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THE DESIGN, CONSTRUCTION AND CALIBRATION OF A THRUST AND TORQUE DYNAMOMETER FOR A WIND TUNNEL PROPELLER MODEL

by A.F. Molland and S.R. Turnock

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1. INTRODUCTION

The design, construction and calibration of a two-component strain gauge dynamometer is described. The dynamometer was to form part of a propeller experimental rig and be capable of measuring thrust and torque on the rotating propeller. The experimental rig would be used in a wind tunnel to investigate interactions between propellers and rudders (Ref. 1).

2. DESIGN LOADINGS AND SIZE

A survey of forces from the range of propellers likely to be tested in the rig led to the following maximum load requirements:

Thrust	:	750N
Torque	:	110 Nm
at max Revs	:	3000 rpm

Requirements for clearance and accessibility to the dynamometer led to a maximum diameter of 100mm.

3. MATERIAL AND DESIGN STRESSES

Alloy steel EN16R (IQ) was chosen as the dynamometer material, the thrust and torque portions being machined separately from solid bar. The material was machined in its 'R' (hardened and tempered) form. The mechanical properties of EN16R are summarised as follows:

UTS	:	700-850 N/mm^2
Yield Stress	:	525 N/mm^2
E	:	2.06×10^5 N/mm^2

As a starting point to size the flexures, the maximum strain at flexure ends was assumed to be 0.0010 (1000 $\mu\epsilon$), leading to a stress of:

$$\sigma = E \times \epsilon = 2.06 \times 10^5 \times 0.0010 = 206 \text{ N/mm}^2.$$

This was less than half the yield stress, half yield stress (263 N/mm^2) being assumed as the maximum for design purposes.

4. FLEXURE DESIGN AND DIMENSIONS

4.1 Thrust Component

The thrust dynamometer is, in concept, a wheel with four 'spokes', the spokes forming the four flexures and bending in contraflexure due to applied thrust (Fig. 1). Such a concept also has a good resistance to shear due to applied torque. The need for caution on contraflexure stress levels and deflections was noted with this concept since, with large deflections, true bending of the flexures in contraflexure is resisted and significant membrane stresses may arise. Design stress levels led to a flexure width of 12.0mm, thickness 2.2mm and an overall length of 25mm which, with 2mm radii at each end, gave an effective flexure length of 21mm. The accuracy of manufacture of flexure thickness was within $\pm$ 0.0025mm. The estimated maximum strain at flexure ends (21mm length) when bending in contraflexure due to a thrust of 750N was 987 $\mu\epsilon$ (σ = 203 N/mm^2). The siting of the gauges away from the ends of the flexures (to allow for gauge backing) led to a final estimate of mean maximum strain at the gauge location of 705 $\mu\epsilon$. Summaries of the calculations are given in Appendix 1.

4.2 Torque Component

A torque cage concept with four flexures working in contraflexure was adopted, Fig. 1. Design stress levels led to a flexure width of 12.0mm, thickness 4.0mm and an overall length of 25mm which, with 2mm radii at each end, gave an effective flexure length of 21mm. The accuracy of manufacture of flexure thickness was within $\pm$ 0.0025mm. The estimated maximum strain at flexure ends (21mm length) when bending in contraflexure due to a torque of 110 Nm was 1068 $\mu\epsilon$ (σ = 220 N/mm^2). The siting of the gauges away from the ends of the flexures (to allow for gauge backing) led to a final estimate of mean maximum strain at the gauge location of 763 $\mu\epsilon$. Summaries of the calculations are given in Appendix 1.

5. GAUGE SIZE, LOCATION AND BRIDGE CIRCUITS

5.1 Size

3mm steel foil gauges were chosen. These offered a suitable compromise between the facility to locate a small gauge close to the position of maximum strain at the ends of the flexures and large enough to provide adequate heat dissipation properties. The larger gauge also offers the possibility of more accurate alignment during the attachment process, thus minimising possible errors and component interactions due to misalignment.

The gauges chosen and used (TML type FCA-3-11) had a length of 3.0mm, width of 1.8mm, resistance of 120Ω and a gauge factor of 2.1. The gauges were bonded to the flexures using polyester adhesive (P2). After bonding, all gauges were coated with silicone rubber compound for sealing and protection.

5.2 Location and Circuits

The gauges were located on the flexures and the bridge circuits wired as shown in

Figs. 2a and 2b. The gauges are arranged in the thrust and torque bridge circuits (Fig. 2b) whereby they react to stresses due to contraflexure but cancel when subjected to shear and tensile/compressive stresses in the flexures. This arrangement also provides temperature compensation. A further property of the chosen circuits (described in Ref. 2) is that, assuming the flexures are perfectly similar and the gauges perfectly aligned, the thrust and torque readings should be independent of the direction of application of the thrust. In other words the thrust and torque readings should be insensitive to any misalignment in the shafting either side of the dynamometer.

5.3 Bridge Excitation voltage and Heat Dissipation

Two gauges per bridge arm were used (Fig. 2b). Besides the cancellation benefits of the circuits, mentioned in the previous section, the use of two gauges in series per arm leads to an increased total equivalent resistance of the bridge. This allows an increased supply voltage for the same gauge current, with consequent increase in total bridge output. A maximum bridge excitation voltage of 7 volts was assumed; this leads to a total current of 29.2 ma, a current per arm of 14.6 ma and a power dissipation of 0.47 watts/cm^2. A summary of the calculations is given in Appendix 2. Recommended maximum values for power dissipation reviewed and reported on in Ref. 3 ranged from about 0.2 to 0.8 watts/cm^2, with 0.8 watts/cm^2 being a satisfactory maximum for flexures with good heat sink properties such as aluminium. A maximum value of 0.47 watts/cm^2 would therefore appear to be reasonable for satisfactory heat dissipation on the steel flexures being employed.

5.4 Date Acquisition/Instrumentation

The bridge circuits were wired up according to Figs. 2b, 3a and 3b. Data signals are carried from the dynamometer to the acquisition system via slip rings located on the shaft close to the dynamometer. Instrumentation comprises a strain gauge bridge unit, a data transfer unit and a digital volt meter. The signals are finally transmitted to a computer based automated acquisition/analysis system.

6. CALIBRATION

6.1 General

The calibration frame described in Ref. 2 was used for the calibration programme. Arrangement of the supports and pulleys allowed thrust to be applied (in the positive direction) and torque (as a couple) to be applied at lever arms of $\pm$203.2mm.

All calibrations were carried out in the static (non-rotating) condition.

The bearings between the propeller (i.e. point of loading) and the dynamometer are axially unconstrained roller bearings. However, at zero or low revolutions frictional resistance in these bearings is enough to produce spurious readings for thrust in the zero or lightly loaded non-rotating condition. This problem did not arise with the torque component. During calibration of the thrust component it was therefore necessary to impart some vibration to the rig before taking zero readings or readings at low thrust levels. This overcame the problem during the calibration procedure.

6.2 Calibration Slopes

The mean calibration slopes for thrust and torque, Figs. 4a and 4b, are linear within acceptable limits. The derived slopes are as follows:

Thrust	:	11.07 μv/N (0.0903 N/μv)
Torque	:	77.87 μv/Nm (0.01284 Nm/μv)

Hysteresis effects were found to be negligible during the loading/offloading procedures.

The torque component did display some non-linearity at the highest loadings. However, under high loadings the torque calibration arm was able to twist slightly relative to the locked shaft, thus decreasing the effective lever. This led to a decrease in the actual applied moment and torque output signal at high loadings as seen in Fig. 4b.

Due to limitations of the calibration rig the maximum thrust applied was 741 N whilst the maximum torque was 103 Nm.

6.3 Interactions

During calibration, the interactions of thrust on torque and torque on thrust were monitored. Changes in the readings for one component due to application of the other were always very small and within the measurement accuracy. It was therefore concluded that interactions, if any, were minimal and could be neglected.

6.4 Combined Loads

Calibration of one component during simultaneous loading of the other component was checked. Changes in the calibration slopes were within the measurement accuracy and could therefore be neglected.

6.5 Comparison Between the Predicted and Derived Calibration Slopes

Calculations for the predicted calibration slopes are given in Appendix 3. The derived calibration slope for thrust of 11.07 μv/N is about 20% less than the predicted (13.80 μv/N), whilst that for torque is about 24% less than the predicted (120.0 μv/Nm).

The main reason for the differences is likely to lie in the use of linear beam theory for the predicted values. This is supported by the fact that the difference between the derived and predicted values is larger for the torque flexures which have a lower length to thickness ratio than that of the thrust flexures. Possible further contributory causes are inaccurate siting of the gauges (unlikely) and the assumption that the mean maximum the gauges (unlikely) and the assumption that the mean maximum strain is at the mid length of the gauge, the assumption for gauge factor (which cannot be checked), the precise properties of the flexure material, the assumption for end fixity of the flexures and the influence of the radii at the ends of the flexures.

Whilst the predictions are only within about 25% of the derived, they do however provide some guidance at the preliminary design stage for the likely output signal levels.

6.6 Derivation of Thrust Zeros During Practical Operation

The effect of friction in the roller bearings on the thrust zeros, mentioned in Section 6.1, was successfully overcome during calibration by forced vibration of the rig. This effect also has to be overcome in operation when it is not practical to vibrate the rig. Spurious thrust zeros at zero rpm are then likely to follow. The frictional effect is effectively removed when the shaft and dynamometer are rotating. During operation, therefore, a basic property of the propeller was utilised. Namely, at zero wind speed thrust is proportional to the square of the revolutions. The propeller was run up at zero wind speed and the thrust measured for at least three levels of revolution. The intercept of a plot of thrust to a base of $(revs)^2$ yielded the true zero thrust reading. This method was found to be satisfactory and repeatable in practice.

7. COMMISSIONING OF DYNAMOMETER

Commissioning of the propeller rig took place in March 1990.

The dynamometer performed successfully during these tests and satisfactory data were obtained. Repeatability, meaning the ability to repeat the data for a particular condition, was also satisfactory.

ACKNOWLEDGEMENTS

The work described in this report covers part of a reserch project funded by the SERC/MOD through the Marine Technology Directorate Ltd under Research Grant Ref. No. GR/E/65289.

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1. Turnock, S.R.: "A Test Rig for the Investigation of Propeller/Rudder Interaction". University of Southampton, Ship Science Report no. 45, 1990.

2. Perry, C.C. and Lissner, H.R.: "The Strain Gage Primer". McGraw-Hill Book Co., 1962.

3. Molland, A.F.: "The Design Construction and Calibration of a Five-Component Strain Gauge Wind Tunnel Dynamometer". University of Southampton, Ship Science Report No. 1/77, 1976.

APPENDIX 1

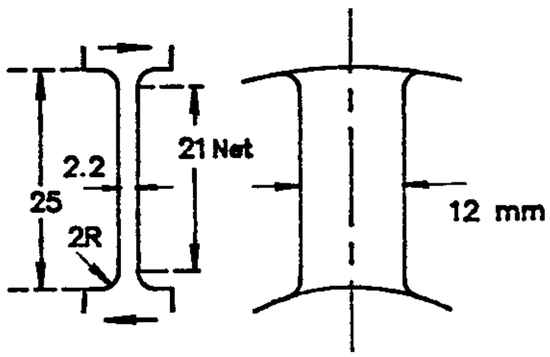
OUTLINE CALCULATIONS FOR THE DIMENSIONS OF FLEXURES

THRUST FLEXURES:

Maximum Design Thrust = 750N

Assuming 4 flexures, thrust/flexure $= \frac{750}{4} = 187.5N$

Flexure Dimensions: length 25mm (21mm net)
section 12mm x 2.2mm (Fig. 1)



In contraflexure:

$$\text{Root strain } \epsilon = \frac{Fl}{2E(I/y)} \quad \text{(Ref. 3)}$$

$$I/y = \frac{wt^2}{6} = \frac{12 \times 2.2^2}{6} = 9.68\text{mm}^3$$

$$\epsilon = \frac{Fl}{2E(I/y)} = \frac{187.5 \times 21}{2 \times 2.06 \times 10^5 \times 9.68} = 0.000987 = 987\mu\epsilon$$

$(= 203\ \text{N/mm}^2)$

$$I = \frac{wt^3}{12} = \frac{12 \times 2.2^3}{12} = 10.648 \text{ mm}^4$$

$$\text{Deflection } \delta = \frac{Fl^3}{12\,EI} = \frac{187.5 \times 21^3}{12 \times 2.06 \times 10^5 \times 10.648} = 0.066\text{mm}$$

$$\text{Shear stress (ford/aft)} = \frac{187.5}{12 \times 2.2} = 7.1 \text{ N/mm}^2$$

$$\text{Max torque force/flexure} = \frac{110/4}{0.015} = 1833.3 \text{ N}$$

$$\text{Shear stress (rotation)} = \frac{1833.3}{12 \times 2.2}$$

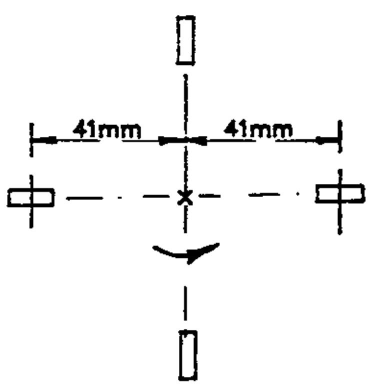
$$= 69.4 \text{ N/mm}^2$$

The above stresses are considered acceptable for alloy steel EN16R. (Yield stress ≑ 525 N/mm^2, Ult. shear stress ≑ 400 N/mm^2).

TORQUE FLEXURES:

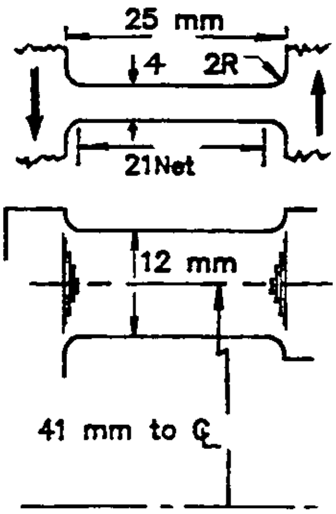
Maximum Design Torque = 110 Nm

Flexures have mean radius of 41mm (0.041m), Fig. 1.



$$\text{Torque Force/Flexure} = \frac{110/4}{0.041}$$

$$= 670.7 \text{ N per flexure}$$



$$I/y = \frac{wt^2}{6} = \frac{12 \times 4^2}{6} = 32.0mm^3$$

$$I = \frac{wt^3}{12} = \frac{12 \times 4^3}{12} = 64.0mm^4$$

In contraflexure:

$$\text{Root strain } \epsilon = \frac{Fl}{2E(I/y)}$$

$$= \frac{670.7 \times 21}{2 \times 2.06 \times 10^5 \times 32.0}$$

$$= 0.001068$$

$$= 1068\ \mu\epsilon$$

$$(220\ N/mm^2)$$

$$\text{Deflection } \delta = \frac{Fl^3}{12\ EI} = \frac{670.7 \times 21^3}{12 \times 2.06 \times 10^5 \times 64.0}$$

$$= 0.039mm$$

$$\text{Tensile/Compressive stress due to thrust} = \frac{750/4}{12 \times 4} = 3.91\ N/mm^2$$

Shear stress (rotation) $= \dfrac{670.7}{12 \times 4} = 14.0\ \text{N/mm}^2$

The above stresses are considered acceptable for alloy steel EN16R (yield stress $\doteqdot$ 525 N/mm^2, ult shear stress $\doteqdot$ 400 N/mm^2).

EFFECTIVE MEAN STRAIN IN WAY OF GAUGES:

The 3mm gauges are sited 1.5mm from end of 21mm effective gauge length, Fig. 2a.

At mid gauge position, mean strain as a percentage of maximum at ends (of 21mm flexure length)

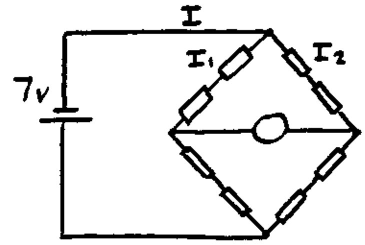
$$= \frac{\frac{21}{2} - 1.5 - \frac{3}{2}}{10.5} \times 100 = 71.4\%$$

Thus mean maximum strain at gauge location:

for the thrust flexures = 987 x 0.714 = 705 $\mu\epsilon$
and for the torque flexures = 1068 x 0.714 = 763 $\mu\epsilon$.

APPENDIX 2

OUTLINE CALCULATIONS FOR BRIDGE CIRCUITS



Assume two 120Ω gauges per arm as shown.
Equivalent total resistance:

$$\frac{1}{R} = \frac{1}{480} + \frac{1}{480} = \frac{1}{240}$$

$\therefore$ Equiv. resist. R = 240Ω

Assuming Excitation voltage = 7v
I = V/R = 7/240 = 0.0292 amps (29.2 ma)

and $I_1 = I_2 = 0.0146$ amps (14.6 ma)

Power/gauge $w = IV = I^2R = 0.0146^2 \times 120 = 0.02558$ watts

Gauge area = 3mm x 1.8mm (0.3 cm x 0.18 cm)

and power dissipation/cm^2 $= \dfrac{0.02558}{0.3 \times 0.18} = 0.474$ watts/cm^2

According to Ref. 3 this is an acceptable power dissipation level.

Also, with bridge excitation 7v, gauge factor ÷ 2, max strain ϵ ÷ 700 $\mu\epsilon$
Output voltage $\Delta v = v.k.\epsilon = 7 \times 2 \times 700 = 9800\ \mu v$.

It is noted that if one gauge per arm were used then, for the same voltage input, the current level and hence power dissipation would be unacceptably high. It was also pointed out in Section 5.2 that the gauges in the two gauges per arm circuit may be arranged whereby bridge response is independent of the direction of the application of thrust.

APPENDIX 3

PREDICTED CALIBRATION SLOPES

(NOTE: These are not the actual derived calibration slopes, which are presented in a separate section).

THRUST COMPONENT:

From Appendix 1, for a thrust of 750N, the estimated mean maximum strain at gauge location $\epsilon = 705\ \mu\epsilon$.

Assuming a bridge excitation voltage $v = 7.0$ volts and gauge factor $k \doteqdot 2.1$

then output voltage $\Delta v = vk\epsilon$

$= 7.0 \times 2.1 \times 705$

$= 10364\ \mu v$

and Predicted Calibration Slope $= \dfrac{10364}{750}$

$= 13.8\ \mu v/N$

TORQUE COMPONENT:

From Appendix 1, for a toque of 110 Nm, the estimated mean maximum strain at gauge location $\epsilon = 763\mu\epsilon$.

Assuming a bridge excitation voltage $v = 7.0$ volts and gauge factor $k \doteqdot 2.1$.

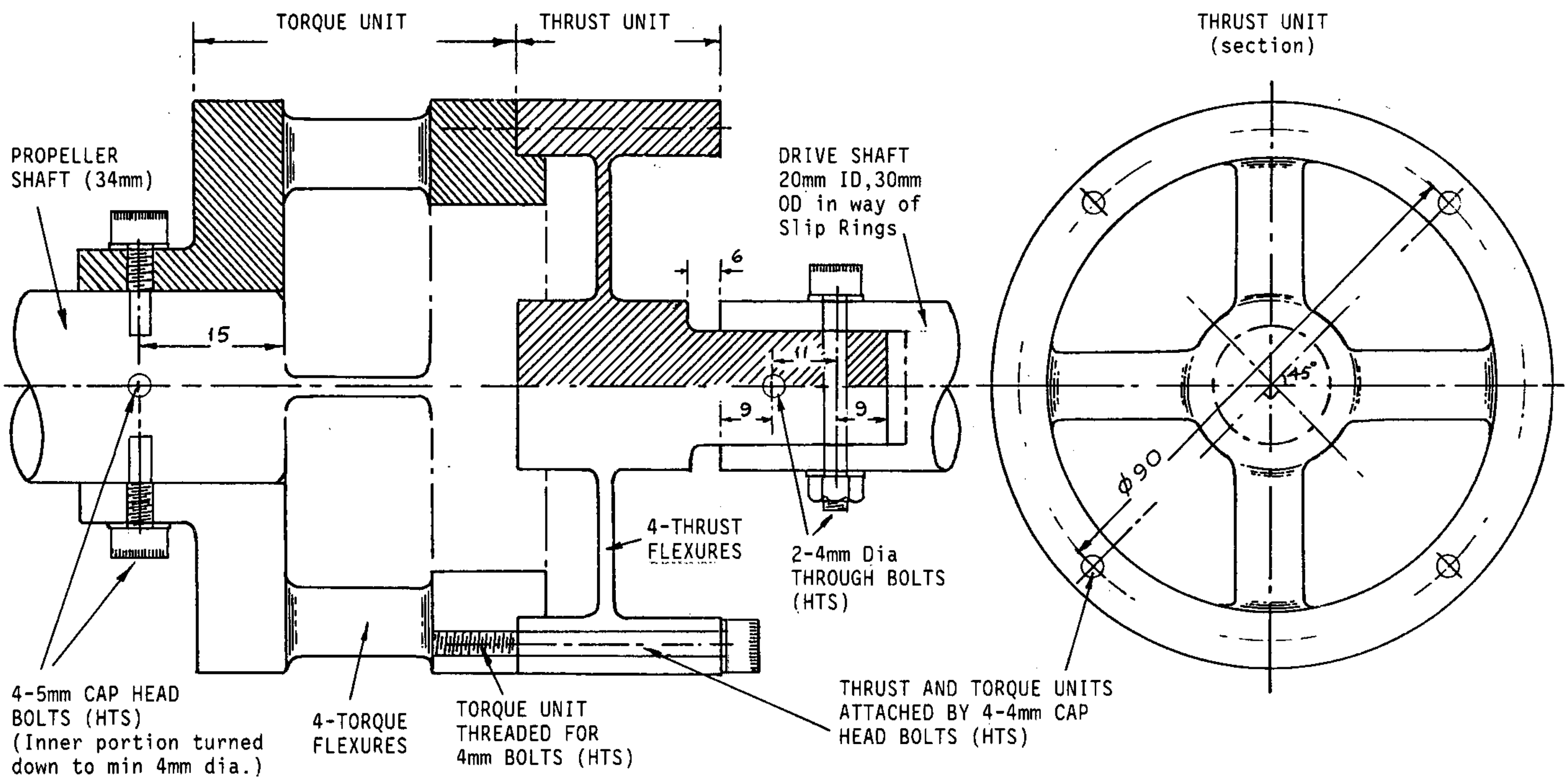
then output voltage $\Delta v = vk\epsilon$

$= 7.0 \times 2.1 \times 763$

$= 11216\ \mu v$

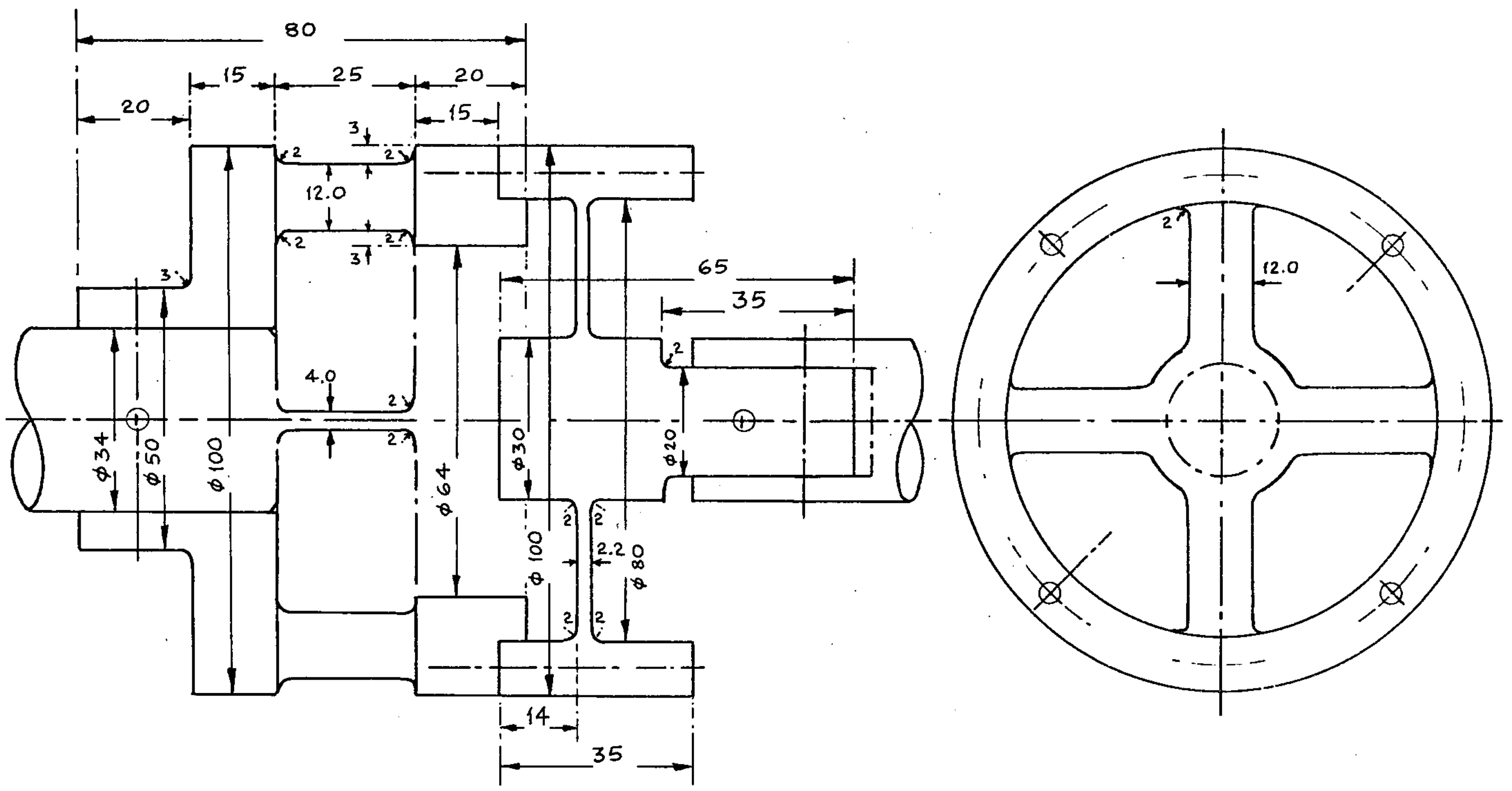
and Predicted Calibration Slope $= \dfrac{11216}{110}$

$= 102.0\ \mu v/Nm$.



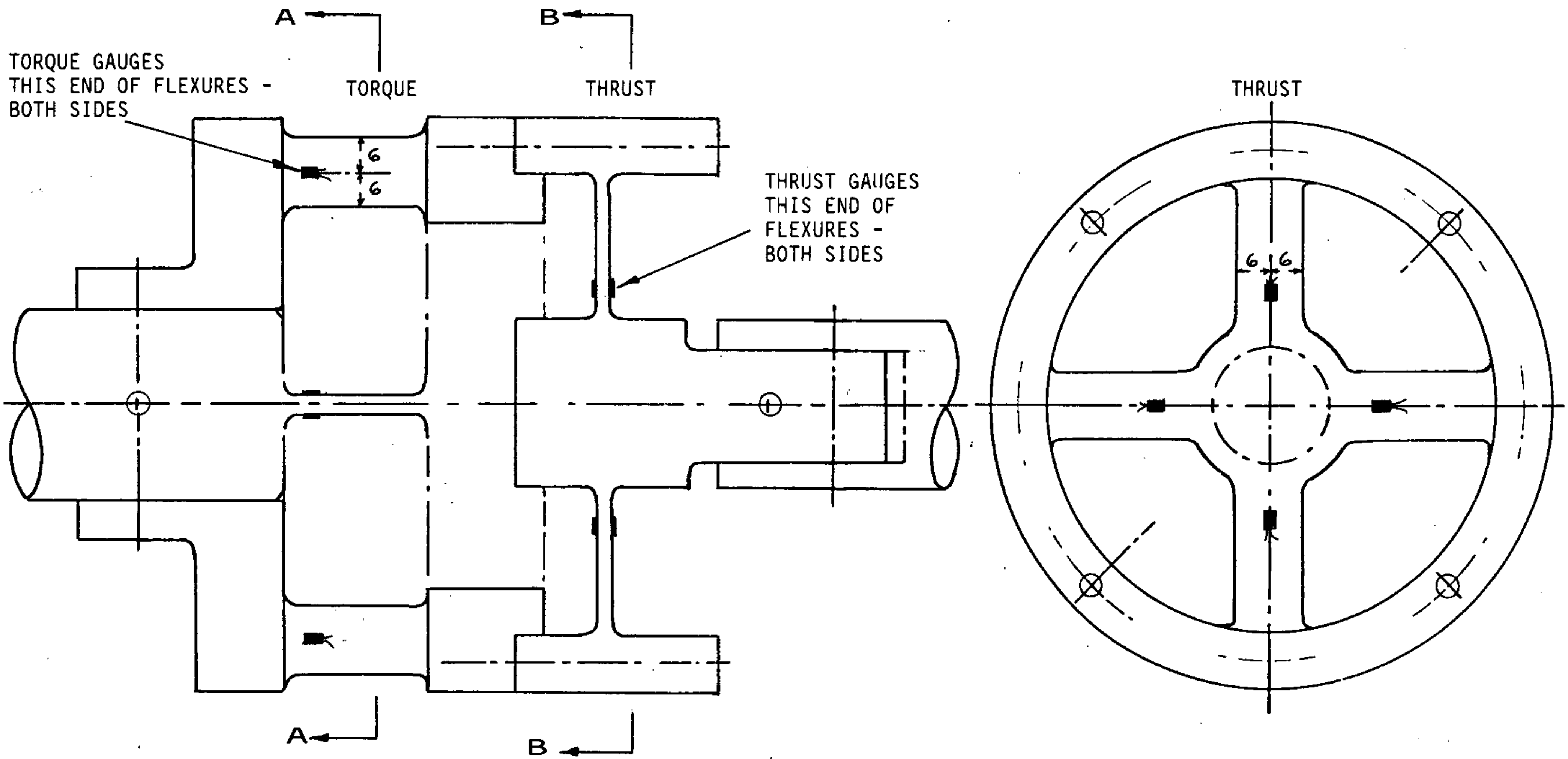
DYNAMOMETER MATERIAL : ALLOY STEEL EN16R IQ

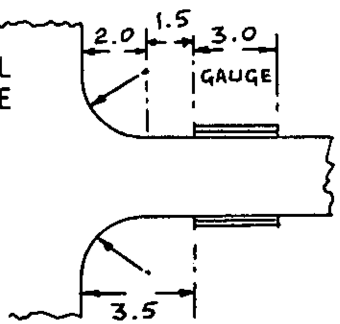
Fig.1a DYNAMOMETER-GENERAL ARRANGEMENT and BOLTED CONNECTIONS



ALL DIMENSIONS IN mm

Fig.1b DYNAMOMETER - DIMENSIONS





LOCATION OF ALL
GAUGES RELATIVE
TO ENDS OF
FLEXURES

See Next Page for
SECTIONS A-A, B-B

GAUGES : TML Type FCA-3-11
ADHESIVE: POLYESTER P2

Fig.2a LOCATION OF STRAIN GAUGES

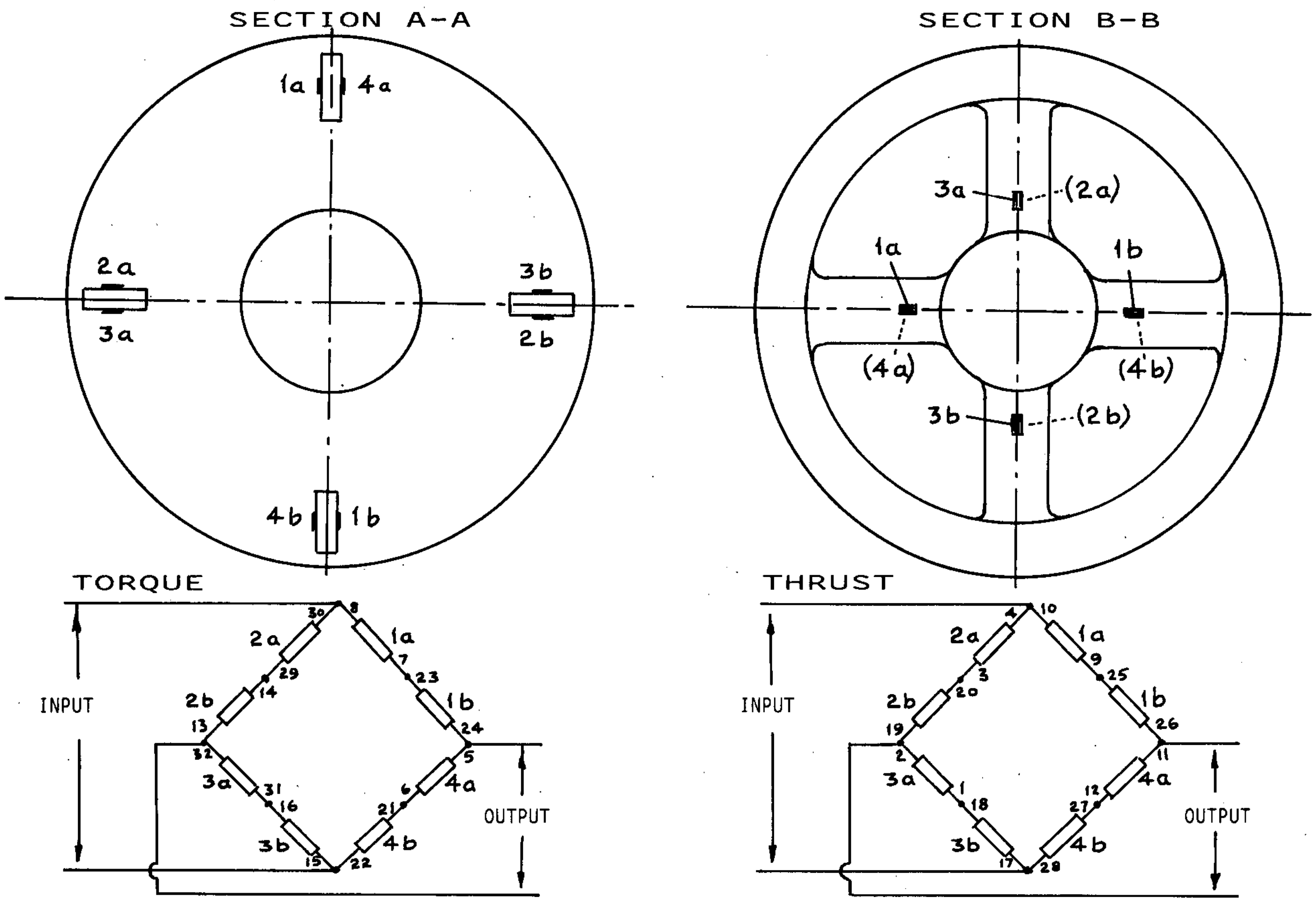
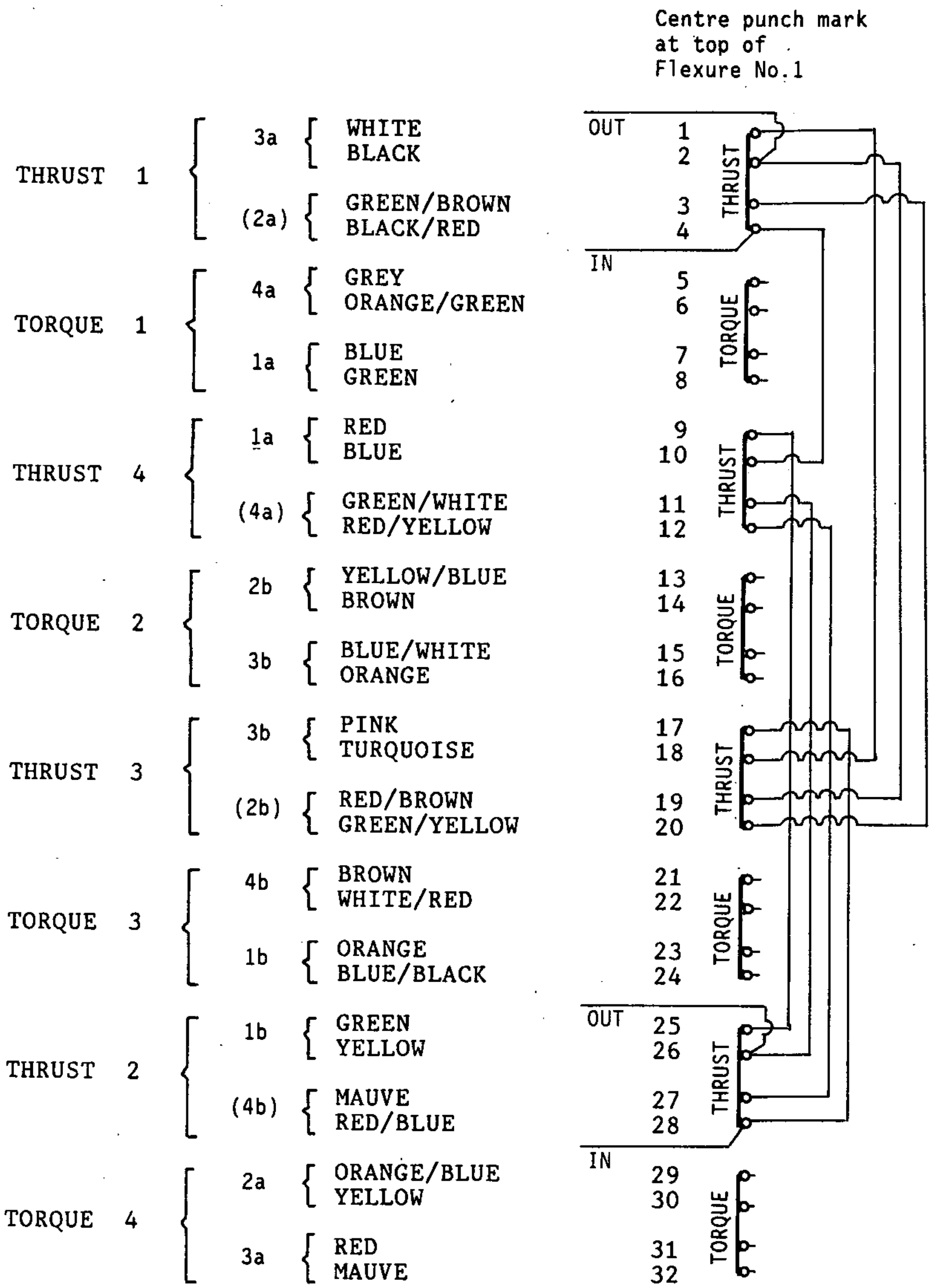


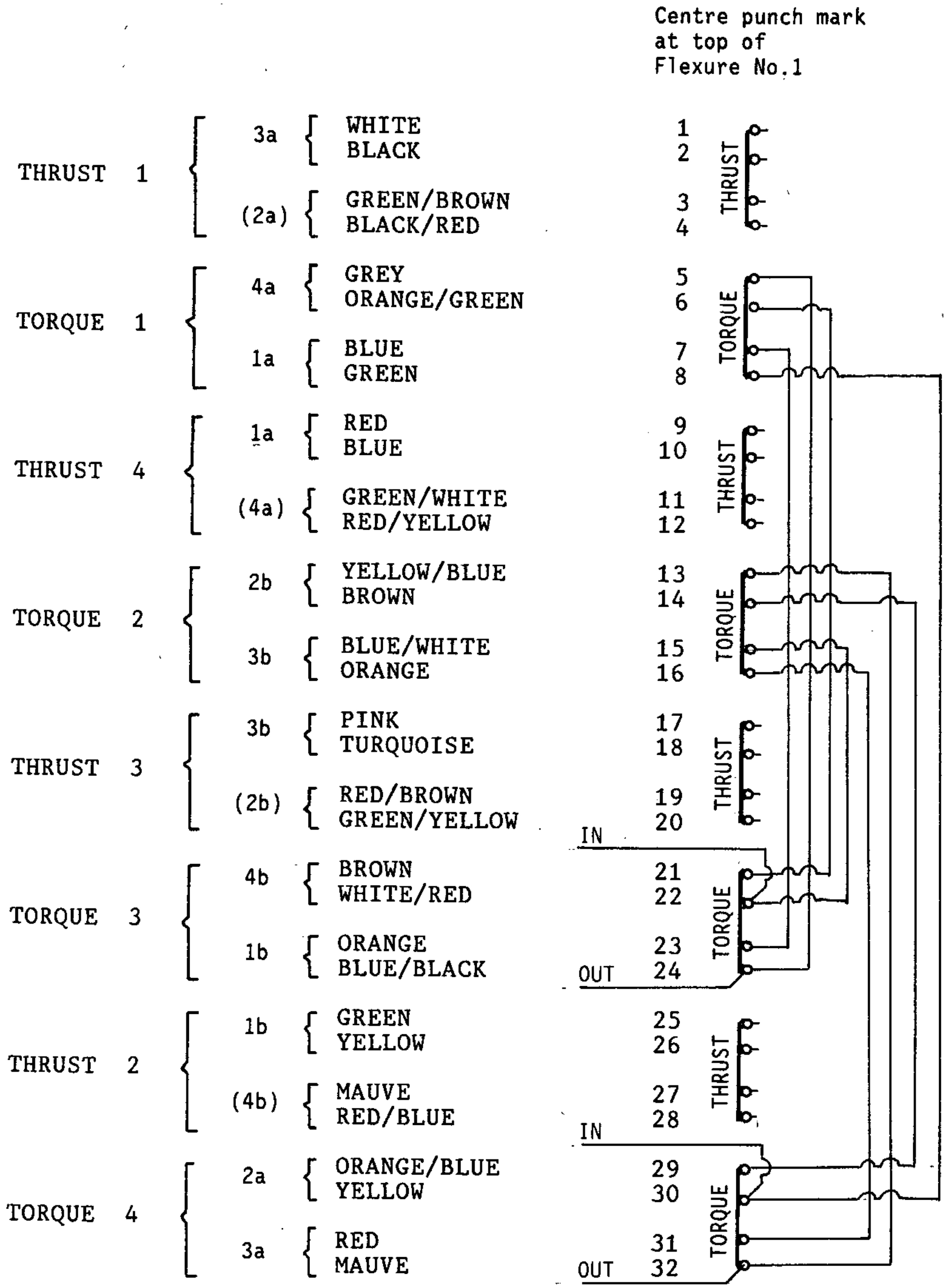
Fig.2b LOCATION OF STRAIN GAUGES AND BRIDGE CIRCUITS



Flexure Arm Numbers running clockwise when viewed from inside the dynamometer halves

Tag strip identification running clockwise from Flexure No.1 on Thrust Half when viewed from Slip Rings

Fig.3a BRIDGE WIRING – THRUST UNIT
(WHITE WIRE)



Flexure Arm Numbers running clockwise when viewed from inside the dynamometer halves

Tag strip identification running clockwise from Flexure No.1 on Thrust Half when viewed from Slip Rings

FIG.3b BRIDGE WIRING - TORQUE UNIT
(BLACK WIRE)

THRUST

Re-Calibration via leads and slip rings - 17/8/90

Excitation Voltage : 7.0v

Calibration Slope : 11.07 μv/N (0.0903 N/μv)

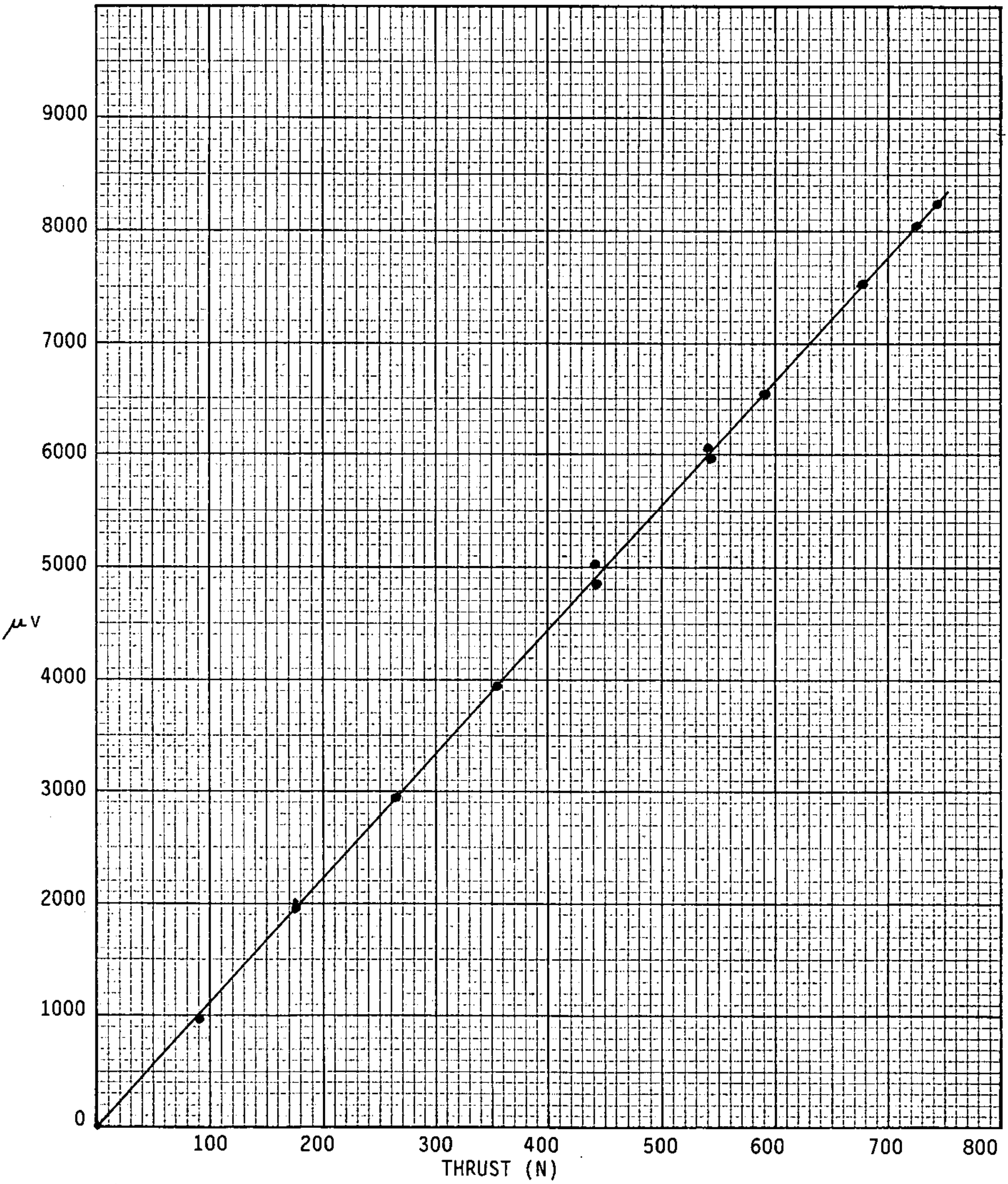


Fig.4a THRUST CALIBRATION

TORQUE

Re-Calibration via slip rings - 17/8/90

Excitation Voltage : 7.0v

Calibration Slope : 77.87 μv/Nm (0.0128 Nm/μv)

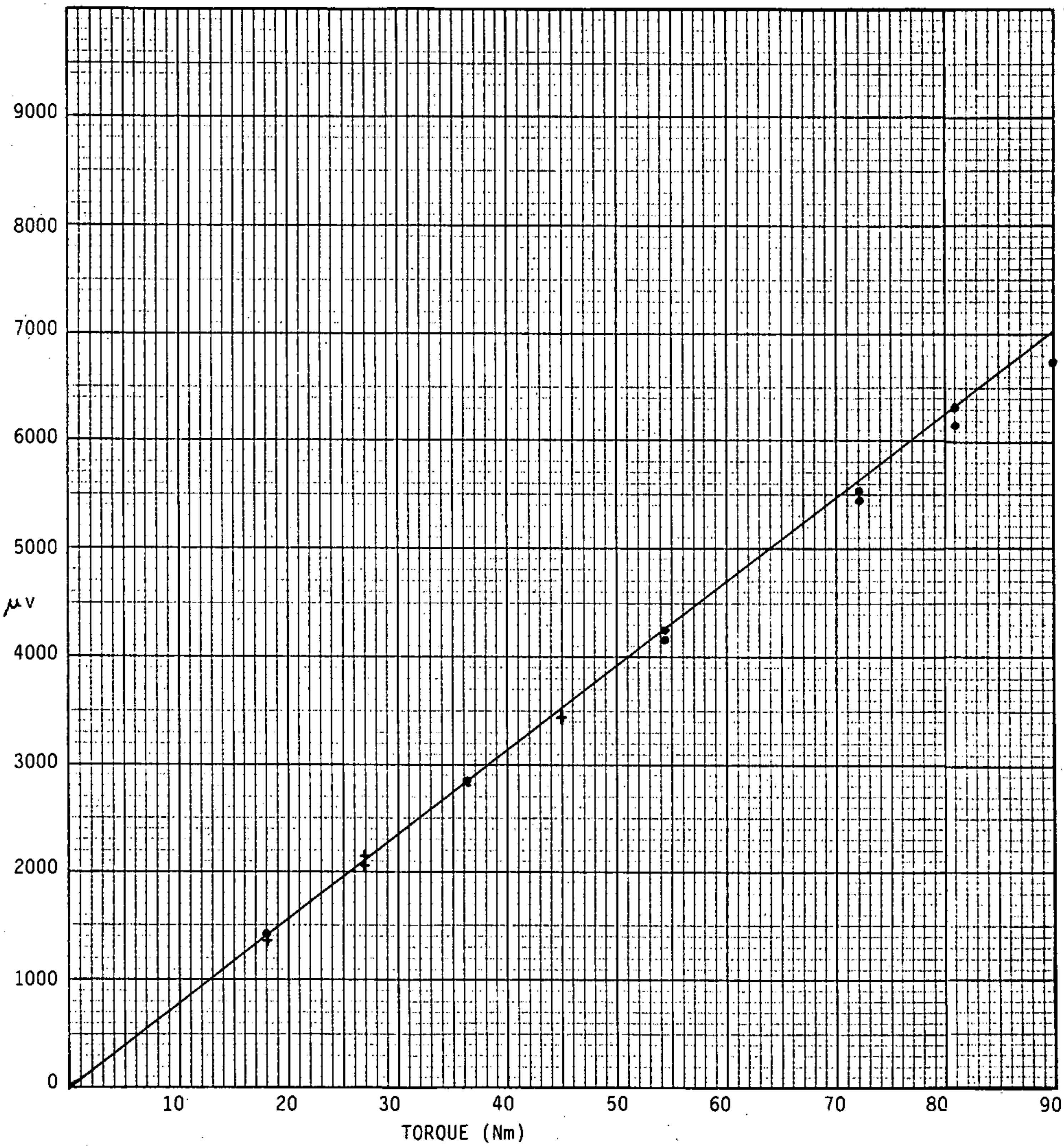


Fig.4b TORQUE CALIBRATION

APPENDIX 1

OUTLINE CALCULATIONS FOR THE DIMENSIONS OF FLEXURES

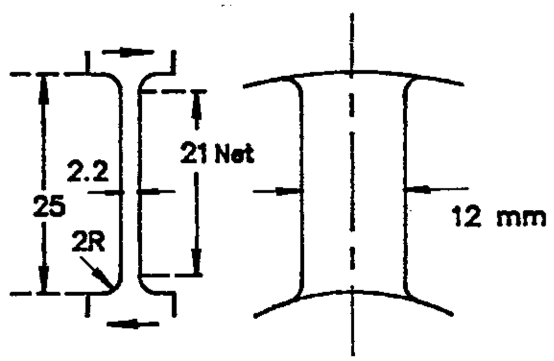
THRUST FLEXURES:

Maximum Design Thrust = 750N

Assuming 4 flexures, thrust/flexure = $\frac{750}{4}$ = 187.5N

Flexure Dimensions: length 25mm (21mm net)

section 12mm x 2.2mm (Fig. 1)



In contraflexure:

Root strain $\epsilon = \frac{Fl}{2E(I/y)}$ (Ref. 3)

$$I/y = \frac{wt^2}{6} = \frac{12 \times 2.2^2}{6} = 9.68\text{mm}^3$$

$$\epsilon = \frac{Fl}{2E(I/y)} = \frac{187.5 \times 21}{2 \times 2.06 \times 10^5 \times 9.68} = 0.000987 = 987\mu\epsilon$$

(= 203 N/mm^2)

$$I = \frac{wt^3}{12} = \frac{12 \times 2.2^3}{12} = 10.648 \text{ mm}^4$$

$$\text{Deflection } \delta = \frac{Fl^3}{12\,EI} = \frac{187.5 \times 21^3}{12 \times 2.06 \times 10^5 \times 10.648} = 0.066\text{mm}$$

$$\text{Shear stress (ford/aft)} = \frac{187.5}{12 \times 2.2} = 7.1 \text{ N/mm}^2$$

$$\text{Max torque force/flexure} = \frac{110/4}{0.015} = 1833.3 \text{ N}$$

$$\text{Shear stress (rotation)} = \frac{1833.3}{12 \times 2.2}$$

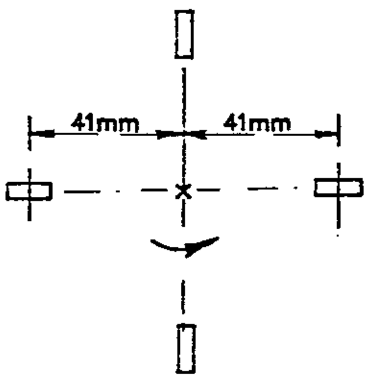
$$= 69.4 \text{ N/mm}^2$$

The above stresses are considered acceptable for alloy steel EN16R. (Yield stress ≑ 525 N/mm^2, Ult. shear stress ≑ 400 N/mm^2).

TORQUE FLEXURES:

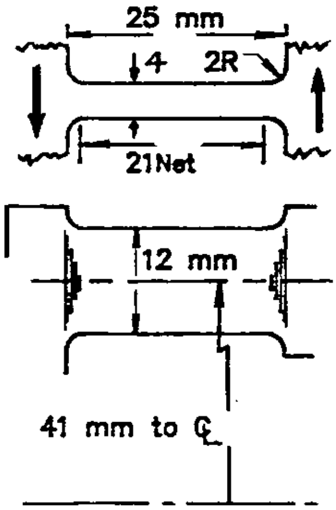
Maximum Design Torque = 110 Nm

Flexures have mean radius of 41mm (0.041m), Fig. 1.



$$\text{Torque Force/Flexure} = \frac{110/4}{0.041}$$

$$= 670.7 \text{ N per flexure}$$



$$I/y = \frac{wt^2}{6} = \frac{12 \times 4^2}{6} = 32.0\text{mm}^3$$

$$I = \frac{wt^3}{12} = \frac{12 \times 4^3}{12} = 64.0\text{mm}^4$$

In contraflexure:

$$\text{Root strain } \epsilon = \frac{Fl}{2E(I/y)}$$

$$= \frac{670.7 \times 21}{2 \times 2.06 \times 10^5 \times 32.0}$$

$$= 0.001068$$

$$= 1068\ \mu\epsilon$$

$(220\ \text{N/mm}^2)$

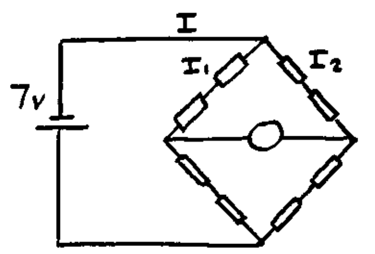
$$\text{Deflection } \delta = \frac{Fl^3}{12\ EI} = \frac{670.7 \times 21^3}{12 \times 2.06 \times 10^5 \times 64.0}$$

$$= 0.039\text{mm}$$

$$\text{Tensile/Compressive stress due to thrust} = \frac{750/4}{12 \times 4} = 3.91\ \text{N/mm}^2$$

APPENDIX 2

OUTLINE CALCULATIONS FOR BRIDGE CIRCUITS



Assume two 120Ω gauges per arm as shown.
Equivalent total resistance:

$$\frac{1}{R} = \frac{1}{480} + \frac{1}{480} = \frac{1}{240}$$

$\therefore$ Equiv. resist. R = 240Ω

Assuming Excitation voltage = 7v
I = V/R = 7/240 = 0.0292 amps (29.2 ma)

and $I_1 = I_2 = 0.0146$ amps (14.6 ma)

Power/gauge $w = IV = I^2R = 0.0146^2 \times 120 = 0.02558$ watts

Gauge area = 3mm x 1.8mm (0.3 cm x 0.18 cm)

and power dissipation/cm^2 $= \dfrac{0.02558}{0.3 \times 0.18} = 0.474$ watts/cm^2

According to Ref. 3 this is an acceptable power dissipation level.

Also, with bridge excitation 7v, gauge factor $\doteqdot$ 2, max strain $\epsilon \doteqdot 700\ \mu\epsilon$
Output voltage $\Delta v = v.k.\epsilon = 7 \times 2 \times 700 = 9800\ \mu V$.

It is noted that if one gauge per arm were used then, for the same voltage input, the current level and hence power dissipation would be unacceptably high. It was also pointed out in Section 5.2 that the gauges in the two gauges per arm circuit may be arranged whereby bridge response is independent of the direction of the application of thrust.

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THE DESIGN, CONSTRUCTION AND CALIBRATION OF A THRUST AND TORQUE DYNAMOMETER FOR A WIND TUNNEL PROPELLER MODEL

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~~Fig 1b~~
~~All Dimensi in mm~~

~~Fig 4b Tippex~~

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