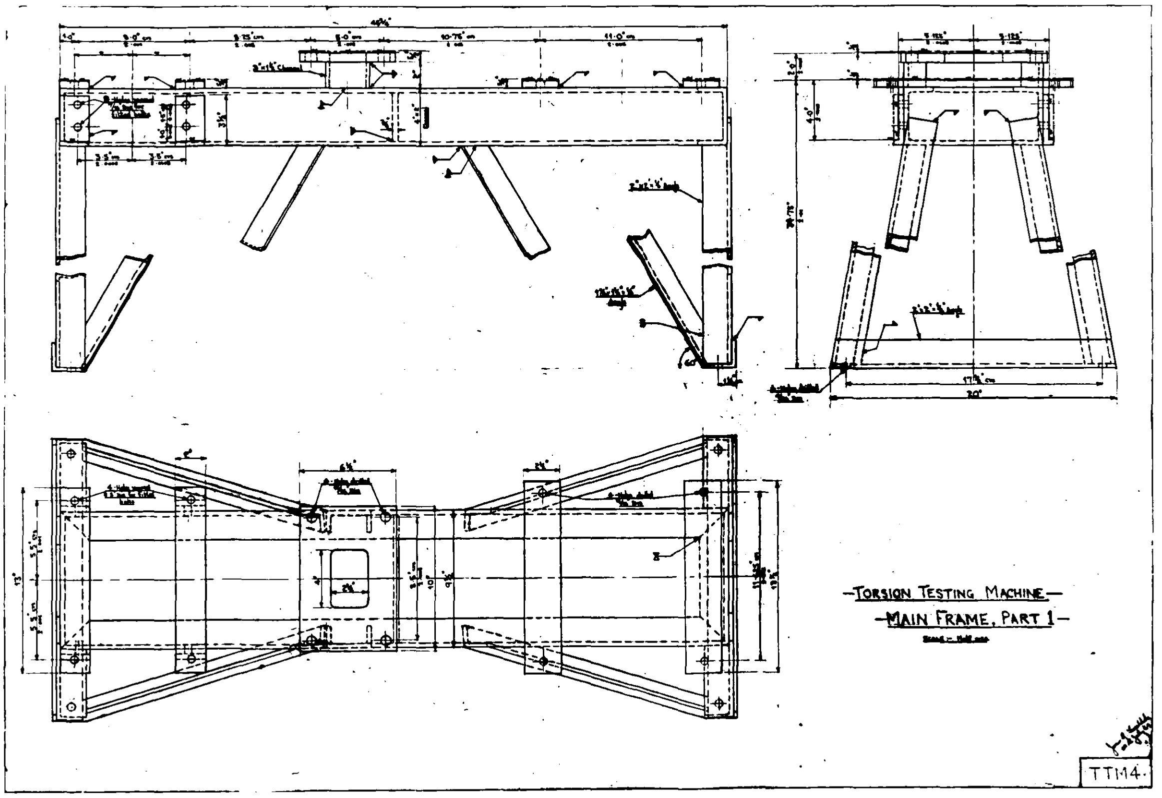
Note. Dimensions are in English measure because the Departmental machine shop has not yet switched to metric measure.



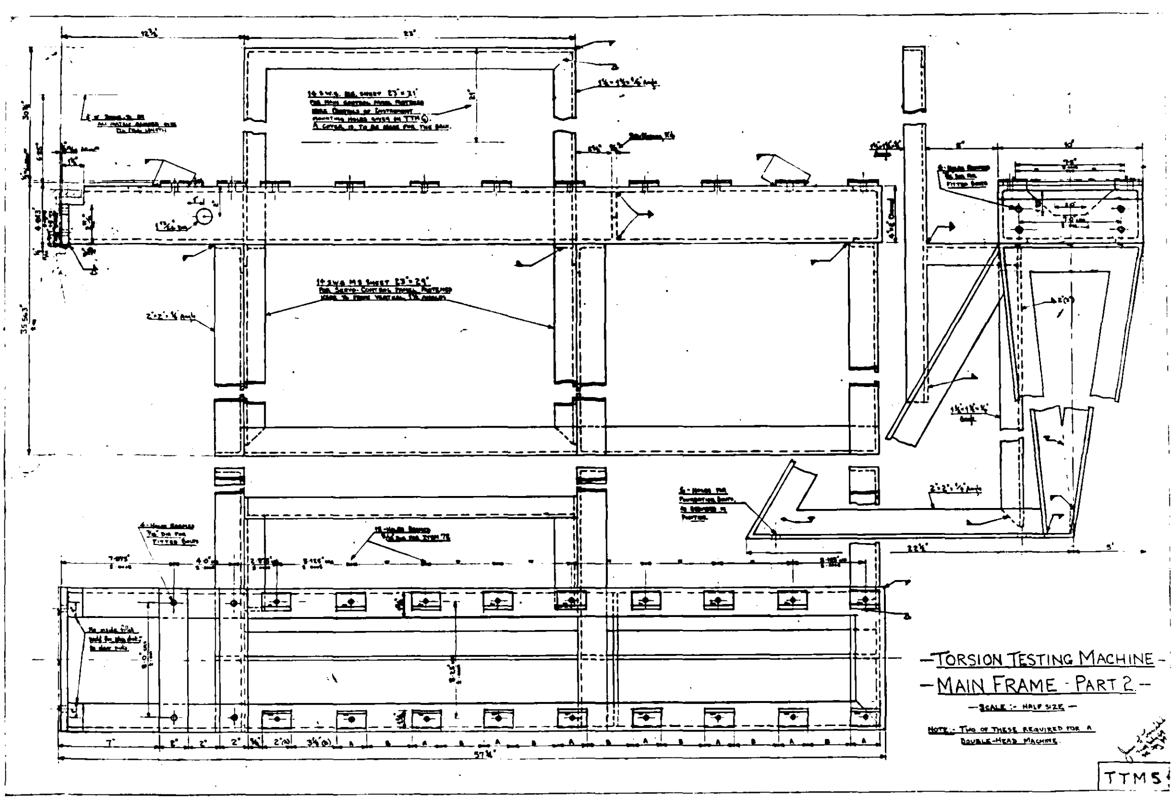




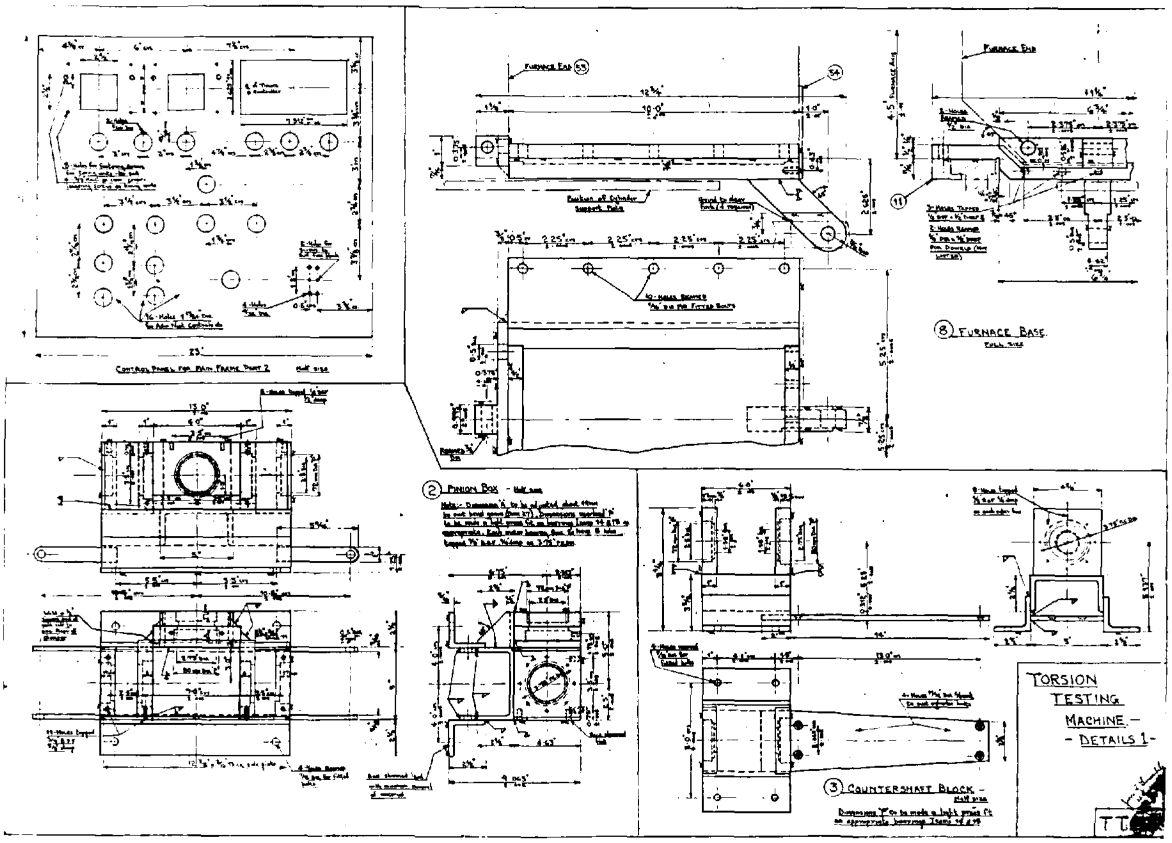
Item	Description	No.	Material
79	Safety Cover & Details	1 (1)	See Sep List
78	Tachometer [illegible] Wheel	1 (1)	M.S.
77	Self-Aligning Bearing N.D.	1	SKF 1204
76	Spring Ring	1	Spring Steel
75	Locking Pin	1	M.S.
74	Link Pin	2	M.S.
73	Link for Clutch Lever	1	M.S.
72	Cheese Head Screw	[illegible]	M.S.
71	Levelling Shim N.D.	[illegible]	M.S.
70	Load Cell Adapter	1 (1)	M.S.
69	Inlet Elbow	1 (1)	Copper
68	Guide Plate	1 (1)	Brass
67	Quenching Jet Block	2 (2)	Brass & Copper
66	Sleeve	[illegible]	Brass
65	Chamber Bolt	5 (5)	Brass
64	[illegible] Pin	2 (2)	Brass
63	Quench Chamber	1 (1)	Perspex
62	Quench Chamber End	1 (1)	Brass
61	Quench Chamber End	1 (1)	[illegible]
60	Terminal Cover	1 (1)	M.S.
59	Terminals	2 (2)	Brass N.D.
58	Thermocouple Sleeve	1 (1)	Alumina
57	Furnace Bolt	[illegible]	M.S.
56	Furnace Tube & End [illegible]	[illegible]	[illegible]
55	Furnace Case	1 (1)	[illegible]
54	Furnace End Plate	1 (1)	M.S.
53	Furnace [illegible]	2 (2)	[illegible]
52	Set Screw Spacer	1 (1)	M.S.
51	[illegible] Sleeve	1 (1)	M.S.
50	Adjustable Anchor	1 (1)	M.S.
49	Load Cell [illegible] Fork	1 (1)	M.S.
48	[illegible] Pin	2 (2)	M.S.
47	Eccentric Pin	[illegible]	M.S.
46	Clamp Screw	1 (1)	M.S.
45	Torque Arm Clamp	1 (1)	M.S.
44	Link Pins	[illegible]	Silver Steel
43	[illegible]	[illegible]	Silver Steel
42	Link for Quench Load Cell	1 (1)	M.S.
41	Torque Arm	1 (1)	M.S.
40	Torque Shaft	1 (1)	En 24
39	Adaptor Nut	2 (2)	[illegible]
38	Retaining Pin	2 (2)	[illegible] N.D.
37	Shaft End Adaptor	2 (2)	Nimonic 75
36	Screwed Collar	2 (2)	M.S.
35	Driven Shaft	1 (1)	En 24 [illegible]
34	Fulcrum Pin	1 (1)	M.S.
33	Roller	2 (2)	[illegible] Bronze
32	Hand Lever	2	M.S. Plated
31	Key	2 (2)	Key Steel
30	Dog Clutch Bobbin	1 (1)	En 24
29	[illegible] Cover N.D.	1	M.S.
28	Pinion Cover	1	M.S.
27	Mitre Gear	3	En 8
26	Clutch Shaft	2	En 24
25	Key	6 (1)	Key Steel
24	Dog Clutch Driver	2	M.S.
23	Key	1	Key Steel
22	Main Drive Shaft	1	En 24
21	Key	2	Key Steel
20	Bearing Pin	1	[illegible]
19	Step Coupling	1	M.S.
18	Alternative Shaft	1	M.S.
17	Cover Plate	7 (4)	M.S.
16	Clamp Plate	1 (1)	M.S.
15	Taper Roller Bearing	5 (1)	SKF 30207
14	Taper Roller Bearing	5 (1)	SKF 30208
13	Guide Race N.D.	4 (4)	SKF RS 22
12	Guide Race N.D.	4 (4)	SKF RS 22
11	Guide Rod	[illegible]	M.S.
10	Locking Pin	1 (1)	M.S.
9	Quench Chamber Base	1 (1)	M.S.
8	Furnace Base	1 (1)	M.S.
7	Guide Rail	2 (2)	Bright M.S.
6	Base Rail	2 (2)	Bright M.S.
5	End Thrust Block	1 (1)	M.S.
4	Torsion Block	1 (1)	M.S.
3	Countershaft Block	1 (1)	M.S.
2	Pinion Box	1	M.S.
1	Main Frame (2 parts)	1	M.S. Welded

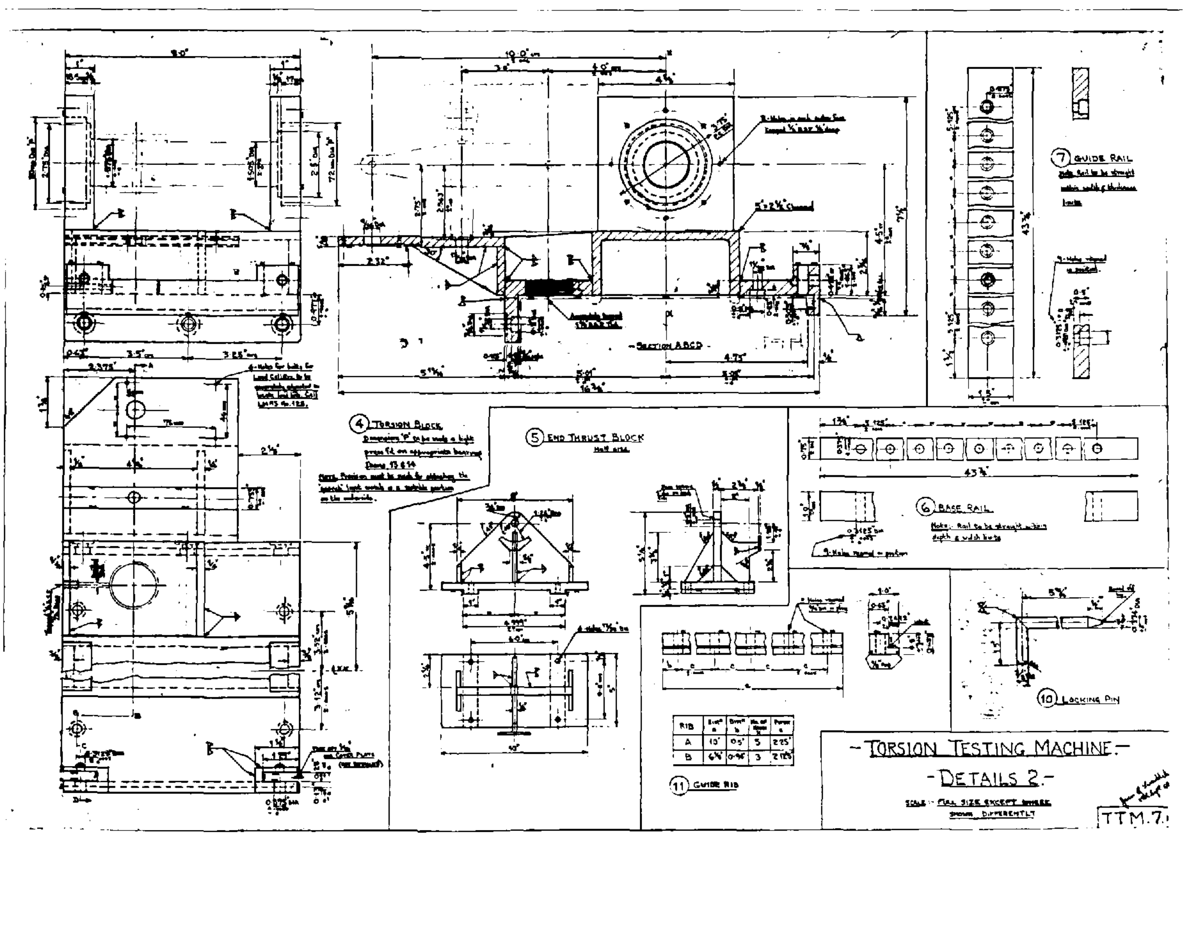


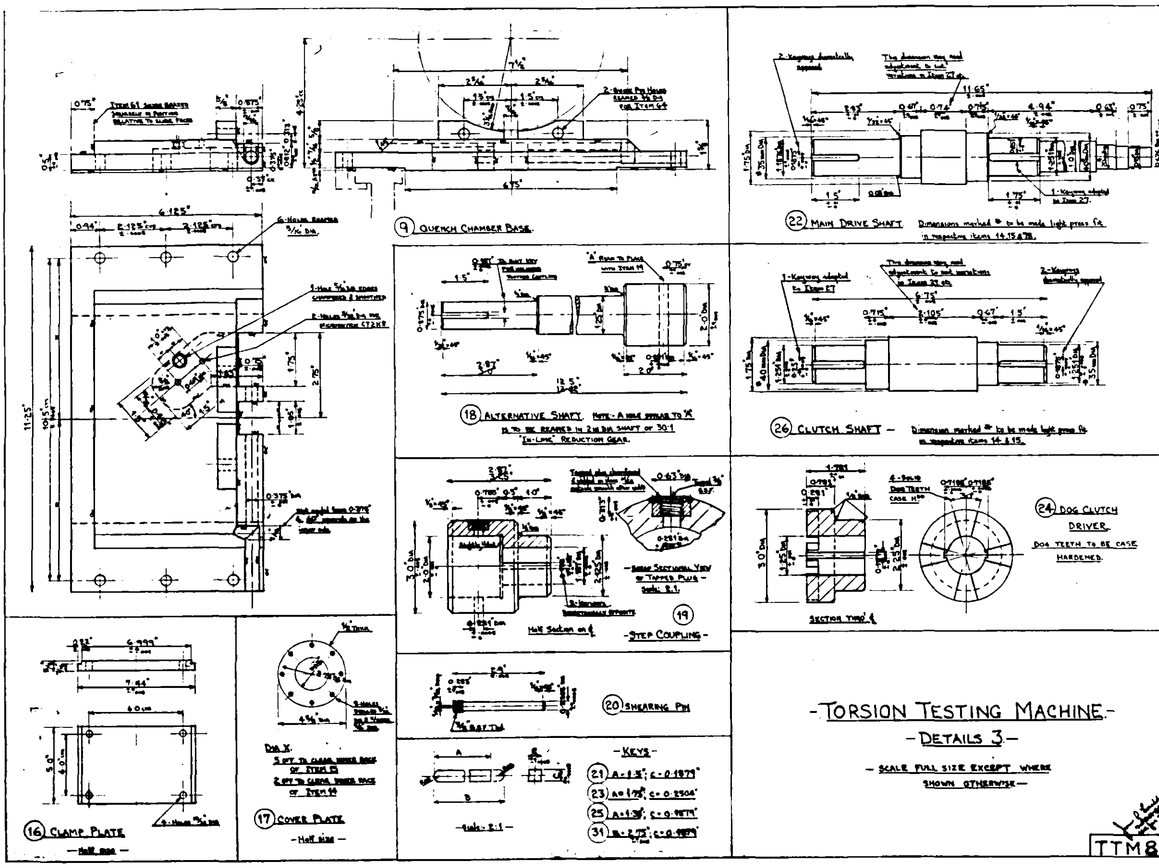
-TORSION TESTING MACHINE-
-MAIN FRAME, PART 1-
TTM4.

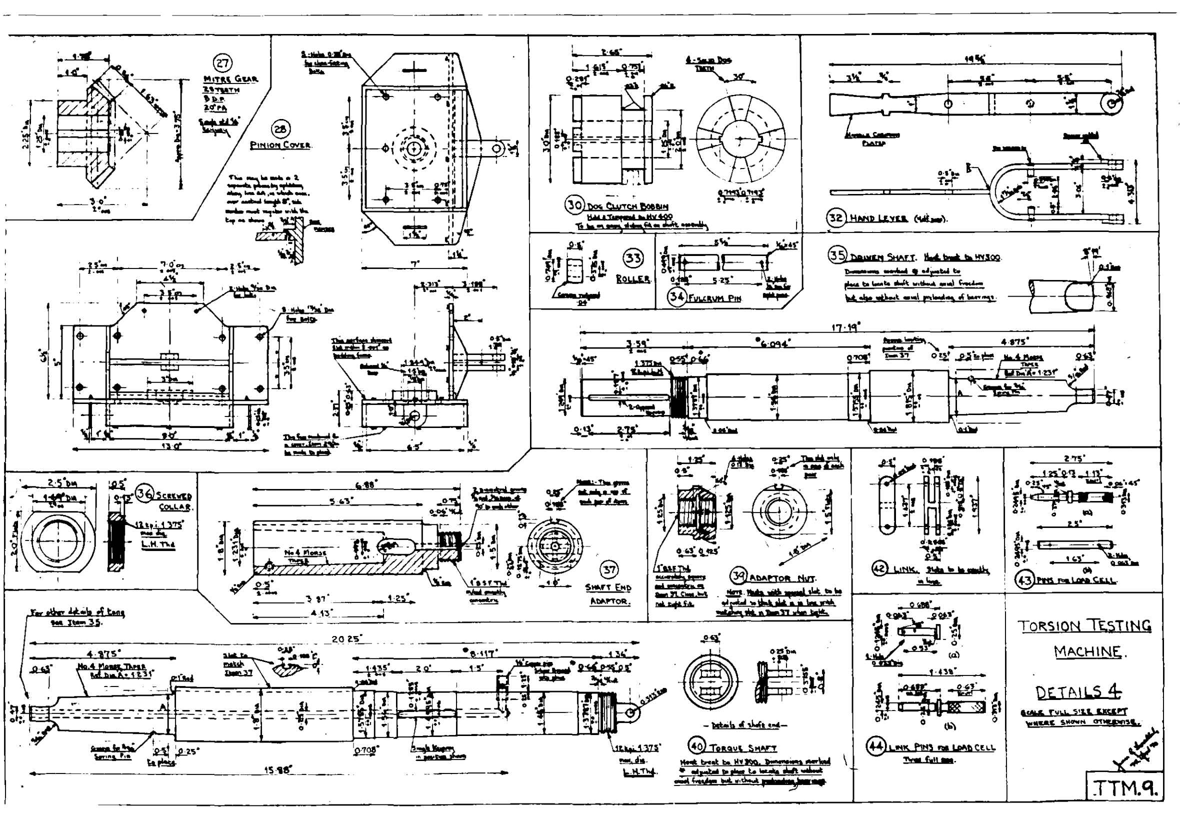


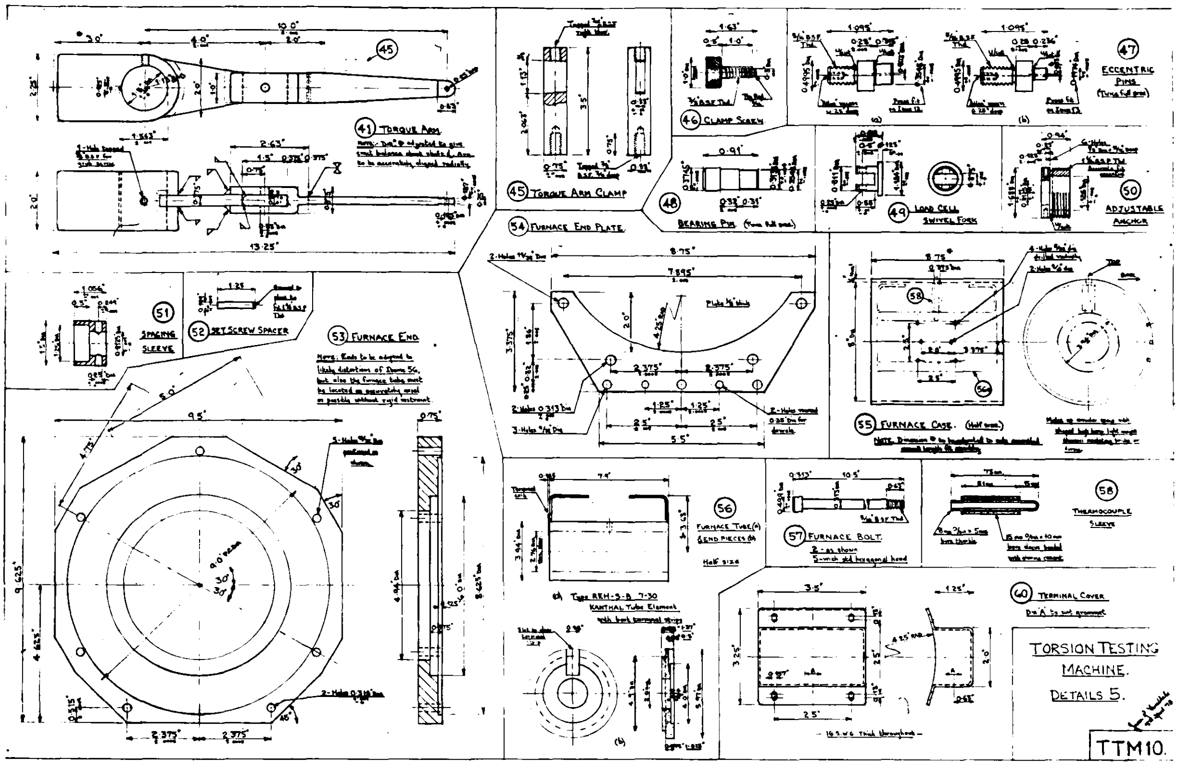
-TORSION TESTING MACHINE-
-MAIN FRAME - PART 2-
TTM5

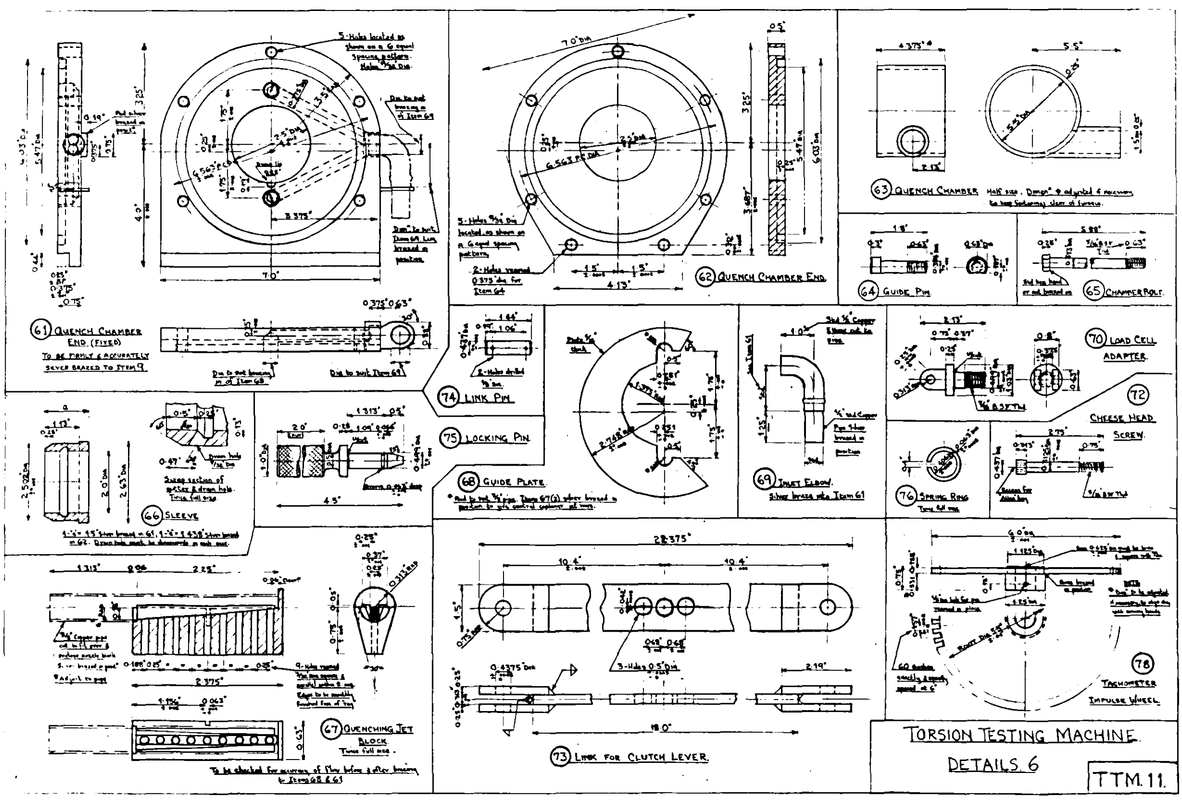

TORSION TESTING MACHINE. - DETAILS 1 -
(2) PINION BOX
(3) COUNTERSHAFT BLOCK
(8) FURNACE BASE

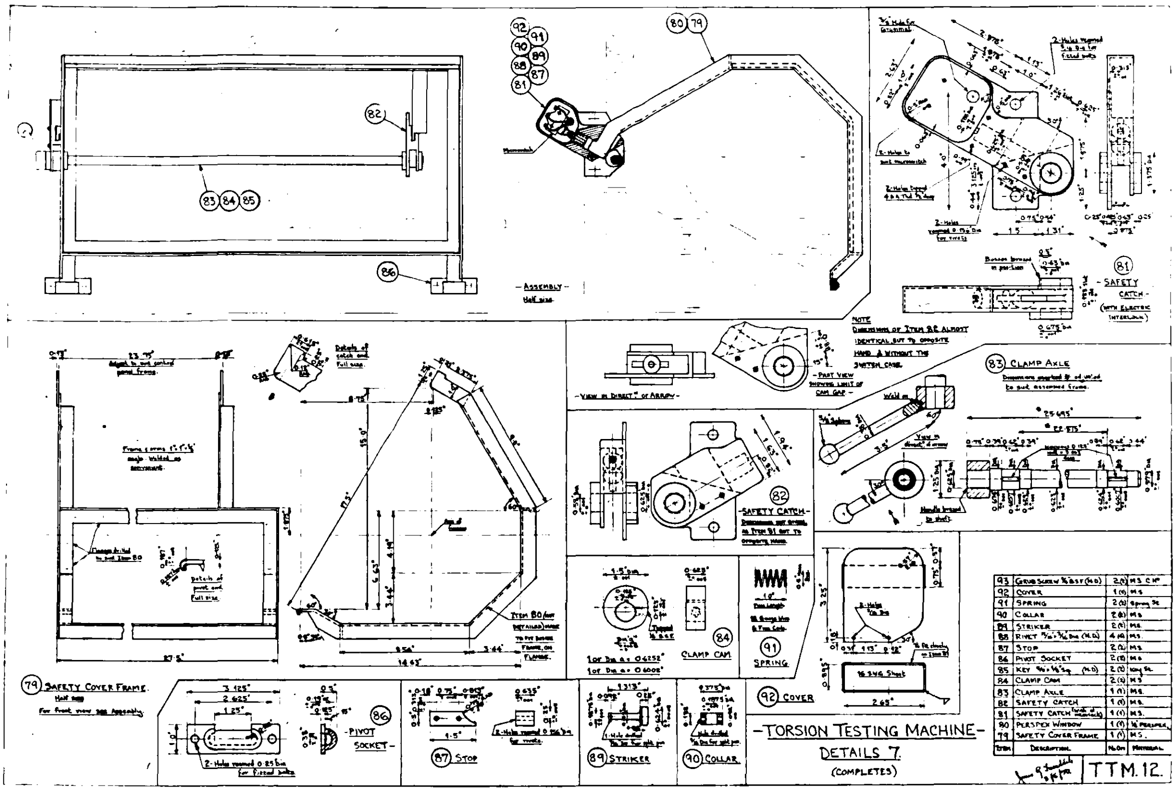

(4) TORSION BLOCK
(5) END THRUST BLOCK
(6) BASE RAIL
(7) GUIDE RAIL
(10) LOCKING PIN
(11) GUIDE RIB
- SECTION ABCD -
- TORSION TESTING MACHINE -
- DETAILS 2 -
TTM.7

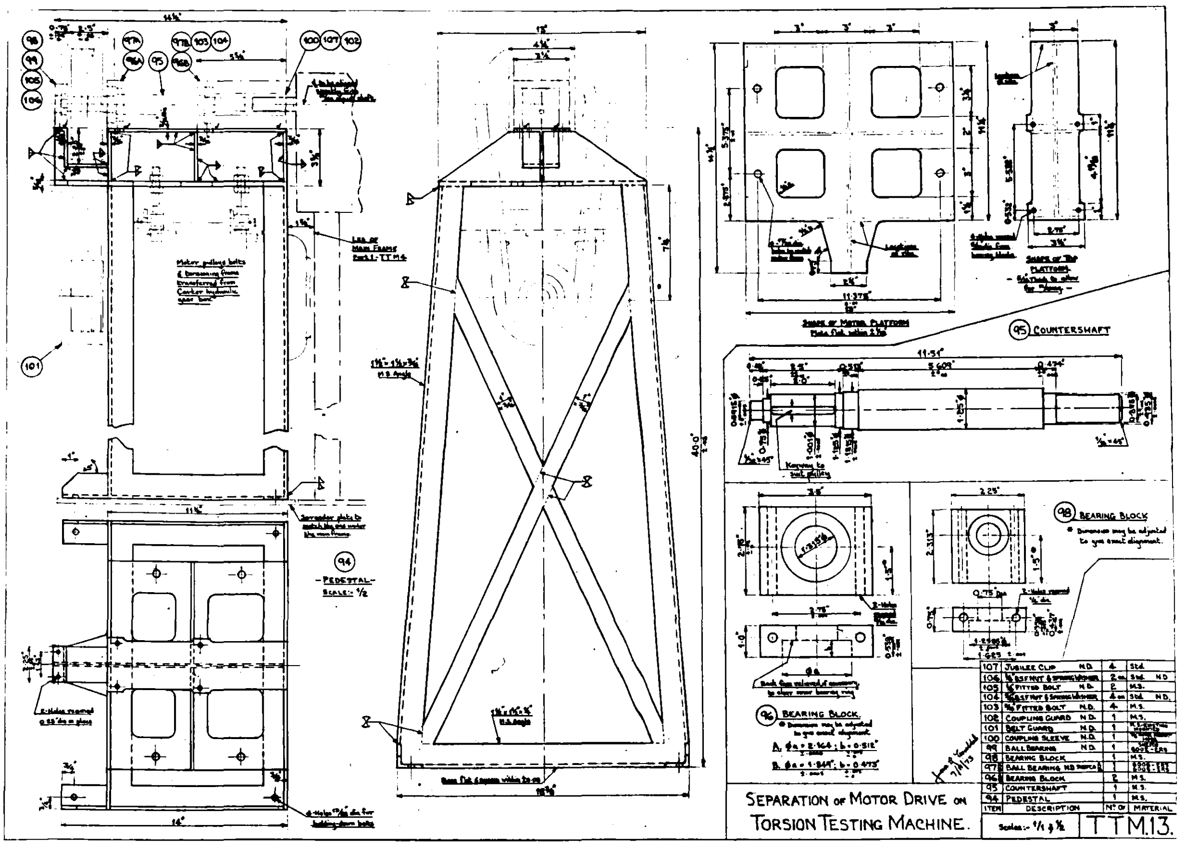

QUENCH CHAMBER BASE.
MAIN DRIVE SHAFT
ALTERNATIVE SHAFT
CLUTCH SHAFT
STEP COUPLING
DOG CLUTCH DRIVER
SHEARING PIN
KEYS
CLAMP PLATE
COVER PLATE
-TORSION TESTING MACHINE-
-DETAILS 3-
TTM8


MITRE GEAR
PINION COVER
DOG CLUTCH BOBBIN
HAND LEVER
ROLLER
FULCRUM PIN
DRIVEN SHAFT
SCREWED COLLAR
SHAFT END ADAPTOR
ADAPTOR NUT
LINK
PINS FOR LOAD CELL
TORQUE SHAFT
LINK PINS FOR LOAD CELL
TORSION TESTING MACHINE.
DETAILS 4
TTM.9.


41 TORQUE ARM
45 TORQUE ARM CLAMP
46 CLAMP SCREW
47 ECCENTRIC PINS
48 BEARING PIN
49 LOAD CELL SWIVEL FORK
50 ADJUSTABLE ANCHOR
51 SPACING SLEEVE
52 SET SCREW SPACER
53 FURNACE END
54 FURNACE END PLATE
55 FURNACE CASE
56 FURNACE TUBE(4) & END PIECES
57 FURNACE BOLT
58 THERMOCOUPLE SLEEVE
60 TERMINAL COVER
TORSION TESTING MACHINE.
DETAILS 5.
TTM10.


61 QUENCH CHAMBER END (FIXED)
62 QUENCH CHAMBER END
63 QUENCH CHAMBER
64 GUIDE PIN
65 CHAMBER BOLT
66 SLEEVE
67 QUENCHING JET BLOCK
68 GUIDE PLATE
69 INLET ELBOW
70 LOAD CELL ADAPTER
72 CHEESE HEAD SCREW
73 LINK FOR CLUTCH LEVER
74 LINK PIN
75 LOCKING PIN
76 SPRING RING
78 TACHOMETER IMPULSE WHEEL
TORSION TESTING MACHINE.
DETAILS 6
TTM.11.


ASSEMBLY
Half size
SAFETY CATCH (with Electric interlock)
CLAMP AXLE
SAFETY CATCH
CLAMP CAM
SPRING
SAFETY COVER FRAME
PIVOT SOCKET
STOP
STRIKER
COLLAR
COVER
-TORSION TESTING MACHINE-
DETAILS 7.
(COMPLETES)
93 GRUB SCREW (N.D) 2(2) M.S.
92 COVER 1(1) M.S.
91 SPRING 2(2) Spring St.
90 COLLAR 2(2) M.S.
89 STRIKER 2(2) M.S.
88 RIVET (N.D) 4(4) M.S.
87 STOP 2(2) M.S.
86 PIVOT SOCKET 2(2) M.S.
85 KEY (N.D) 2(2) Key St.
84 CLAMP CAM 2(2) M.S.
83 CLAMP AXLE 1(1) M.S.
82 SAFETY CATCH 1(1) M.S.
81 SAFETY CATCH 1(1) M.S.
80 PERSPEX WINDOW 1(1) Perspex
79 SAFETY COVER FRAME 1(1) M.S.
ITEM DESCRIPTION No. Off MATERIAL
TTM.12.


PEDESTAL
SCALE:- 1/2
COUNTERSHAFT
BEARING BLOCK
SEPARATION OF MOTOR DRIVE ON
TORSION TESTING MACHINE.
107 JUBILEE CLIP N.D. 4 Std.
106 BSF NUT & SPRING WASHER 2 on Std. N.D
105 FITTED BOLT N.D. 2 M.S.
104 BSF NUT & SPRING WASHER 4 on Std. N.D.
103 FITTED BOLT N.D. 4 M.S.
102 COUPLING GUARD N.D. 1 M.S.
101 BELT GUARD N.D. 1
100 COUPLING SLEEVE N.D. 1
99 BALL BEARING N.D. 1
98 BEARING BLOCK 1 M.S.
97 BALL BEARING N.D. 1
96 BEARING BLOCK 2 M.S.
95 COUNTERSHAFT 1 M.S.
94 PEDESTAL 1 M.S.
ITEM DESCRIPTION No. OF MATERIAL
Scales:- 1/1 & 1/2
TTM.13.

APPENDIX B.

SEQUENCE OF OPERATION.

B.1. Preparatory

Connect in the appropriate load cell, or cells if end load is also to be measured.

B.2. To insert t.p.

(i) Ensure that power, air and water are off and that the torque arm is clamped, then disengage the dog clutch.

(ii) Remove the pin between quench chamber and furnace.

(iii) Slide the furnace to the extreme right and the quench chamber to the extreme left.

(iv) Unclamp the end-load measuring block

(v) Slide the right hand torsion head assembly whichever way is needed.

(vi) Fit the screwed adaptors (Items 39, drawings TTM 2 and 3).

(vii) With the r.h. assembly a little to the right, screw the r.h. end of the t.p. firmly into the r.h. adaptor.

(viii) Slide the r.h. assembly gently towards the left and start the l.h. end of the t.p. into the l.h. adaptor by rotating the l.h. spindle with the left hand,from between the two bearings next to the dog clutch. Screw the t.p. firmly home.

(ix) Check that the end load-measuring device is either:(a) completely disconnected and clear, if no end-load restraint is to be applied, or (b) that it is properly engaged, with zero load applied, and firmly clamped.

(x) Push the furnace back to the left and engage the pin connection with the quenching chamber.

(xi) Re-engage the dog clutch.

(xii) Release the torque arm clamp and set it to prevent the torque load cell being overloaded more than 20% over its normal maximum range.

B.3. <u>To remove t.p.</u>

(i) Cut off power, air and water, clamp the torque arm and disengage the dog clutch.

(ii) Separate the quench chamber and the furnace. (Normally, if a test has been made, the furnace will be in the quench position, so the quench chamber will have to be moved to the left.)

(iii) Slacken the end-load measuring device.

(iv) If the specimen is unbroken, pull the r.h. assembly to the right whilst unscrewing with the l.h. spindle. Unscrew the unreleased end by hand. If the t.p. is broken unscrew each half independently by hand.

B.4. <u>To operate a test.</u>

(i) Proceed as (2) to insert the t.p. and prepare for the start.

(ii) Switch on single phase current (panel light should show) and press 'furnace position' 'heat' button (this is important to ensure that the air control valve is properly set - see Note, below.)

(iii) Turn on first the air and then the water (panel lights should show).

(iv) Set the furnace temperature control and press the 'furnace live' button (furnace panel light should show).

(v) Let the furnace reach temperature (furnace warning light will begin to go out for increasing periods as control temperature is reached).

(vi) Select the appropriate strain timing circuit ('manual' or 'automatic') and 'single', or 'repeat' cycling system when 'automatic' is engaged, and allow the circuit to warm up (1-2 minutes). Set the appropriate cycle times as relevant.

(vii) Press the 'clutch live' button. If 'manual' is engaged straining will start immediately and continue with the clutch light showing until the 'clutch stop' button is pressed; if 'auto' is engaged nothing will happen until the 'cycle start' button is pressed. The latter operation will give either only one-cycle of strain operation, if 'single cycle' is engaged, or it will give continuous cycling, if 'repeat' is engaged, until the 'clutch stop' button is pressed. Note that, if single cycling is engaged, single cycles may be repeated at will by pressing the 'cycle start' button as long as the 'clutch stop' button is not pressed. <u>This latter system is intended primarily for setting purposes and it could be dangerous to leave the clutch potentially active when not using single cycling</u>.

Auxiliary recording systems (e.g. u.v. recorder) may be interlinked with the clutch circuit for automatic starting or they may be switched on manually at an appropriate time, for continuous operation.

(viii) Close the safety cover.

(ix) When straining is complete, press the 'clutch stop' button and immediately press the 'furnace position' 'quench' button. The quench button will not operate until the clutch stop button has been pressed and the cover is closed.

(x) Allow quenching to continue as long as required for cooling, then turn off the water, press the 'furnace off' and 'furnace position' heat buttons and open the cover.

(xi) Clamp the torque arm.

(xii) Proceed as (3) to remove t.p.

<u>Note</u>:

The main solenoid-operated air valve is a 'balanced' valve which will stay in its last-operated position when the machine is switched off; therefore, before turning on the air-pressure <u>always</u> operate the 'furnace position' 'heat' button (see 4(iii)). If this is not done and the furnace is in the 'heat' position with the valve in the 'quench' position, as air is turned on the furnace will be shot violently into the 'quench' position <u>without the protective cover being in position.</u> Normally the furnace will not move to the 'quench' position if the cover is not in place.

APPENDIX C.

SOME CALCULATIONS RELATING TO MACHINE COMPONENT STRENGTH, VIBRATION ETC.

C.1. Some Calculations Relating to Strengths of Particular Components.

The calculations that are given in this Section refer either to some feature of design that is outside the normal elastic design approach or to some aspect of special importance. The section is divided into sub-sections each of which deals with an individual component as indicated in the heading.

There are three aspects of the calculations.

(i) Calculations based on stiffness under the normal maximum torque of 23Nm which require a safety factor of about 40:1 on the strength of the material

(ii) Calculations based on compromise stiffness under the occasional maximum of 46Nm torque with a safety factor of about 20:1 on the strength of the material.

(iii) Calculations of working stresses in components not restricted by need for stiffness, where the safety factor may be quite low, perhaps as low as 4:1.

C.1.1. Taper Roller Bearings Items 14 and 15.

Each main shaft in the direction-changing gear box and the torque measuring head is to run on two accurately opposed taper roller bearings. For reasons of fitting etc. one of each pair has to be slightly larger than the other. Preliminary study of makers' safe load values showed that a bearing of 30mm bore should be just about satisfactory; but it is logical to play safe so the minimum size of the pair was chosen as 35mm (standard Skefco bearings increase in 5mm steps on shaft diameter) which is safely capable of taking a maximum static load of 26kN against a radial gear load of 622N x shaft leverage of about 1.5:1 maximum i.e. a total radial load of 933N. This gives a safety factor of 27.5 against the maximum static <u>safe</u> load and greater factors on safe rotating load strengths (within certain limits load bearing capacity increases with increasing speed of rotation). Since the reserve factor on breaking is certainly at least 2:1 this brings the safety factor on breaking to, at the very least, 55:1 for direct radial load, which is well above the equivalent required minimum value for the shafts at 23Nm torque.

However, there is also the question of the possible additional axial load of 500N from axial contraction or expansion of the test specimen.

Equivalent radial load from axial thrust P_A is: YP_A where Y is an arbitrary thrust factor derived from makers' experience. Y in this case is quoted as 1.55

Hence, possible equivalent additional radial load

$$= 500 \times 1.55$$

$$= 775N$$

Giving a total max. radial load, at 23Nm torque, of 1708N, say 1.7kN which would bring the absolute minimum total factor down to 27 or 13.5 on the maximum <u>safe</u> static loads. This is still satisfactory, both for normal use at 23Nm and for regular use at 46Nm torque (6.8 reserve on the makers maximum safe static load) since the strengths of these components do not affect the rigidity of the machine and there is advantage in keeping such components as light as possible, consonant with effective operation of the machine, to reduce friction to the minimum possible value.

C.1.2. <u>Mitre Gear Key Item 23.</u>

The key is 6.3mm wide by 31.7mm ~~long~~ effective length at the mean radius of 15.8mm.

∴ Shearing force under 23Nm torque

$$= \frac{23 \times 10^3}{15.8}$$

$$= 1456\text{N}$$

∴ Shearing stress $= \frac{1456}{6.3 \times 31.7}$

$$= 7.3\text{N/mm}^2$$

The shearing strength of the key material would be at least 300N/mm² (rather low in fact) but even this gives a minimum safety factor of 41:1.

C.1.3. Mitre Gear, Item 27.

Standard gear formulae relate to gears intended for continuous long-duration operation at full working load; but, in this application, operation is very intermittent and durations are likely to be very short; therefore, a simpler design approach may be used. The maximum torque may be related to the stress developed in a single tooth and a correction factor added as considered necessary.

The maximum force at the average pitch radius (37mm) of the gear under a torque of 23Nm

$$= \frac{23 \times 10^3}{37}$$

$$= 622\text{N}$$

average depth of tooth below pitch line

$$= 4\text{mm}$$

∴ B.M. at tooth root

$$= \frac{4 \times 622}{10^3}\ \text{Nm}$$

$$= 2.5\text{Nm approx.}$$

Average thickness of tooth at root

$$= 3.82\text{mm}$$

Length of tooth

$$= 21.8\text{mm}$$

Nominal average modulus of section of the root of the tooth

$$= \frac{1}{6} \times 21.8 \times 3.82^2$$

$$= 53\text{mm}^3$$

$\therefore$ Max. normal stress in static bending

$$= \frac{2.5 \times 10^3}{53}$$

$$= 47.3\text{N/mm}^2$$

But this should be adjusted upwards to allow for the effects of velocity.

The standard empirical formula is:-

$$\sigma_m = \frac{183 + (V \times 60)}{183}\sigma_s$$

where σ_m = nominal max. dynamic stress

σ_s = static max. stress

V = max. velocity in m/s at the max. pitch radius

The maximum speed of rotation is 18 rev/s giving a maximum velocity, at the maximum pitch radius of 44.4mm, of 5.022 m/s.

$$\therefore \sigma_m = \frac{183 + (5.022 \times 60)}{183} \times 47.3$$

$$= 2.65 \times 47.3$$

$$= \underline{125}\text{N/mm}^2$$

If the gear material is En9 steel, heat treated to 900N/mm² tensile strength, this gives a safety factor of 7.2 which should be quite adequate; particularly, as the deflection of a gear tooth at the max. normal working stress will not add significantly to the total deflection and the frequency of resonant vibration of the tooth will be so high that it is unlikely to respond to any source within the system.

Should the machine be operated at 46Nm the safety factor will fall to 3.6. This is too near the limit for frequent operation at this level but should be just about adequate for occasional use.

It is advisable to check the max. shearing stress in a tooth.

$$\text{Direct shearing stress} = \frac{622}{21.8} \times 3.82$$

$$= 7.5\,\text{N/mm}^2 \quad \text{at } 23\,\text{Nm torque}$$

∴ Max. resultant shearing stress

$$= \sqrt{7.5^2 + \left(\frac{125}{2}\right)^2}$$

$$= \underline{63}\,\text{N/mm}^2$$

This gives a safety factor of 8:1 which is just a little more adequate than the tensile safety factor.

C.1.4. Torque Transmission Shafts Items 35 and 40 (Shafts between strain measuring system and torque measuring system).

These two shafts are made from En24 steel (Range S; O.Q. 830°C; tempered 650 - 700°C) with a tensile strength of 750 N/mm² and shearing strength of 500 N/mm².

To give a minimum safety factor of 40:1 on the sizes of these shafts, at temperatures up to about 400°C, at 23Nm torque, the maximum stress should not be more than:

$$\frac{500}{40} = 12.5\,\text{N/mm}^2$$

which requires a minimum diameter $\not<$ 22mm.

These shafts have to be mounted in the taper roller bearings, items 14 and 15, (see C.1.1.), with internal diameters of 35mm and 40mm, respectively, so there is an adequate range for working sizes greater than the minimum.

It is not expected that the temperatures of the shaft ends, near the hot zone, will rise above 400°C, so there should be no loss of stiffness or safety from this cause.

C.1.5. Torque Arm Item 41.

The arm has two leverage points: one at 100mm from the pivot centre and one at 250mm from the pivot centre.

Within the 100mm radius the maximum bending load will be governed, normally, by the 23Nm torque or occasionally by the 46Nm torque; but, outside the 100mm radius the maximum working torque will be 11.5Nm.

Thus, where the arm meets the central boss it will be subjected to a bending moment M_{100}.

$$M_{100} = \tau \times \frac{(100 - r)}{100} \text{Nm} \quad \text{where } \tau = \text{torque and}$$

r = rad. of central boss

= 25mm

$\therefore M_{100}$ (normal maximum) = 19.5Nm

and M_{100} (occasional maximum) = 39Nm

Maximum permissible tensile stress at normal safety factor 40:1

$$= \frac{460}{40} \quad \text{(the arm is made of mild steel)}$$

$$= 11.5\text{N/mm}^2$$

$$\therefore Z_t = \frac{19.5 \times 10^3}{11.5}$$ where Z_t = modulus of section in bending of arm, tangent to the boss

$$= 1696\text{mm}^3 \text{ minimum}$$

$\therefore$ A solid rectangular arm 10mm wide needs to be at least 46mm deep at the tangent section.

Similarly, the normal maximum bending moment at the 100mm radius,

$$M_{250} = 11.5 \times \frac{(250 - 100)}{250}$$

$$= 6.9\text{Nm}$$

$\therefore$ Minimum modulus of section in bending at 100mm rad

$$Z_u = 600\text{mm}^3$$

$\therefore$ If depth of section is maintained at 46mm the width may be reduced to 4mm at 100mm rad.; or, if width is kept at 10mm the depth may be reduced to 19mm.

These values will maintain the safety factors at the normal 40:1 and the occasional 20:1 levels.

The torque resisting key is stressed to the same safe level as the other keys in the direct train (7.3N/mm^2).

C.2. <u>Some Calculations Relating to Deflection.</u>

In general, provided that normal reasonable safety factors are maintained, deflections in components outside the test bed area do not affect the operational accuracy of the machine. However, in the train from the point at which strain measurement begins, i.e. from the main drive shaft, item 22, up to the end thrust block, item 5, deflections can be critical with respect to (i) backlash in the system, (ii) accuracy of operation and (iii) accuracy of measurement of load.

Two modes of deflection are likely to be important within this critical zone, namely, twisting and bending. The former is likely to be most critical throughout the torque drive train and within the test bed frame and the latter principally within the test bed frame, where it can cause misalignment in the drive train on the lines discussed in Section 4.5. The bending deflections must be considered in both the vertical plane and the horizontal plane systems.

C.2.1. Torsional Deflection.

The total torsional deflection inherent in the system is made up of angular deflections in the gear and clutch bosses, angular deflections in the gear teeth, angular deflections in the keys and angular deflections in the shafts. Of these, due to inherent rigidity of construction, deflections in the bosses and deflections in the keys (assuming correct fitting) are completely negligible and may be ignored. Total torsional deflection is thus the sum of the other factors.

Gear Teeth Deflection.

As a simple first approximation each gear tooth may be regarded as a simple cantilever beam of rectangular sectional shape and, if only one tooth on each side is regarded as taking the thrust at any one time, the total gear tooth deflection will be twice that of a single tooth. Furthermore, if the effect of the tooth root radius is ignored this will give the extreme value of possible deflection from this cause.

Min. root area of tooth section $= 21 \times 3.82 \text{mm}^2$

Second moment of area $= 97.543 \text{mm}^4$

Max. beam length $= 3.96\text{mm}$

Max force at 23Nm torque $= 560\text{N}$

Plasticity modulus in tension $= 206.83 \times 10^3 \text{N/mm}^2$

$\therefore$ Deflection/tooth $= \dfrac{560 \times 3.96^3}{3 \times 206.83 \times 10^3 \times 97.543}$

$= 0.000723\text{mm}$

$\therefore$ Total deflection $= 0.001446 \text{mm}$

$= \dfrac{0.001446}{44.4}$ radians (where mean pitch rad. = 44.44mm)

$= 0.00003483$rads.

<u>Shaft Angular Deflection.</u>

Taking elastic modulus in shearing (G) $= 82.732 \times 10^3 \text{N/mm}^2$

and angular deflection in torsion $\alpha = 10.186 \dfrac{TL}{GD}$

where L = shaft length

and D = shaft dia

C.1. <u>Table of shaft portion sizes and associated angular deflections.</u>

Item	Dia of Section		Length of Section		Deflection rads.
	in	mm	in	mm	
26	1.75	44.4	2.105	53.5	0.000037912
	-	40	0.715	18.2	19579
	-	35	0.67	17.05	31290
	1.25	31.8	1.629	41.4	111490
				Total	0.000200271

Item	Dia		Length		Deflection rads.
	in	mm	in	mm	
35	1.815	46.1	2.631	66.8	0.000040713
	1.75	44.4	0.708	18.0	012756
	1.56	39.6	4.78	121.4	135950
	1.379	35.0	0.66	16.75	030739
	1.25	31.8	1.925	49.0	131960
	1.23 (Mean)	31.7	4.565	116.0	316370
				Total	0.000668488
37*	1.5	38.1	1.685	42.80	0.000055939
	0.8719	22.3	0.75	19.07	212370
				Total	0.000268309
40	1.8	45.7	5.89	150.0	0.000094703
	1.5	38.1	2.435	62.0	511280
	1.23 (Mean)	31.7	4.565	116.0	316370
				Total	0.000922353

*Adjustment made to G for n.t.p. working section; assumed about same as steel for max. elevated temp. section.

Grand total elastic angular deflection at 22.3Nm = 0.00236256 rads. which is negligible relative to the likely range of plastic angular deflections.

Angular deflection of the double-channel test frame (Item 1, Part 2) can add a little to this (0.00001138 radians); but this may safely be neglected here, although it can add significantly to the lateral deflection.

Even at 46Nm torque the total elastic angular deflection is still only 0.00488 radians.

C.2.2. Bending Deflections.

Bending deflection in the drive train is attributable to two causes: (i) bending and twisting in the main test bed frame and (ii) out-of-balance forces due to misalignment.

The first cause results from (a) dead loading from machine parts and (b) side thrusts resulting from the application of the torque. Both amounts are negligible with the bedplate sizes chosen; since the bedplate section sizes and proportions were selected mainly for this very purpose. However, the angular displacement due to the torque can contribute to misalignment of the torque drive but this effect is minimized by the fact that the twist spreads proportionately from end to end of the bed; hence intermediate fittings (bearings etc.) tend to tilt in proportion so the resultant concentrated effect at the test position is very small and may be neglected.

The second cause, out-of-balance forces, develops from inaccuracy in the setting of the alignment of the torque applying mechanism, particularly in the close vicinity of the test specimen where the leverage from the overhang of the shaft is at its maximum.

The values of the out-of-balance forces depend directly on the degree of misalignment, at the centre of application of the torque within the specimen. Such a misalignment will give a difference in concentricity between one end and the other of the gauge length of the specimen, creating an out-of-balance torque proportionate in magnitude to the unconcentric area of one side or the other relative to the total area.

As long as the degree of eccentricity is not great, say not more than 20% of the radius, it can be shown that the out-of-balance area (A_o) will be closely approximate to twice the eccentricity (e) times the radius (r)

i.e. $A_o = 2er$

Hence, for an eccentricity of 0.05mm,

$$A_o = 0.1 \times 3\text{mm}^2 \qquad (r = 3\text{mm})$$

$$= 0.3\text{mm}^2$$

$\therefore \frac{A_o}{A} = 0.0107$ where A = full area

Out-of-balance torque $\Upsilon_o = 0.0107\,\Upsilon$ where Υ = applied torque

$\therefore$ when $\Upsilon = 23$Nm

$\Upsilon_o = 0.244$Nm

The effective radius of action of this torque may be taken as $3 - \frac{0.05}{2}$ (i.e. the mean radius of eccentricity)

$= 2.975$mm

$\therefore$ Eccentric force $P = \frac{0.244 \times 10^3}{2.975}$

$= 82$N

Similarly, for an eccentricity of 0.5mm at 23Nm torque,

$A_o = 3\text{mm}^2,$

$\frac{A_o}{A} = 0.107,$

$\Upsilon_o = 2.44$Nm.

$\therefore P = 890$N

The necessity for accurate alignment is obvious. Provided that the required accuracy of alignment is maintained lateral deflection of the torque shafts from this cause may be ignored.

C.3. Some Calculation Relating to Vibration.

Once the danger of misalignment or malfunction due to deflection of the machine structure has been overcome, there remain the dangers of vibration and resonance in the members of the structure.

Vibration can originate either within the machine or within the test specimen. The behaviour of the latter cannot be predicted for unknown materials and new test conditions; but a rough examination of a variety of test records from different sources suggests that the maximum frequency of vibration likely to result from "see-saw" yielding is probably about 10Hz. This frequency is well down in relation to the range of excitation frequencies likely from the machine itself; therefore, if the latter are safely taken into account, specimen vibration generation should also be safely covered. Consequently, specimen vibration generation is ignored in subsequent calculations.

There are three likely sources of vibration in the machine; taken in order from the drive motor:-

(i) Motor vibration from slight out-of-balance masses in the rotor.

(ii) Belt "thump" due to slight irregularity in V-belt contour as it meets a pulley.

(iii) Gear tooth chatter.

(i) Motor vibration will occur at an approximately constant frequency equal to the nominal motor rotational speed of 1400 rev/min

i.e. 23.35Hz, say 24Hz.

(ii) Belt thump is proportionate to the number of irregularities in the belt length related to the frequency with which they hit the pulleys. There are three V-belts and each appears to have only one irregularity. Hence, since motor speed is about constant, the impact frequency from each belt is a fixed fraction of the motor vibration frequency. This fraction is the ratio of the motor pulley mean drive circumference to the length of the belt - about 0.442.

This gives a belt thump frequency, for each belt,

= 10.5Hz approx.

However, there are three belts and their interaction must be considered.

Out-of-phase thumping may be ignored since this supply constitutes a particular state of basic frequency application. On the other hand if any two belts are exactly in half phase with each other then a second frequency of double the basic frequency is present i.e. 21Hz excitation also exists. If the three belts are exactly in progressive $^1/_3$ phase with each other, then a second frequency of treble the basic frequency is present i.e. there is 31.5Hz excitation.

However, the chance of exact three phasing being present is so extremely small that it may be ignored.

Two phasing is more likely, although still highly improbable, and should be considered in addition to basic excitation.

(iii) Gear tooth chatter is the most likely source of excitation since a wide range of frequencies of excitation is certain in contrast to the fixed frequencies of the other two sources.

Because there are 28 teeth in the direction-charging mitre gears, item 27, the frequency of excitation from this source will be 28 times the selected speed of strain drive rotation. The latter is likely to range from 1 rev/min to 970 rev/min (max. hydraulic variable-speed gearbox output speed) giving a potential tooth chatter frequency range from about 0.5Hz to 452Hz.

If the resonant frequencies in the machine members are kept as far as possible above the maximum of this range (although this may not always be possible) then risk of serious vibration should be small.

It is essential to consider both torsional and transverse modes of vibration and this is done in the subsequent parts of this section. No consideration is given to axial vibration as this seems to be an extremely unlikely source of trouble in this construction.

(Note. In the eventual operation of the machine it was found that belt thump significantly affected the sensitivity of the load measuring ~~cells~~ used for the low torque range and it was found desirable, for accurate low torque work, to isolate the belt drive system from the main frame system, see drawing TTM 13. Whether or not the choice, in the first place, of a direct-coupled motor system see Section 3.1.1, would have avoided this difficulty is open to question because the inherent motor vibration might then have caused similar or more serious problems).

C.3.1. Torsional Vibration.

Torsional vibration is most likely to occur in the torque drive members, items 18,26,35 and 40. There might also be a possibility of torsional vibration in the test bed frame, item 1, part 2, but the heavy damping imposed by the system of leg mounting, by the guide rail system and by superficially mounted equipment is considered to be sufficient to eliminate this possibility.

The main torque driven shaft, item 35, is the most likely source of torsional vibration so it is considered first.

All frequency calculations are based on the standard equation

$$n = \frac{1}{4l} \frac{GAg}{w}$$

where n = frequency in Hz., l = length of shaft under consideration, G = shearing modulus of elasticity = 82.732×10^3 N/mm^2, A = area of circular section in mm^2, g = acceleration due to gravity (9.8067m/sec^2), w = weight/ /unit length, based on a density of 7.8gms/cc.

This simplifies to

$$n = 14242 \times \frac{1}{l} \times \sqrt{\frac{A}{w}}$$

The overall torsional vibrational frequency of a complete shaft is approximated by calculating a mean effective diameter and assuming that the shaft is a single uniform cylinder. This is a very rough approximation but is good enough for the present purposes since it gives an indication of the minimum natural frequency.

Table C.2. Basic frequencies of torsional vibration.

Ref.	d_{mm} Mean	A_{mm^2}	$W_{g/mm}$	l_{mm}	$n_{Hz.}$
1	39.6	1233.0	96.173	436.0	3698.5
2	31.7	789.23	61.156	114.31	14106.6
3	31.8	794.22	69.948	62.36	25860.0
4	39.6	1233.0	96.173	138.02	8531.2
5	46.1	2101.3	163.9	66.83	24128.5

Table C.3. Inherent frequency ratios.

Denominator	Section ref. (Table C.2.) - Numerator			
	2	3	4	5
1	3.81	6.98	2.31	6.53
2	1	1.84	-	1.72
3	-	1	-	-
4	1.655	3.03	1	2.83
5	-	1.072	-	1

\- = less than1

Table C.4. Harmonic frequencies.

Basic	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$
25860.0	12930.00	8286.67	6440.0
24128.5	12064.25	8042.88	6032.13
14106.6	7053.3	4702.2	3526.65
8531.2	4265.6	2843.7	2107.8
3998.5	1999.25	1332.83	999.63

It can be seen, from Tables C.2. and C.4, that both the lowest basic frequency and the lowest harmonic frequency are significantly above the maximum potential generating frequency (452Hz) and hence any torsional resonance is unlikely.

Table C.3. shows that there is no self stimulating resonances in the shaft itself.

Item 26.

By simple proportion, all the frequencies in this shaft are significantly higher than those in item 35 so resonant torsional vibration is even less likely. In addition, damping is likely from the clutch components and keys.

Item 40.

This item will have natural frequencies and harmonics closely similar to those of item 35 and, in addition, there will be damping through the specimen and from the load lever system. Hence, torsional resonance is at least no more likely in this item than in item 35.

Item 18.

This item, when in use, might be stimulated by backward feed of vibration from the gear teeth. It has only one likely basic torsional vibration mode.

Table C.5. Resonant frequencies.

d.mm	A.mm	w.kg/mm	l.mm	n.Hz.
31.8	794.22	0.061948	193.8	8320.6
		Harmonics	$\frac{1}{2}$	4160.3
			$\frac{1}{3}$	2773.53
			$\frac{1}{4}$	2080.15

Torsional resonance is clearly unlikely.

C.3.2. Transverse Vibration.

Transverse vibration is a possibility in every significantly long number of the structure.

The most dangerous location for transverse vibration would be in the torque shafts on either side of the specimen, items 35 and 40; but, the inherent rigidity of these shafts is such that their absolute minimum frequency of transverse vibration in "free" resonance would be significantly higher than the maximum stimulating frequency. In fact they cannot resonate in the "free" mode because the bearings are deliberately not set in nodal positions. Hence neither natural or harmonic resonance is likely. Cantilever vibration might be possible in the large overhangs in the test zone but the two overhangs are neither of equal length nor of harmonic length ratios and in any case the test specimen acts as a damper, therefore resonant transverse vibration is unlikely from the normal excitation sources.

However, if the torsion block, item 4, drawing TTM 7 is not firmly guided against the rails, item 7, then transverse horizontal or vertical inertial vibration or "hunting" might be possible from the excitation of fluctuating transverse forces in the specimen caused by misalignment of the specimens. It is not possible to calculate the frequency of inertial oscillation of a complex member, such as item 4, with superimposed fittings, such as item 40; but it is certain that the frequency would be low perhaps lower than 1Hz. On the other hand, gear tooth chatter at low straining speed is an unlikely stimulus to this mode of vibration, due to the poor coupling that would exist, and all other stimulating sources are of too high a frequency to matter in this case. Only the side-thrust effect is left and this can be kept to a safe limit by:-

(i) ensuring sufficiently accurate alignment and

(ii) limiting the freedom of item 4 by keeping its roller guides under significant positive load, sufficient to stop lateral movement.

The only remaining potential likely source of dangerous transverse vibration is the frame, item 1, parts 1 and 2, which ~~are~~ is analysed below.

Item 1, Parts 1 and 2.

In both parts the only likely troublesome transverse vibration sources are the main rectangular frame tops, made from the 102 x 51mm standard channel sections, which might vibrate vertically or horizontally. The lowest and most dangerous frequencies will occur when ~~the~~ portions of channel vibrate independently, perhaps with one side portion resonating the other equal side portion (each significant portion is paired since each frame is constructed nearly symmetrically from two opposed channels e.g. see Figs. 23 and 27). Each frame side member divides into three portions since between the end supports used on each side there are two basic intermediate supports. The exact locations of these supports cannot be specified to precise vertical planes, because the supports themselves are of significant size (51 x 51mm and 38 x 38mm angle sections); on the other hand if the extreme end planes or mean planes, as appropriate, are used, the potential frequencies will be at their lowest. This might be regarded as an extreme assumption giving excessively low values of frequencies; but, if it is offset by assuming rigid end fixings, the calculated values should not be far from the true values. That is, a compromise is achieved between low values from excessively long assumed lengths and high values from assumed excessive stiffness. (Frequency of transverse vibration for rigidly fixed ends varies with $\frac{3.57}{l^2}$ and for one end free and the other rigidly fixed with $\frac{2.571}{l^2}$).

The basic equation for transverse vibration of a beam is:-

$$n = \frac{C}{l^2}\sqrt{\frac{gEI}{w}}$$ where n = frequency in Hz,

C = a constant related to mode of end fixing (3.57 in this case), l = effective length of beam in mm, g = acceleration due to gravity ($9.8067 m/sec^2$), E = tensile modulus of elasticity $206.83 \times 10^3 N/mm^2$, I = second moment of area of section - mm^4, W = weight/unit length of beam 0.0067kg/mm.

For each portion there are two modes of vibration, N_{xx} in the vertical plane and N_{yy} in the horizontal plane.

$$I_{xx} = 18.78 \times 10^3 mm^4$$

and $I_{yy} = 2.91 \times 10^3 mm^4$ for this size of channel (standard tables)

$$\therefore N_{xx} = \frac{3.57}{l^2} \times 75.399 \times 10^6 \text{ Hz}$$

and $$N_{yy} = \frac{3.57}{l^2} \times 29.68 \times 10^6 \text{ Hz}$$

Each part 1 side is made up of two 448.5mm lengths and one 246mm length and each part 2 side is made up of one 590.55mm, one 546.1mm and one 317.5mm length.

Table C.6. Natural frequencies of vibration of frame side member portion.

Position	l_{mm}	N_{xx} Hz	N_{yy} Hz
Pt 1 Ends (2)	448.5	1338.1	526.3
Middle	246.0	4447.8	1750.4
Pt.2 End	317.5	2121.0	1050.8
Middle	590.6	771.8	303.8
End	546.1	905.9	356.5

There are no positive self-stimulating resonances. Although the Pt 1 ends are nearly in first harmonic resonance with one Pt 2 end, in the N_{yy} mode, it is considered unlikely that self-stimulation will occur since the parts are at right angles to each other and do not form a continuum.

Table C.7. List of basic and harmonic frequencies in main frame in Hz.

Basic	Harmonic ½	Harmonic ⅓	Harmonic ¼
4447.8	2223.9	1482.6	1111.95
2121.0	1060.5	707.0	530.25
1750.4 *	875.2	583.47	437.6
1338.1	669.05	446.03	334.53
1050.8 *	525.4	350.27	262.7
905.9	452.95	301.97	226.48
771.8	385.9	257.27	192.95
526.3 *	263.15	175.43	131.58
356.5 *	178.25	118.83	89.13
303.8 *	151.9	101.27	75.95

* N_{yy} mode

Frequencies below the dividing line would be open to stimulation from gear chatter, but, in every case, the mode is modified, or modified and damped by superimposed bracing and/or attachments. Due to this modification the frequencies are likely to be raised by a factor of at least two, which would bring the danger range down to below the broken line i.e. no basic frequency is likely to be stimulated, and external damping is probably well able to stop resonance with any of the sensitive harmonics. Hence, it is considered unlikely that any serious vibrations will be set up in frame members.