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BOREHOLE SHEAR DEVICE PHASE II DEVELOPMENT.(U)
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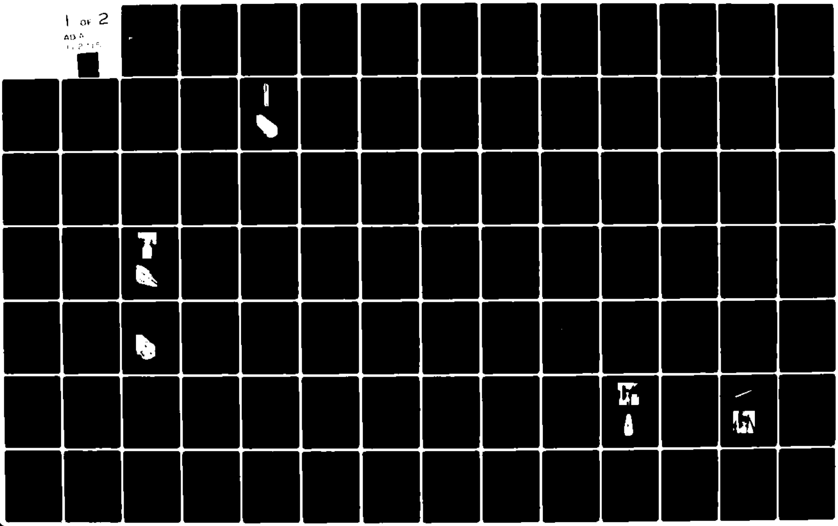

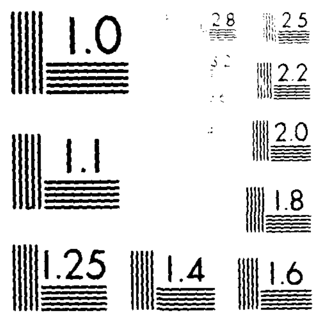
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BOREHOLE SHEAR DEVICE PHASE II DEVELOPMENT

Roger Sidey

Dames & Moore
445 South Figueroa Street
Los Angeles, CA 90071

Final Report

February 1982

Approved for public release; distribution unlimited.

AIR FORCE WEAPONS LABORATORY
Air Force Systems Command
Kirtland Air Force Base, NM 87117

AFWL-TR-81-74

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This technical report has been reviewed and is approved for publication.

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The report describes the phase II development of the borehole shear device. Further design development work and in some cases hardware fabrication is documented and covers the following aspects of the device: Dynamic and static actuators, self boring attachment, modifications to borehole shear device probe, torque tube, couplings and instrumentation.

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PREFACE

The author wishes to acknowledge the invaluable contribution of the engineers who participated with him on the design and development of the equipment described in this report. Much is owed to their unstinted effort. Thus, to Mr David J. Rainford for his work on the Borehole Shear Device probe and the static actuator unit, and to Mr Jack M. Hale for his work on the dynamic actuator system, many thanks. Also within Dames & Moore thanks are due to; Mr Tony F. Abbs and Dr David C. Andrews for managing the project, Miss Susie Backhouse for her diligent work in illustration and organisation of the report, to Miss Sylvia Nicholson for typing the report, and to Mr David V. Hinds for his willing assistance with the many practical problems that arose.

Thanks are also due to Dr R.H. Bassett of the Civil Engineering Faculty of Kings College, London for his advice and assistance with the experimental work in the project, and to PM In Situ Limited of Cambridge, United Kingdom for their involvement on the aspects of self boring.

For the professional photographic work, thanks to Mr Colin Maker of Mortlake, London.

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CONTENTS

ILLUSTRATIONS

ILLUSTRATIONS (Concluded)

I. INTRODUCTION

1. BACKGROUND

This report represents the second phase of the development of the Borehole Shear Device (BSD). It is a follow on from a first phase study undertaken by Dames & Moore under contract number F29601-78-C-0058, AD A090 647 "Borehole Shear Device Feasibility and Preliminary Studies", (AFWL-TR-79-48). In this first phase the concept of the torsionally driven cylinder as a means for determining the in situ shear modulus of soils over a broad range of strains was investigated. The contract scope of work encompassed mathematical and computer modeling, system design studies and operational specifications for the equipment. Finally the construction and testing of some prototype hardware was undertaken. The outcome of this effort was the identification of a suitable configuration for the apparatus, and the development of certain analytical tools for interpretation of the field data.

Much of the content of this report concerns the detailed aspects of design and development of hardware related to the BSD test according to the specifications or guidelines of the preliminary study. The underlying concepts, while being briefly discussed or introduced, are not explored. For this the reader is referred to the first phase report.

2. SCOPE OF WORK

As mentioned above, the effort involved in this follow on contract has been primarily that of hardware design and development. The statement of work of the subject contract divides this effort into two subprograms or development areas:

a. Modification and continued development of the BSD probe.

b. Design and development of the actuator systems.

For the development of the BSD probe a certain amount of prototype hardware was in existence. This was due to the first phase contract, and it consisted of a full scale prototype of the coupling cylinder section of the probe. While the basic components of this assembly were suitable for use in the field version of the apparatus, a relatively large amount of the hardware required addition and modification. Examples of this were the modifications to allow axial loading of the shear coupling shoes, the implementation of a hydraulic retraction system, and a means for providing a soil particle barrier for the moving parts of the system. In addition to modification and development of the coupling cylinder hardware, there was the support or accompanying hardware which makes up the remainder of the in situ apparatus. While not being as much the heart of the equipment as the coupling cylinder, these systems are vital to the BSD test. The following are examples of the design requirements that were identified in these areas:

a. A method of terminating and distributing the many electrical and hydraulc circuits within the apparatus, and for connecting them to the uphole lines.

b. The means for coupling the torsion tubes to each other and to the probe together with the system required for transportation of soil cuttings (due to the self-boring process) through the probe.

c. A method of fitting a self-boring attachment to the probe and of lifting the soil cuttings through the apparatus.

d. The instrumentation for torque and rotation measurement and for monitoring normal loads (push-down) and jacking (lateral) pressure in the system.

The matter of the design and development of the actuator for the system was the subject of some fundamental study in the early part of the contract work. The maximum design torque for the BSD apparatus is high leading to an inevitably bulky and, hence, relatively slow acting device. The project office of the Air Force Weapons Laboratory (AFWL), however, had stated a specific need for a device with a range of frequency of operation up to 50 Hz. After some consideration of the matter Dames & Moore proposed that two separate actuator systems should be considered: (1) a lower-output, high frequency or dynamic actuator, and (2) a static (or quasi-static) device capable of driving the system at full torque. It was additionally proposed that since the static actuator was a relatively simple device it should only reach the design stage under this contract, and that the hardware development effort should be concentrated into the more challenging task, that of the dynamic actuator. This proposal was subsequently accepted by the United States Air Force and the design and development of the dynamic actuator as described in this report, duly commenced.

Design of the static unit commenced during the closing phases of the contract work. It is felt that sufficient design drawings and discussions are included to enable a working unit to be produced with little difficulty.

3. STRUCTURE OF THE CONTRACT REPORT

The concentration on hardware design and development of this second phase of the BSD project has been mentioned. In producing this report the main aim has been to set out as clearly as possible the detailed workings of the equipment which has been designed and developed under the terms of the subject contract. It is thus hoped that this document will serve as a useful manual for the equipment and will provide a basis and direction for future design and development work on the BSD apparatus. To avoid excessive bulk in the text with the resultant difficulties this often causes in finding essential information, introductions or discussions of the fundamental concepts related to the equipment are brief.

The text is divided into two main sections corresponding respectively to the design and development work on the BSD probe and the actuator systems. In each section the equipment is described in a brief introduction, using appropriate figures to identify the various subsections and to define some terms. Following this the equipment is broken down and dealt with part by part in the subsections of the report. Components are distinguished as being either specially designed and manufactured (identified by a component number) or as brought-in. Detailed or engineering drawings are available for specially designed components.

A selection of photographs and drawings of the equipment appear where appropriate in the report text. More detailed assembly and component drawings and additional photographs are available separately on request from AFWL/NTES.

II. THE BSD PROBE

1. GENERAL DESCRIPTION

A schematic of the borehole shear device probe and of the remainder of the in situ part of the measurement system is given in Figure 1. Functionally, the probe can be divided into three basic parts, the upper and lower sections and the central coupling cylinder section. Emplacement and pull-out loads, and the torsion applied during testing, are coupled downhole to the apparatus by a string of thin walled torque tubes. These loads are sensed in the upper section of the probe by the torque and normal load transducer assembly. This is a strain gauge which measures the applied loads by sensing elastic deformation of the outer wall of the section. Housed within the upper section are the hydraulic and services manifold assemblies. The services manifold is a sealed chamber into which all supply lines and electrical cables from various parts of the probe converge. The manifold provides space for termination of these services and for the installation of an electronic telemetry system, and houses a system of demountable connectors which allow for the probe to be detached from the uphole lines. Upon leaving the services manifold, the uphole lines are contained within a protective encasement known as the service conduit.

Leading through the upper section and extending down through the probe is the auger tube. In the upper section, the tube is used to mount the base plate of the services manifold, and to form the inner wall of the compartment. Below the services manifold is the hydraulic manifold assembly. Here, the hydraulic pipes feeding the actuators in the coupling cylinder are commoned to reduce the number of feed lines to the system to two; one

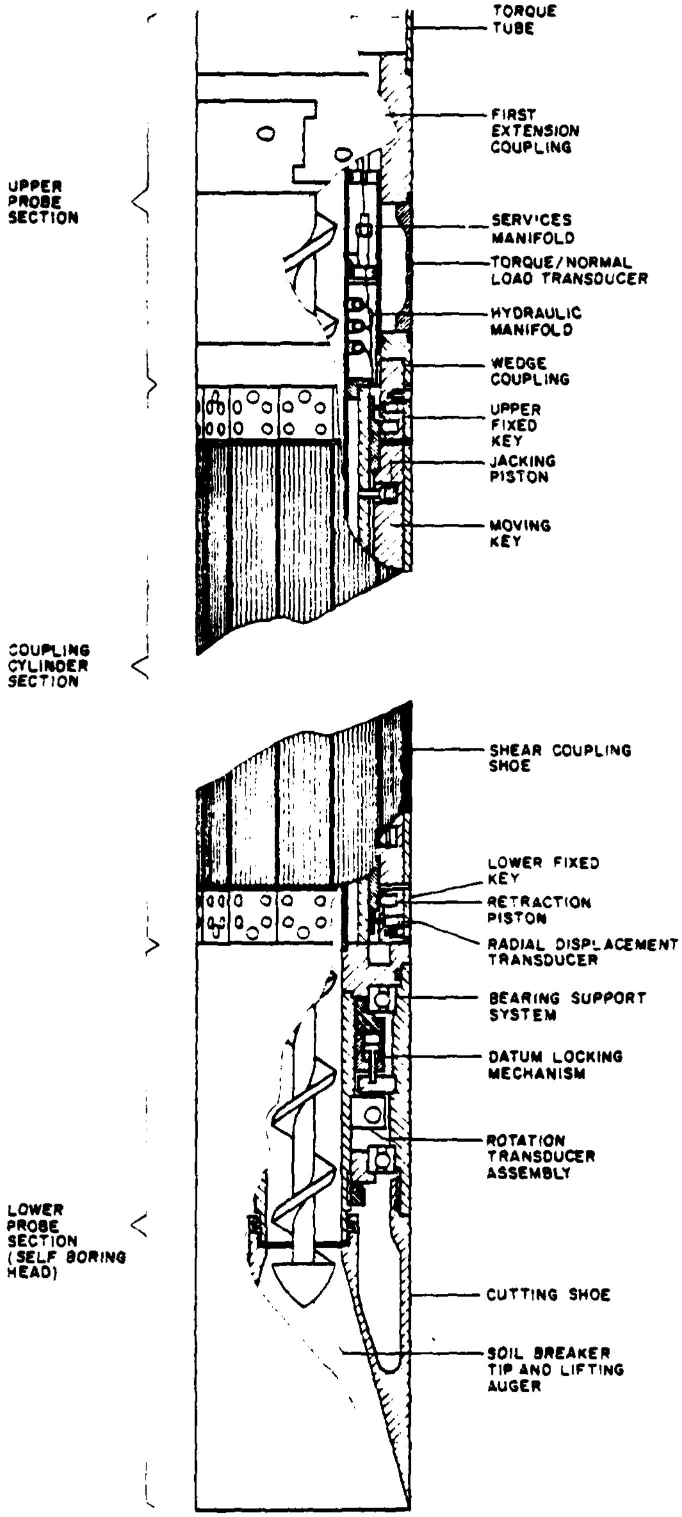


Figure 1. Schematic of the BSD probe.

for radial expansion of the coupling cylinder and one for retraction. The upper section of the probe (Fig. 2) is attached to the coupling cylinder (Fig. 3) by a special wedge coupling. Acting on an arrangement of tapered section keys protruding from the core of the coupling cylinder, the wedge system uniformly distributes the torsional loads applied to the cylinder during testing. Torsional load applied to the cylinder is coupled into the soil by an arrangement of moving keys. These keys are set into grooves in the core of the coupling cylinder and are fitted with shoes to provide a circular cylindrical profile. A system of jacking pistons along the keys allows the cylinder to maintain a constant contact pressure with the surrounding soil.

In order to emplace the apparatus with a minimum of disturbance to the surrounding soils a self-boring head is incorporated in the design for the BSD probe. Forming the lower section of the probe, the head acts under a vertical push-down load to remould the displaced material so that the horizontal and vertical stresses in the soil surrounding the probe are not significantly disturbed. The head channels the displaced soil into a throat section where it is broken up, and the cuttings are lifted through the probe by a helical flight auger. Being in the undisturbed soils ahead of the coupling cylinder, the self-boring head is used as reference for measuring the cylinder in rotation. The head is arranged with a bearing support system to allow transmission of push-down/pull-out loads and to withstand bending, and still let the head to rotate freely with respect to the cylinder. During emplacement the head is prevented from rotating by a locking mechanism. This allows a sensitive transducer to measure the coupling cylinder rotation without risk of over-ranging due to spurious movement during emplacement. The rotation measurement system is housed within the lower section immediately below the rotation locking mechanism.

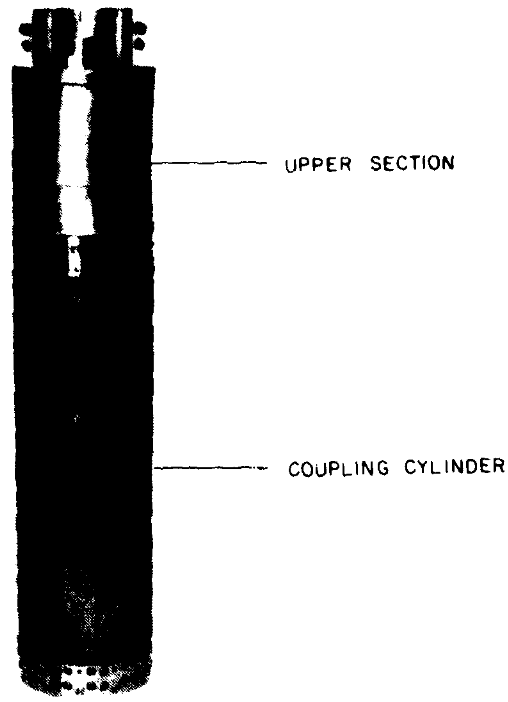


Figure 2. General view of the upper and coupling cylinder sections of the BSD probe.

Figure 3. The coupling cylinder section.

2. THE UPHOLE COUPLING SYSTEM

a. The Torque Tube The system of torque tubes used in the BSD apparatus perform a number of functions:

(1) They are used to apply torsional loads to the in situ apparatus during testing.

(2) They are required to withstand push-down loads during the self-boring emplacement phase and to pull-out loading loads during subsequent retrieval of the apparatus.

(3) Soil cuttings due to the self-boring method of emplacement are required to be either accommodated within the system or transported uphole. For testing in noncohesive soils where water flush cannot be used to transport the cuttings, it is proposed to raise them through the probe with a helical auger and to use the lower sections of the torque tube as a catchment tank.

The configuration thought to best satisfy the above requirements and, hence, proposed in the final design for the units is a large diameter thin walled tube. The tube is rated for strains approaching 10^{-3} in both simple shear and compression at the rated maximum loads ($\pm 10^4$ Nm torsion and 200 kN compression/tension), but it is not capable of withstanding these maximum loads applied simultaneously. By taking this nonconservative approach to the tube design, a relatively lightweight unit has resulted (the weight of a 2-m-long torque tube complete with extension couplings is 35 kg). The torque tube units are uniformly cylindrical in outside profile. There are a number of reasons for choosing this geometry:

(1) In order to facilitate self-boring emplacement, the section of tube coupled to the BSD probe must provide a continuous flush exterior profile as does the probe.

(2) The region of tube emerging from the casing of the pilot boring is particularly susceptible to jamming. The torque tube must be of plain profile here to prevent loose soil or other particles from locking the tube within the casing.

(3) The optimum torsional stiffness for the torque tube string for a given wall thickness is obtained by using the largest diameter tube. This minimizes the strain energy stored in the system due to the emplacement and testing loads.

(4) The largest attainable volume for storing soil cuttings is required to minimize the lift required of the helical auger.

(5) Manufacturing and material costs are minimized.

(6) Buckling will occur under the emplacement push-down load for certain lengths of torque tube string. The larger the diameter of the torque tube, the more effective becomes the pilot borehole casing in preventing damaging deflections of the tube.

(7) A plain cylindrical profile for the torque tube string allows attachment of the dynamic actuator (see Section III-2) to any section of tubing and at any desired height above ground.

As shown in Figure 4, the sections of torque tube are terminated at each end with identical mating couplings (A2, Fig. 5). Known as the torque tube extension couplings, the units are split in a stepped manner and are designed to transmit torsion via a compression face arrangement where the plane of the faces is normal to the axis of the tubes. Push-down loads are accommodated by end faces on the couplings and an overhanging location provides

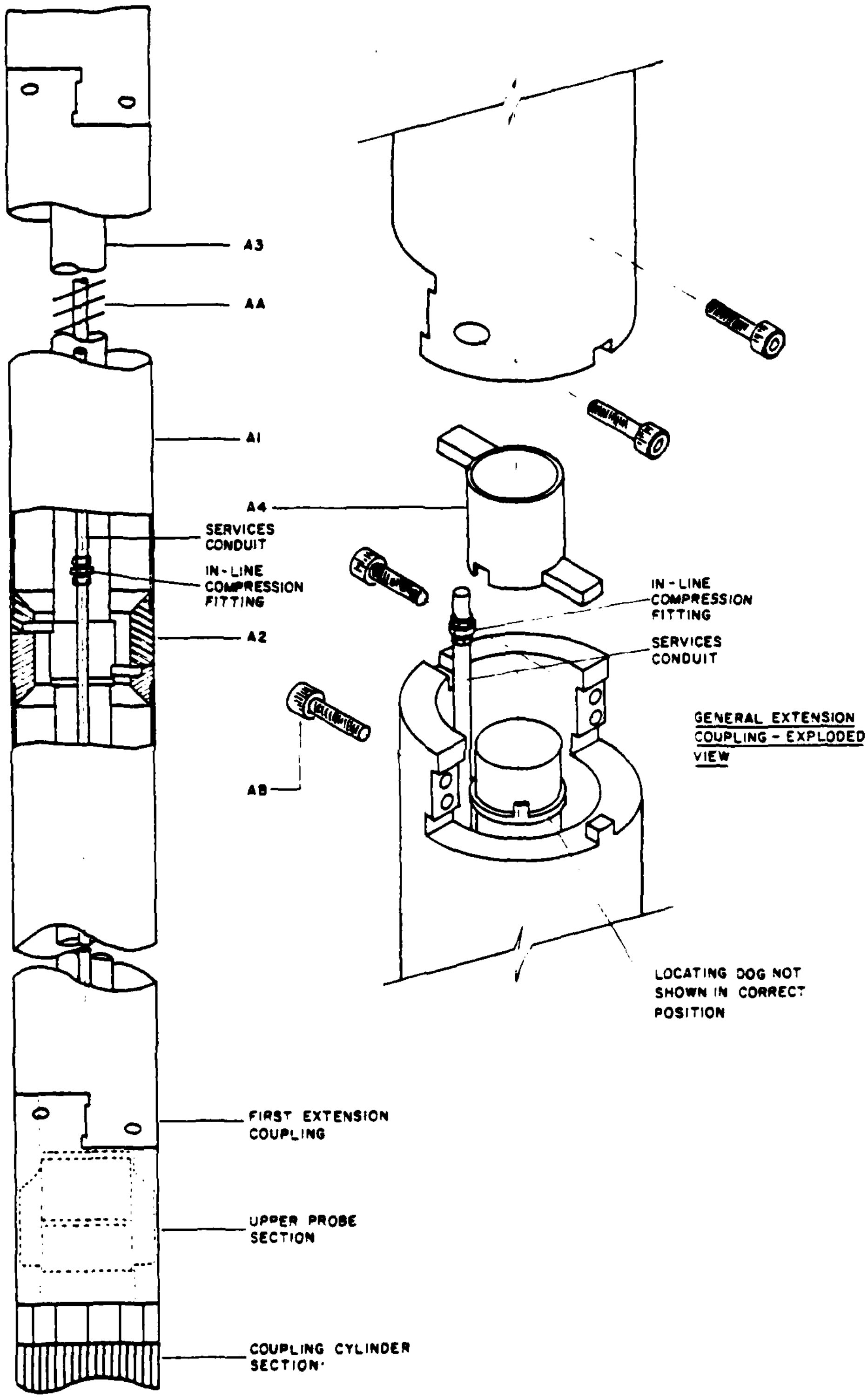


Figure 4. Assembly of the uphole coupling system.

KEY TO FIGURE 4

ITEM	QUANTITY	DESCRIPTION
A1	-	Torque Tube.
A2	2 x qty A1	Extension Coupling.
A3	1 x qty A1	Auger Extension Tube.
A4	1 x qty A1	Auger Extension Tube Socket.
AA	-	63.5mm (2 1/2 in) "Hex-Cor" auger and extension accessories. Supplier - Atlas Copco Craelius Limited, Long March, Daventry NN11 4DX, UK.
AB	2 x qty A1	Extension Coupling Fastening - M14 x 50 socket cap screw.

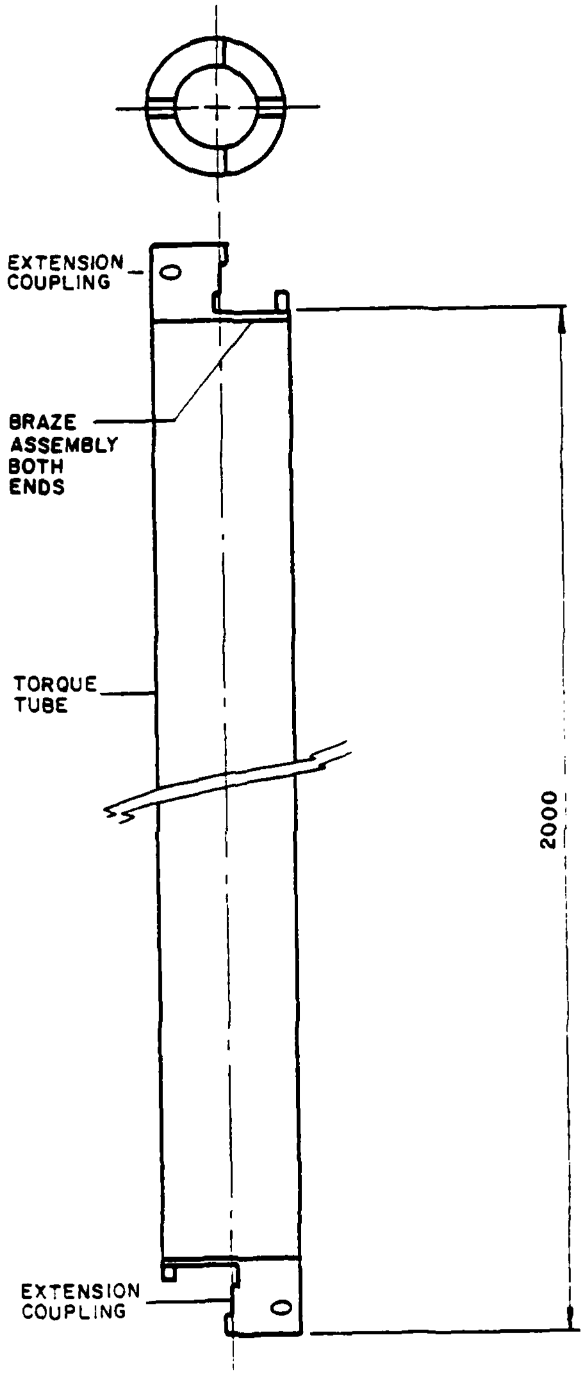


Figure 5. Assembly of the torque tubes.

the means for transmitting tension or pull-out loading. The couplings are secured by four retaining bolts (AB, Fig. 4) which serve to provide compressional preloading on the lateral faces of the units. Provided that the torsion applied to the torque tube string does not cause this preload to be relieved, the retaining bolt tension remains relatively constant and a torsionally stiff, low damping coupling results.

In keeping with the design requirements discussed above for the torque tube units, it was decided that brazed joints would be suitable for attaching the extension couplings to the central tube section. The couplings are accordingly arranged with a rear locating spigot which fits into a machined recess in the tube. Torsional loading of the assembly gives rise to shear around these joints and the area of spigot in the final design is such as to ensure low operational stress. Brazing was chosen because the process does not introduce problems with brittle behaviour. The relatively low temperatures involved also assist in minimizing the severity of heat induced distortion of the components. The central section of the torque tubes (A1 in Figs. 4 and 5) supplied as part of the hardware consignment were manufactured from hot finish seamless steel tubing, and the stock size of the chosen tubing was such that only the outside diameter was required to be machined. No particular difficulties were experienced in the manufacturing operations and it is felt that the final design for the torque tubes is a practical one. The units are robust and simple, and are relatively lightweight and easy to handle.

b. The Auger Tube A helical flight auger is proposed for lifting soil cuttings through the BSD probe during self-boring emplacement. Extending concentrically through the catchment tank section of the torque tubes and the probe to the throat of the self-boring head, the auger is encased throughout its entire lifting length. The encasement, or auger tube, is required to confine the soils around the auger, and to resist the torsional and vertical loads which arise from frictional coupling with the material in transit. Stainless steel seamless tubing was chosen for this application, and the dimensions of the selected tubing were based around a commercially available flight auger system. As with the torque tubes, a demountable system is required and it was decided that the auger tubes should be jointed at the same intervals, and at the same locations as the torsion tubes. In the final design, the auger tubes are butt jointed by the use of a special socket arrangement (A4, Fig. 4). Located within the torque tube extension couplings, the socket provides lateral, vertical, and torsional restraint for the auger tube. As shown in Figure 4 the socket nests within the torque tube coupling and is located and restrained by way of a dog arrangement configured on the end faces of the half couplings. The dogs on the couplings align with suitably positioned lugs on the auger tube socket, and thus when the extension couplings are made the socket is securely located. The unit is compact, allowing space for the services conduit to pass and for the soils falling into the catchtank sections of the torque tube to flow freely. Torsional and vertical restraint of the auger tube is provided by the socket via a sleeve welded onto the tube at the upper end. The sleeve prevents upward movement of the tube by butting on the lower face of the socket. A vertical dog engaging with a cut out in the wall of the socket prevents rotation of the tube. In the first extension coupling, ie., the coupling between the torque tube string and the upper section of the BSD probe, a special auger tube socket is used. For this socket the dog feature is not required, the auger tube in the upper section of the probe being securely attached to the core of the coupling cylinder. However, there is the need for space to accommodate the adaptor or lower end of the services conduit, and to achieve this, flats are machined on the sleeve section of the unit.

Within the BSD probe the auger tube comprises three basic units. In the upper section of the probe, a length of plain tubing is fitted with a machined flange which is bolted to the core of the coupling cylinder. As discussed in Section II-3b, this run of tube is used to mount the base plate of the services manifold and to provide the inner wall of the manifold compartment chamber. Socketed into a recess machined in the flange of the upper run of tube is the length of auger tube which passes through the coupling cylinder section of the probe. This run is socketed, in turn, at the lower end of the coupling cylinder into the last or bottom run of tube leading down to the throat of the self-boring head. In this section, however, a portion of the central tube is required to transmit the pull-out loads of the cutting head into the coupling cylinder core. The relatively thin walled tube used in the rest of the system is not suitable for this application. Therefore, a specially designed more sturdy machined part is used.

c. The Services Conduit The services conduit is the name given to the protective encasement for the electrical cables and hydraulic pipes which lead uphole from the probe. The conduit is required to provide a hermetically sealed channel or passage for the uphole lines and to protect them from abrasion, crushing or other forms of damage during set up or dismantling of the apparatus. By providing this degree of protection, the need in the system for connectors and lines with stringent environmental performance is eliminated. This yields some significant advantages.

(1) Standard electrical connectors and hydraulic fittings can be used. A wide choice is then to be had which reduces problems of availability or with delivery. Design changes or modifications are much easier to carry out if environmentally protected units are not required.

(2) A relatively large number of uphole electrical conductors are required by the fully instrumented system. This could lead to severe design problems with the corresponding bulk of the ruggedized and sealed connectors.

(3) Environmentally protected connectors are significantly more expensive than the types designed for less-exacting duty.

For the BSD apparatus, the design of the conduit system permits a choice for the type of protective tubing to be used. In the uphole run, the conduit is jointed by compression-type pipe fittings (Fig. 4). This type of fitting is useable with a wide range of different tube types, ranging from rigid metal to flexible plastic. The advantage of using of plastic tubing is that prethreading of the conduit through the torque tubes can be achieved without unmaking the conduit joints and hence exposing the uphole lines. However, the thicker wall and lack of stiffness compared to metal tubing limit the cross-sectional area of uphole line that can be accommodated within the conduit and the length of run through which it can easily be threaded. It is thought that the choice of tubing type is best determined by considering the individual requirements of a particular testing program. Given the number of electrical connections required, the problems of space within the conduit can be assessed.

As mentioned previously, the services conduit terminates at the probe end in the cover plate of the services manifold. This cover lies within the central bore of the first extension coupling on the upper section of the BSD probe. A special adaptor (Fig. 6) screws into the cover and extends upwards through the coupling to allow access for attaching the conduit. The adaptor is finished with a plane reduced diameter for attaching a simple in-line compression fitting. The fitting diameter is chosen to be compatible with the metric standard 16-mm or Imperial 5/8-in standard fittings.

3. *UPPER PROBE SECTION*

a. The Torque and Normal Load Transducer As shown in Figure 7, the torque and normal load transducer assembly comprises three basic components:

(1) The lower half of the first torque tube extension coupling.

(2) An intermediate thin walled gauge tube section.

(3) A special wedge coupling for attaching the assembly to the coupling cylinder unit.

The device is incorporated into the apparatus to measure the vertical push-down loads due to self-boring emplacement and the torsional loads applied during the subsequent testing. The transducer action is provided by strain gauges which sense axial compressional strain and diagonal elongation in the gauge tube. The gauges are bonded to the inside wall of the gauge tube, this region being a hermetically sealed part of the services manifold chamber. A full bridge configuration is recommended for the two transducer circuits which will result in an array of at least eight gauges on the tube. The gauge tube, manufactured from a steel alloy which is commonly used in transducer manufacture has a wall thickness of 2.5 mm. Four diagonally orientated metal foil strain gauges are connected in a bridge configuration to sense the torsional

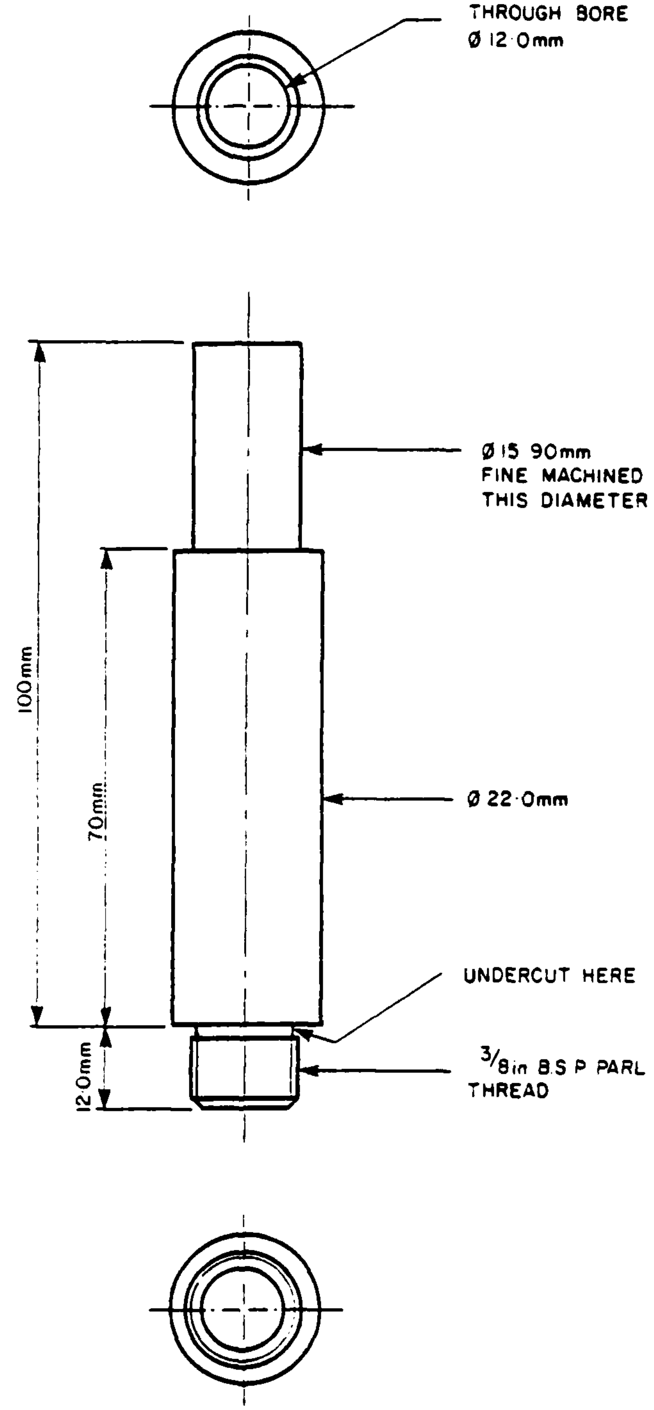


Figure 6. Services manifold - services conduit adaptor (component 55).

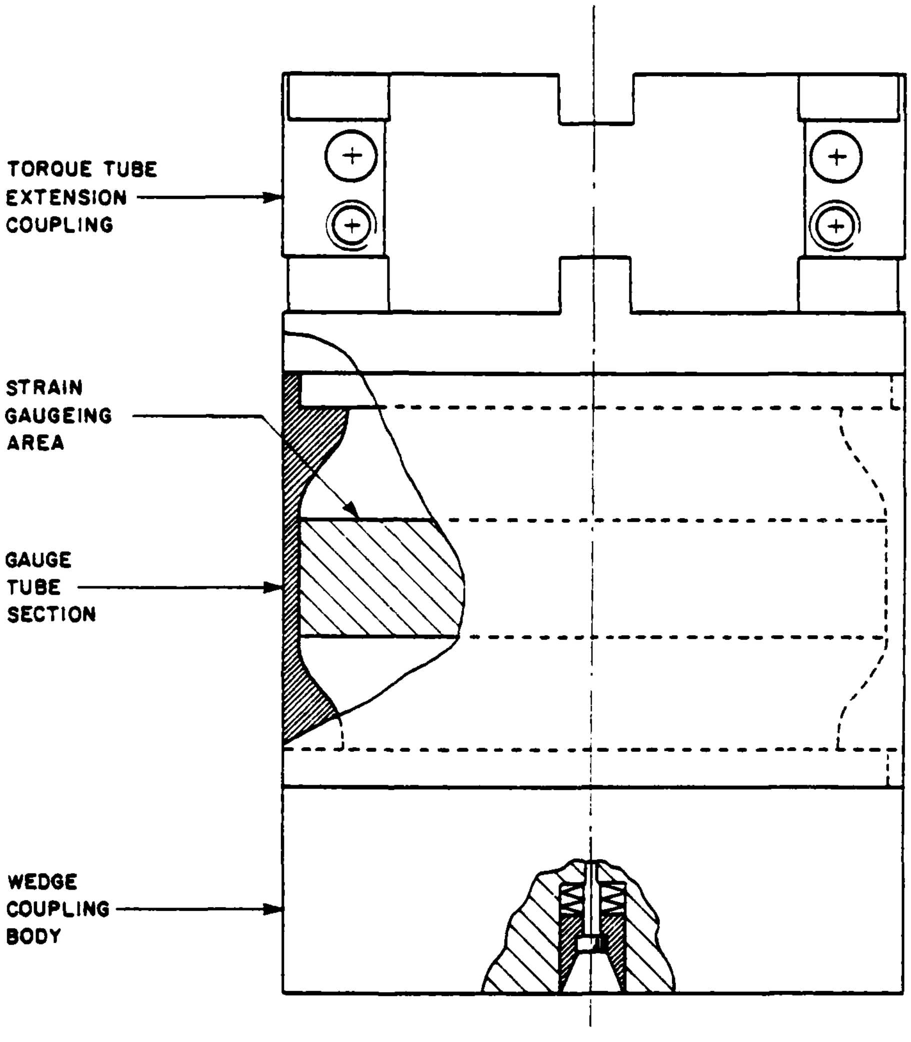
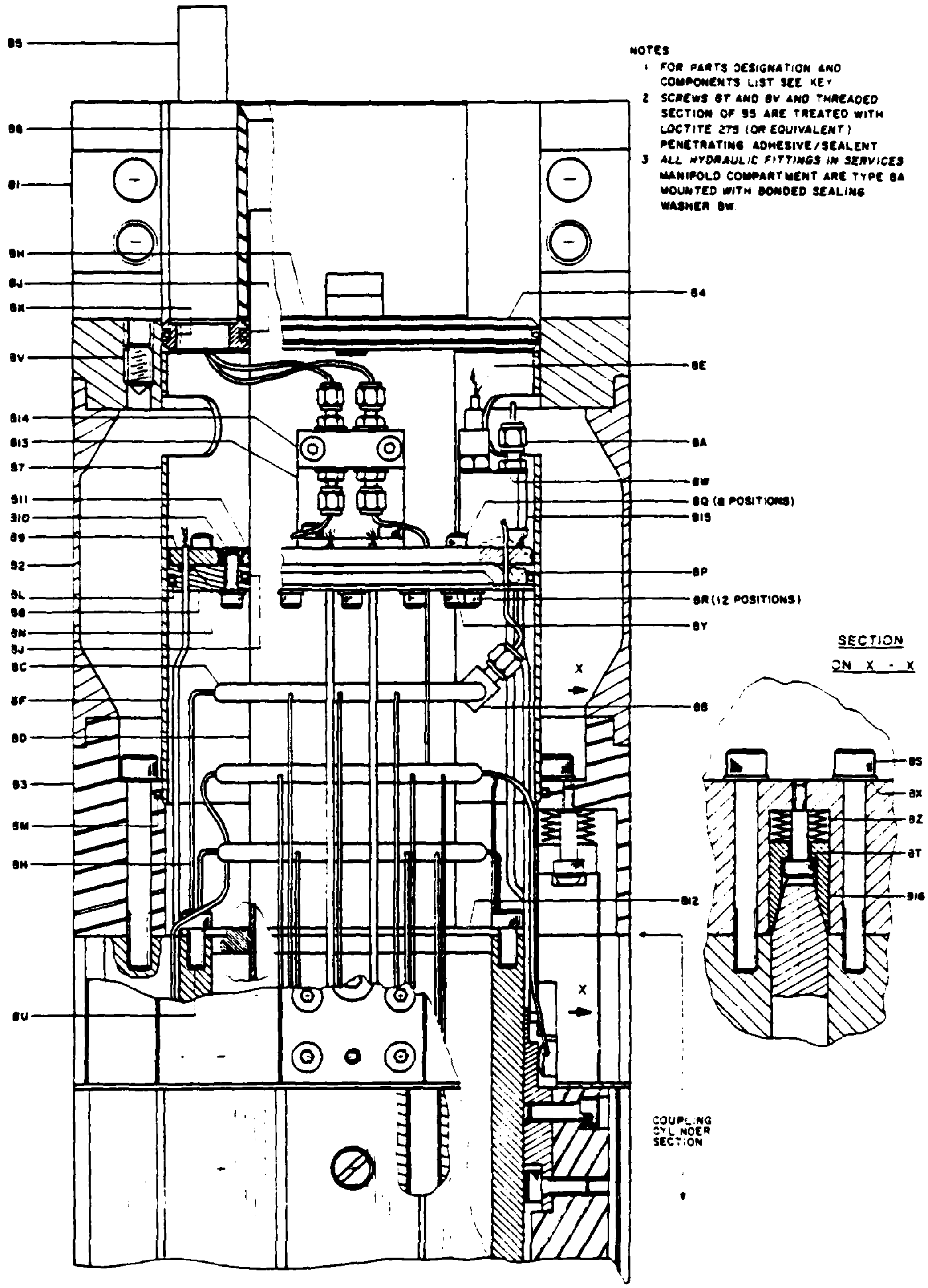


Figure 7. The torque and normal load transducer unit

deflection of the gauge tube. A calculated transducer constant of 2.03×10^{-4} mV/V/Nm results (millivolts output per volt of energisation per Newton-meter of applied torque). For normal loads the calculated constant is 9.01×10^{-6} mV/V/N. Appendix A shows the configuration of the gauges and their connecton in the bridge circuits. The calculations for the transducer contants are also presented.

In order to transmit torsional loads into the coupling cylinder section of the probe, a special wedge type coupling is configured in the lower part of the torque and normal load transducer unit. This coupling mates with a set of fixed keys which protrude from the upper end of the coupling cylinder core. These keys are tapered in section, and the design of the coupling is such that a system of spring driven wedges is used to engage with the keys. This feature is highlighted in the detailed section on X - X on Figure 8. The core of the coupling cylinder is a complex shape not readily lending itself to applying relatively large torsional loads. The arrangement of fixed keys and spring driven wedges is designed to overcome this problem and ensure a uniform distribution of load into the core. To retain the wedges in preload and to allow the system to transmit pull-out loads, the torque/normal load assembly containing the wedge system is bolted to the core of the coupling cylinder. Being accessed through key wrench holes in the top extension coupling, these bolts (BS, detailed section on X - X on Fig. 8) secure the machined body of the wedge coupling to the top face of the coupling cylinder core. In order to maintain sealing of the services manifold chamber, the through bolt fastenings must be watertight. This is achieved by using bonded sealing washers (BX, detailed section on X - X on Fig. 8) under the heads of the retaining bolts. The access holes in the extension coupling must likewise be plugged, and to this end the key wrench holes are threaded. Following



Assembly of the upper probe section.

KEY TO FIGURE 8

ITEM	QUANTITY	DESCRIPTION
B1	1	Torque/Normal Load Transducer - First Extension Coupling.
B2	1	Torque/Normal Load Transducer - Gauge Tube.
B3	1	Torque/Normal Load Transducer - Wedge Coupling Body.
B4	1	Services Manifold - Cover.
B5	1	Services Manifold - Services Conduit Adaptor.
B6	1	Services Manifold - Auger Tube Extenson Socket.
B7	1	Services Manifold - Sealing Sleeve.
B8	1	Services Manifold - Base Plate.
B9	1	Services Manifold - Countersink Seal Clamping Plate.
B10	1	Services Manifold - Cone Driving Plate.
B11	1	Services Manifold - Wedge Ring.
B12	1	Auger Tube Mounting Flange.
B13	1	Hydraulic Connector Block - Mounting Bracket.
B14	1	Hydraulic Connector Block.
B15	1	Pressure Transducer Mounting Block.
B16	24	Wedge Plates.
BA	-	Hydraulic Compression Fitting, in line coupling, 1.63 mm (1/16 in) type "Gyrolok" 1U. Supplier - Hoke Incorporated, One Tenakill Park, Cresskill, N.J. 07626, USA.
BB	3	Hydraulic Compression Fitting, all tube T (hydraulic manifold oilways) "Gyrolok" ITTT.
BD	-	Specification 316 Seamless Stainless Steel Tubing - 66.68 mm (2 5/8 in) OD x 1.63 mm (1/16 in) wall Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.

KEY TO FIGURE 8 (continued)

ITEM	QUANTITY	DESCRIPTION
BE	1	Pressure Transducer (lateral jacking pressure), 0-13.8 MPa (0-2000 lb/in^2), 10-32 UNF mounting, model XTM-1-190-2000. Supplier Kulite Semiconductor Products Incorporated, 1039 Hoyt Avenue, Ridgefield, N.J. 07657, USA.
BF	-	Specification 316 Seamless Stainless Steel Tubing, 2.38 mm (3/32 in) OD x 0.2 mm (0.008 in) wall. Supplier - Tubesales (UK) Limited (see BD).
BH	-	Specification 316 Seamless Stainless Steel Tubing, 1.63 mm (1/16 in) OD x 0.4 mm (0.016 in) wall. Supplier - Tubesales (UK) Limited (see BD).
BJ	1	O Ring (services manifold base plate - inner ring), 66 ID x 3 section, ref 206-366.
BK	1	O Ring (services manifold cover - outer ring), 116 ID x 3 section, ref 206-416.
BL	1	O Ring (services manifold base plate - outer ring), 112 ID x 3 section, ref 206-412.
BM	1	O Ring (wedge coupling body - sealing sleeve), 122 ID x 2 section, ref 206-222.
BN	24	O Ring (services manifold countersink seal - transducer cabble sheath), 1.78 ID x 1.78 section, ref 004.
BP	4	O Ring (services manifold countersink seal - hydraulic pipe lead through), 1.42 ID x 1.53 section, ref 003.
BQ	8	Socket Cap Screw (retaining B9), M4 x 10 HTS.
BR	12	Socket Cap Screw (retaining B10), M4 x 14 SS.
BS	12	Socket Cap Screw (torque/normal load transducer retaining), M8 x 60 HTS.
BT	12	Socket Shoulder Screw (wedge plate retaining), 6 (M5) x 16 SS.
BU	6	Socket Cap Screw (auger tube flange retaining), M6 x 12 SS.

KEY TO FIGURE 8 (continued)

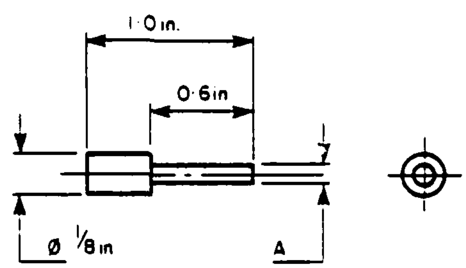
ITEM	QUANTITY	DESCRIPTION
BW	4	Bonded Sealing Washer (bulkhead sealing hydraulic fittings BA), 2BA (equivalent 10-32 UNF), Dowty ref 400-003-4490-02.
BX	12	Bonded Sealing Washer (sealing under heads of BS), 8 mm, ref 400-007-4490-02.
BY	12	Bonded Sealing Washer (sealing under heads of BR), 4BA (equivalent 4 mm), ref 400-002-4490-02.
BZ	8 x 12	Disc Springs (wedge coupling), 18 OD x 6.2 ID x 0.9 thick.

completion of assembly of the upper section of the probe, these holes are closed off using grub screws and hydraulic thread sealant (Loctite 542 or equivalent). To ensure alignment of the access holes in the extension coupling with the retaining bolt holes in the wedge coupling body, the units are pin located on the gauge tube section. The machined recesses which form the brazed joint faces are drilled radially and pinned, and a corresponding pair of locating slots are machined in the gauge section. The final stage in manufacturing (prior to applying the Cadmium plated finish) is the brazing operation. This is carried out under an inert gas atmosphere in a tempertature controlled furnace. By this process the risk of heat induced distortion in the final component is minimized, and thermal stresses are relieved by the annealing action of the operation.

b. The Services Manifold The services manifold is essentially the point in the BSD probe where the demountable couplings and connectors for the uphole cables and hydraulic lines reside. As discussed is Section II-2c, the design of the system is such that complete mechanical protection and hermetically sealed encasement is provided for the uphole lines. This is achieved by a sealed compartment formed within the upper section of the probe. Figure 8 shows how this chamber is formed. The annular cover of the manifold (B4, Fig. 8) is piston sealed on the inner and outer rims against the auger tube (BD, Fig. 8) and the machined inner bore of the extension coupling (B1, Fig. 8). An inner sealing sleeve (B7, Fig. 8) provides a seating for the cover, and it extends down to seal off the wet region of the upper probe section surrounding the hydraulic manifold system. The O ring piston seals in the outer rim of the services manifold base plate (B8, Fig. 8) and the upper recess of the wedge coupling body (B3, Fig. 8) are the means for achieving this. Thus the region outside the sealing sleeve bounded by the gauge tube section (B2, Fig. 8) and the annular space in the services manifold chamber is the dry region. The openings in the upper section of the sealing sleeve are incorporated to allow the lead wires from the torque and normal load transducer to enter the services manifold compartment. In assembling the upper section of th system, these transducer lead wires must be terminated after the sealing sleeve is inserted. Relatively large cut-outs are provided in the sleeve to assist in leading the wires through it.

The base plate (B8, Fig. 8) of the services manifold provides the mechanical support or mounting for the uphole line terminations. The plate is secured to the central auger tube by a special compression type coupling. This coupling, comprised of the compression ring (B11, Fig. 8) the cone plate (B10, Fig. 8) and the driving screws (BR, Fig. 8) is a compact and rigid fixing which allows free adjustment of the position of the base plate. Transducer cables and hydraulic supply pipes enter the services manifold through the plate and are sealed by countersunk compression seals which are machined into it. The annular compression plate (B9, Fig. 8) located outside the cone plate (B10, Fig. 8) squeezes the O rings (BN and BP, Fig. 8) onto the pipes to form a watertight seal. For their passage through the wet or unsealed regions of the probe, the transducer cables are sheathed within thin-walled stainless steel tubing. This method, being essentially a scaled down version of the services conduit approach described earlier, is a convenient method of unifying the problems of routing and sealing lines within the probe. Regardless of whether they contain a hydraulic medium or electrical conductors, outwardly they are physically identical and the methods used to affect bulkhead sealing are the same. As a consequence of this approach to sealing the electrical systems in the probe, the sheathed cables entering the services manifold compartment do not fan out. Each encased cable assembly leads to only one transducer, and the sheaths or protective tubes are hermetically sealed into the individual housings in which the transducers are mounted. This results in a relatively large number of encased cables leading into the manifold chamber. To minimize difficulties in assembly, the countersink sealing system is configured at regular geometric intervals around the base plate. These intervals coincide with the grooves in the coupling cylinder core from which the cables emerge and, hence, the tubing runs past the hydraulic manifold assembly are virtually straight. Where a particular instrumentation system is not installed into the probe, the corresponding entry port in the baseplate of the services manifold must be blanked off. Figure 9 gives the design of suitable sealing pins for both the hydraulic pipe and the sheathed cable entry seals. Before using the probe in saturated soils or in any other situations where there is risk of water entering the unit, it is advisable to check that all unused ports in the manifold baseplate are plugged.

Two hydraulic mounting fixtures appear inside the services manifold; one for terminating the uphole pipes, the other for mounting and feeding a pressure transducer. The uphole pressure lines (one for rotation locking and coupling cylinder retraction, the other for radial expansion of the cylinder), terminate in the connector mounting block (B14, Fig. 8) This block, secured to the base plate by means of the bracket (B13, Fig. 8), is a simple through-port block which provides a secure anchor for the pressure lines from the hydraulic manifold. Block B15 carries the transducer (BE, Fig. 8) which measures the radial jacking pressures. As shown in the plan view, Figure 10, two hydraulic fittings and transducer ports are included in the design to accommodate another pressure measurement, if required. Feed lines to the transducer port block are accommodated in the arrangement of countersink seals in the services manifold baseplate.



BLANKING PLUG TYPE	DIMENSION A	NUMBER REQUIRED
1	Ø 1/16 in	6 OFF
2	Ø 3/32 in	24 OFF

Figure 9. Services manifold - base plate blanking pins.

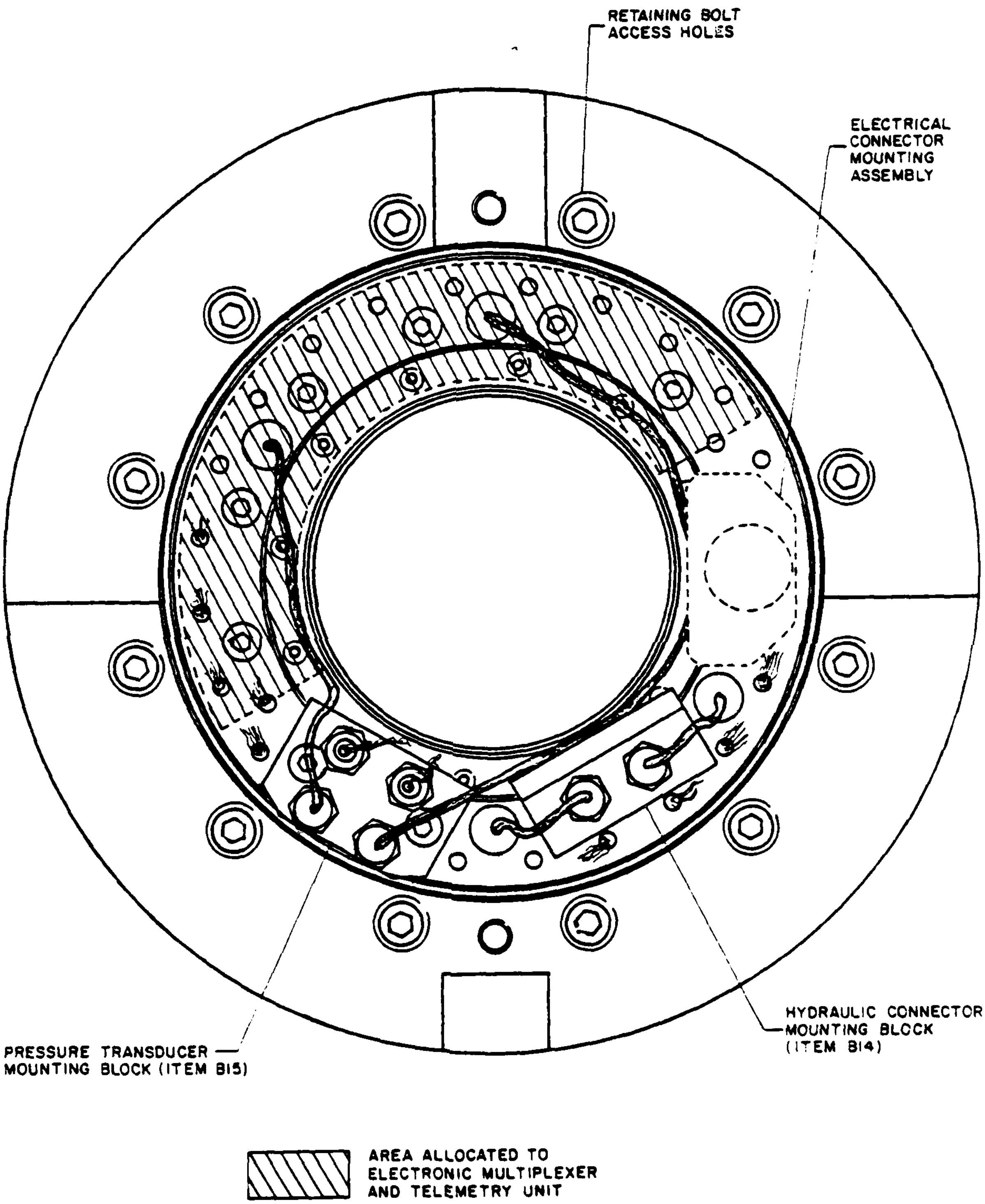


Figure 10. Plan view of services manifold.

In order to reduce the number of electrical conductors in the uphole line, some degree of commoning or multiplexing of the transducer leadwires must be carried out in the services manifold compartment. Apart from the problems of handling bulky cables and of threading them through the services conduit, there is the problem of accommodating the connectors. It can be seen from the plan view of the manifold chamber of Figure 10 that the compartment is relatively constrained for space, particularly in the radial direction. A market review of connectors revealed that, with the constrained width of the manifold compartment, few devices could offer the contact density required by the fully instrumented probe. The problem is discussed in more detail in the section on instrumentation, but, in summary, it is concluded that the optimum arrangement incorporates an electronic multiplexer system into the manifold. Using only a small number of signal conductors, this system connects the outputs from the transducers within the probe sequentially to the uphole leadwires in response to an external command signal. This technique shows that the fully instrumented probe requires 25 uphole connections. For the system described in this report the conductors are distributed between two connectors which, for this number of ways, are relatively small and hence easy to accommodate within the manifold compartment. Figure 10 shows the position on the services manifold baseplate proposed for the electrical connectors. Mounted on a platform positioned between the hydraulic port blocks and one end of the electronic multiplexer unit, the connectors are allowed relatively good access.

Recommendations for particular types of connectors and for the associated uphole cables are not given in this report. Since these will depend on the amount of instrumentation to be installed into the probe and whether or not the multiplexer system is to be used, it was felt that the task of specifying these components was best left to the stage where the precise requirements of a particular testing program are known.

The electronic multiplexer (omitted for clarity in the sectional view of Figure 8) is shown in Figure 10 located above the countersink seal arrangements in the baseplate and occupying half of the annular space in the manifold compartment. It is proposed that this unit be configured on a semicircular shaped printed circuit board mounted on pillars which are attached to the compression plate (B9, Fig. 8). Further detail of the mechanical aspects of this assembly is not given in this report, it being felt that such matters would be better left to the specialist manufacturers. It may be decided by a particular manufacturing or prototyping facility that the unit should be constructed with high packing density using double sided printed circuit boards. On the other hand, an alternative and equally valid approach is to use a less dense component packing, and stack a number of boards vertically. Whatever the choice, an important point to consider is the connection of the transducer lead wires to the system. Despite the fact that the design of the manifold system allows complete access around the base plate by removal of the torque/normal load transducer unit and the sealing sleeve, there are still a large number of connections to be made. These must be arranged in an orderly fashion to allow functional checking of the system and to minimize the risk of damage during assembly or dismantling of the upper probe section.

c. The Hydraulic Manifold The hydraulic manifold functions as an intermediate system between the lower sections of the probe and the services manifold. It serves to reduce the relatively large number of hydraulic lines feeding the probe to two basic supply lines:

(1) The radial expansion or lateral jacking pressure supply.

(2) Coupling cylinder retraction and rotation locking pressure.

Radial expansion of the coupling cylinder is achieved by a system of pistons and cylinders which jack an arrangement of moving keys out against the core of the unit. Five pistons per key are employed to distribute the jacking load, and the pistons are coupled together within each keyway groove by means of microbore hydraulic pipe. In order to avoid problems with excessive flow impedance, series linking of the piston units is not continued for each keyway. An individual supply pipe is brought out for the jacks in each keyway and, thus, a manifold with 12 outlet ports is required to supply the system. A still larger number of supply lines are needed to retract the moving keys. Being accomplished by a set of radially inward acting pistons operating on heel extensions at each end of the keys, the system uses a total of 24 piston units. No convenient means for commoning the supply lines within the probe was found and hence the full complement of pistons are also required to be fed directly from the hydraulic manifold.

Figures 8, 11, and 12 show that the manifold comprises a series of tubular ring oilways. The rings are manufactured from small bore steel tubing which is bent into shape and closed by silver soldering the ends into modified T compression fittings. Silver soldering is also used for fitting the supply lines, these entering the oilways via a series of radially drilled holes. These outlets are arranged at regular intervals around the rings to coincide with the routes for the pipes through the coupling cylinder. For simplicity of manufacture a three ring system was chosen, each ring being physically identical and being arranged to provide 12 outlet ports to feed the probe. Additional pipes are attached to common two of the oilways for the

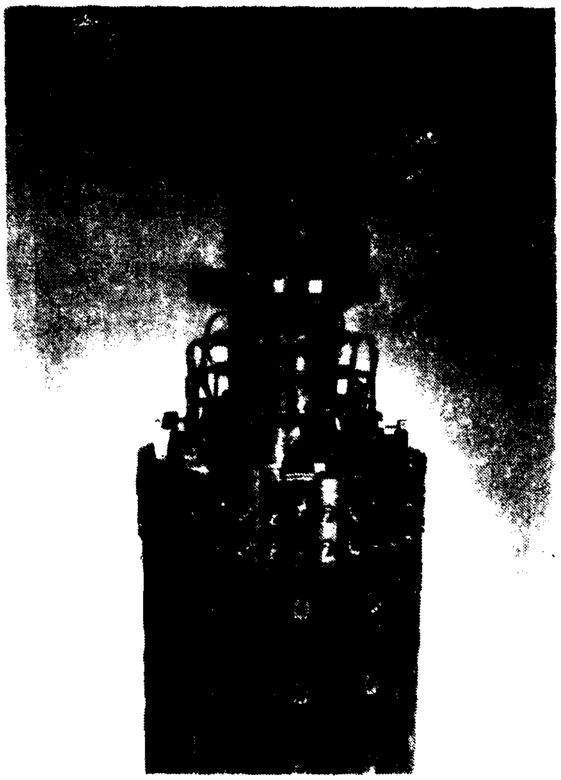


Figure 11. General view of the hydraulic and services manifolds.

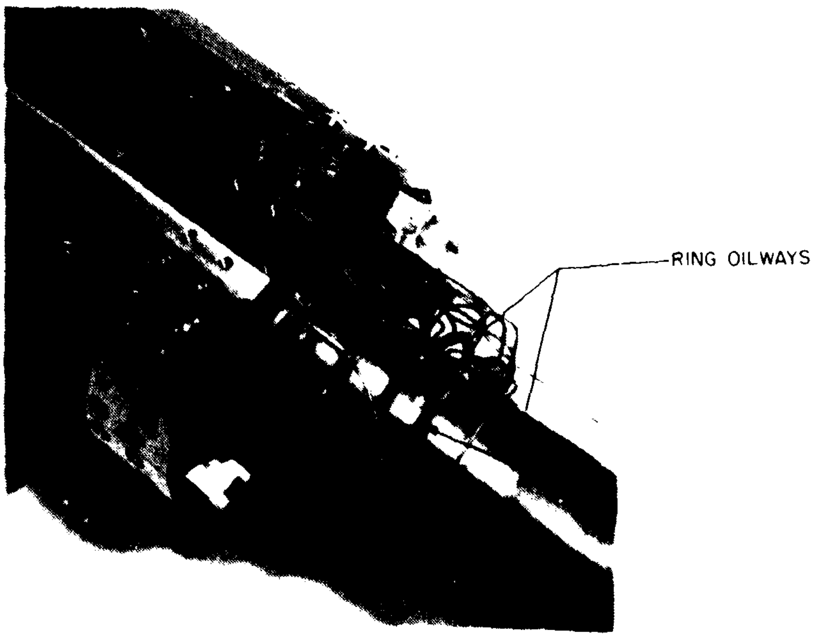


Figure 12. Routing of the services into the coupling cylinder.

retraction function, and to provide final feed and pressure monitoring lines to the services manifold as described in the previous section. The tubing chosen for the oilways in the hydraulic manifold is specially designed for environmental applications in hydraulic systems. Known as bundy tubing it is manufactured by a laminating process which gives it a high tolerance to bending. The material also features a special protection against corrosion which is provided by a tinned exterior finish. Silver soldering was adopted as the method for jointing the piping in the system largely because of the lack of space available for conventional compression fittings. Lower temperature soldered joints (lead-tin solder and certain proprietary brands of special solder repair alloys) were experimented with, but they were not found to possess sufficient strength to resist bending and torsional loads from the feed pipes. However, with a correctly made silver soldered joint it was found that the fatigue strength of the piping was the limiting factor. The chief disadvantage to the process was found to be that it required more skill than low temperature soldering, there being the need to use a flame; but with the low thermal mass of the pipes and the solderability of the materials involved, it was soon found that acceptable joints were made with relative ease.

4. COUPLING CYLINDER SECTION

a. The Radial Expansion System Radial expansion is incorporated into the design of the BSD probe to ensure that the soils surrounding the coupling cylinder section are maintained under a constant or controlled lateral stress. The surface of the coupling cylinder is discretized into 12 segments, each of which (due to the action of a system of keys set into the core of the cylinder) is free to move radially. Torsional or shear stress can be applied to the keys and, hence, to the surrounding soil, while lateral pressure is maintained. Jacking forces are derived from a series of pistons and cylinders acting between the core of the coupling cylinder and the moving keys. The cylinder bores of the jacking units are machined directly into the keys, the system comprising of 5 pistons per key distributed at regular intervals. The pistons are mounted in the core of the coupling cylinder protruding from the bottom of the keyway grooves in the unit. Figure 13 is a section through the coupling cylinder showing one of the moving keys and the arrangement of the retraction system in the upper fixed key assemblies.

During self-boring emplacement and pull-out of the BSD probe, loading in the direction of the axis of the system will be applied to the shear coupling shoes. Being liable to cause high stressing and possible damage to the lateral jacking system if otherwise unrestrained, this load must be diverted into the coupling cylinder core. This is achieved by thrust washers which restrain the ends of the keys. The washers are specially shaped to nest into the keyway grooves in the core and to finish flush on the outside with the cylindrical profile of the probe. Load is transferred into the core of the coupling cylinder by the butting action of upper and lower fixed key assemblies. Comprising a cover plate and fixed key, the assemblies are bolted to the coupling cylinder core and, hence, provide restraint for the moving keys. In order for this restraint or load bypassing system to be effective, however, allowance must be made for component manufacturing tolerances in the system and for the axial working clearances required by the moving keys. If the radial jacking system cannot allow for this movement it will incur loading. As shown in Figure 13 this problem is avoided by incorporating a special floating head into the lateral jacking piston units.

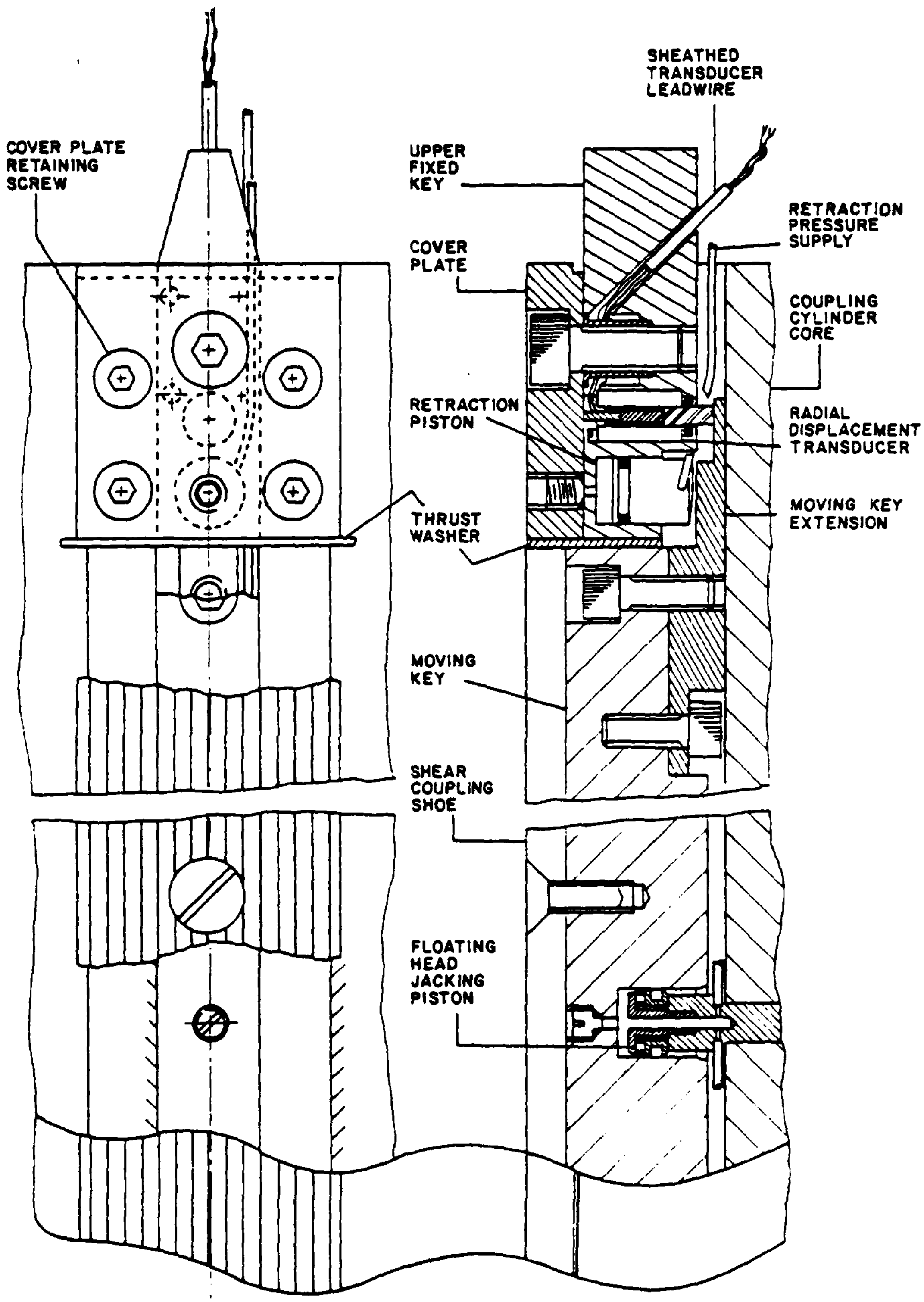


Figure 13. The radial expansion retraction system.

A detailed assembly drawing of the arrangement is given in Figure 14 where it can be seen that the head of the piston is allowed to move radially due to the action of an undersized shaft section of the central retaining peg C15. The moving part or piston head C16 is faced sealed to the underside of the peg and piston sealed in the usual manner on the outside edge. By this arrangement, the working clearance of the sealing ring system (a crucial factor in determining the maximum operating pressure) is unaffected by lateral movement of the piston head. Because of space restrictions in the keyway grooves, a number of features are incorporated into the floating head piston unit to yield a compact design. Examples of these are silver soldered joints for the hydraulic supply pipes and interference fits used to locate the units in the core of the coupling cylinder.

b. The Retraction System Previous sections here mentioned the upper and lower fixed keys. These units, as their name implies, are not free to move but are bolted via a cover plate to the core of the coupling cylinder. They perform a number of important functions:

(1) They provide radial reaction for the forces required to retract the moving keys.

(2) They provide axial restraint for the moving keys against push-down and pull-out loads.

(3) They provide a housing for the transducers that measure radial displacement of the moving keys.

(4) The upper set of fixed keys transmit the torsional loads applied during testing into the coupling cylinder core.

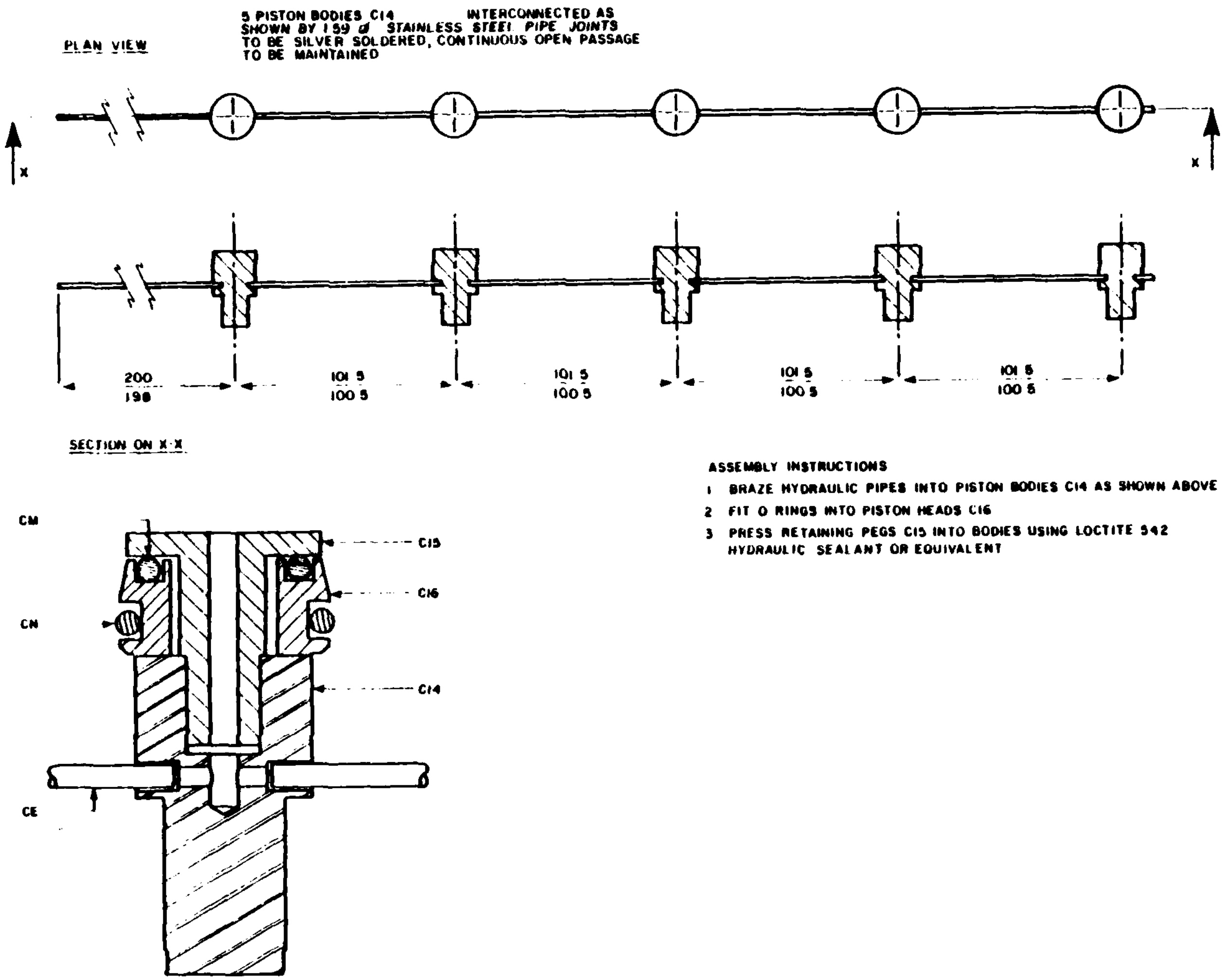


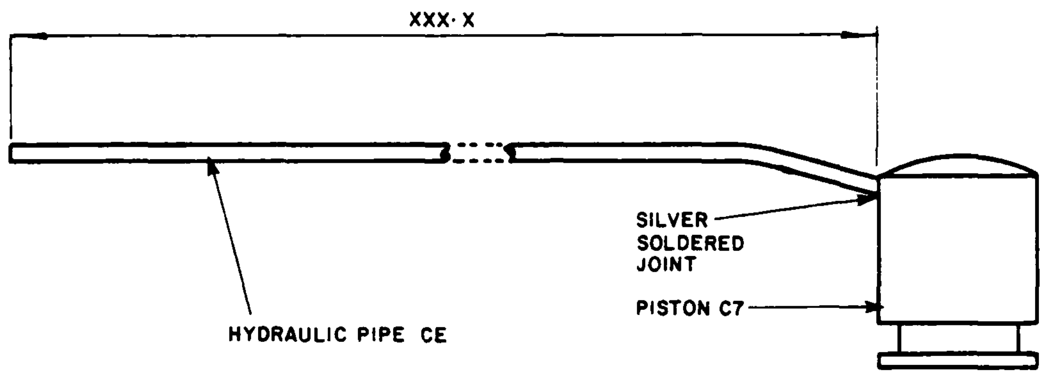
Figure 14. Assembly of the expansion jacking pistons.

KEY TO FIGURE 14

ITEM	QUANTITY	DESCRIPTION
C14	60	Jacking Piston Body.
C15	60	Head Retaining Peg.
C16	60	Jacking Piston Head.
CE	-	Specification Seamless Stainless Steel Tubing, 1.63 mm (1/16 in) OD x 0.4 mm (0.016 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
CM	60	O Ring (floating piston head top seal), 7 ID x 1.5 section, ref 206-007.
CN	60	O Ring (floating piston head edge seal), 9 ID x 1.5 section, ref 206-009.

From a structural viewpoint, the fixed keys and their associated cover plates can be regarded as a single unit. As shown in Figure 13, they are bolted and pinned together and the resultant assembly is secured to the outer flats on the coupling cylinder core by four retaining bolts. These retaining bolts are recessed into the curved outer surfaces of the cover plates and are loaded in shear to restrain the moving keys axially, and in tension when retracting them. The exterior surfaces of the cover plates are machined to provide a plain cylindrical profile, and the ends facing axially inwards are finished flat and flush with the ends of the fixed key. These surfaces, as shown in Figure 13, are similar to the end faces of the moving keys, both being arranged to butt against the intervening thrust washer.

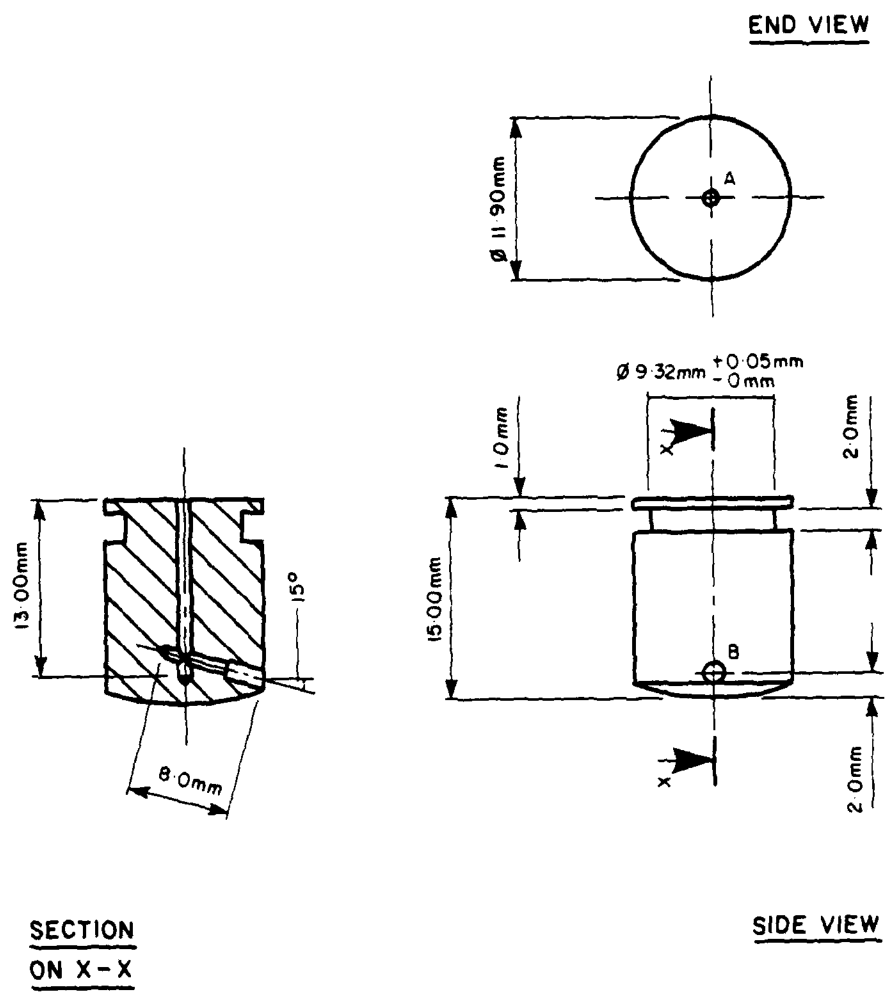
The retraction pistons act radially inward on heel extensions attached to the moving keys. The cylinder bores are machined directly into the bodies of the fixed keys, and cone pointed screws threaded into the cover plate are used as bleed valves for the system. Hydraulic pressure is applied via microbore piping which is silver soldered into the sides of the protruding ends of the pistons, and the supply reaches the rear face of the pistons by a drilled oilway (Figs.15 and 16). This arrangement allows convenient routing of the retraction pressure lines in the bottom of the keyway grooves, and avoids complication in the design of the fixed key bodies by eliminating the need to route hydraulic supplies to the cylinder from within the key. To secure the cover plate and fixed key components together, a pinned and bolted joint is used. Here positive location and shear stiffness is provided by four spring pins arranged around a central retaining bolt. The bolt head is recessed into the cover plate as are the heads of the cover plate retaining bolts, and the spring pin holes finish blind in both the cover plate and the fixed key. For ease of dismantling, the axially outward facing flat of



MANUFACTURING SCHEDULE:

	QUANTITY	DIMENSIONS XXX·X
UPPER RETRACTION PISTON ASSEMBLY	12	150·0
LOWER RETRACTION PISTON ASSEMBLY	12	750·0

Figure 15. Assembly of the retraction pistons.



HOLE DATA

"A" HOLE 1·0mm Ø DRILLED BLIND TO INTERSECT B HOLE AS SHOWN

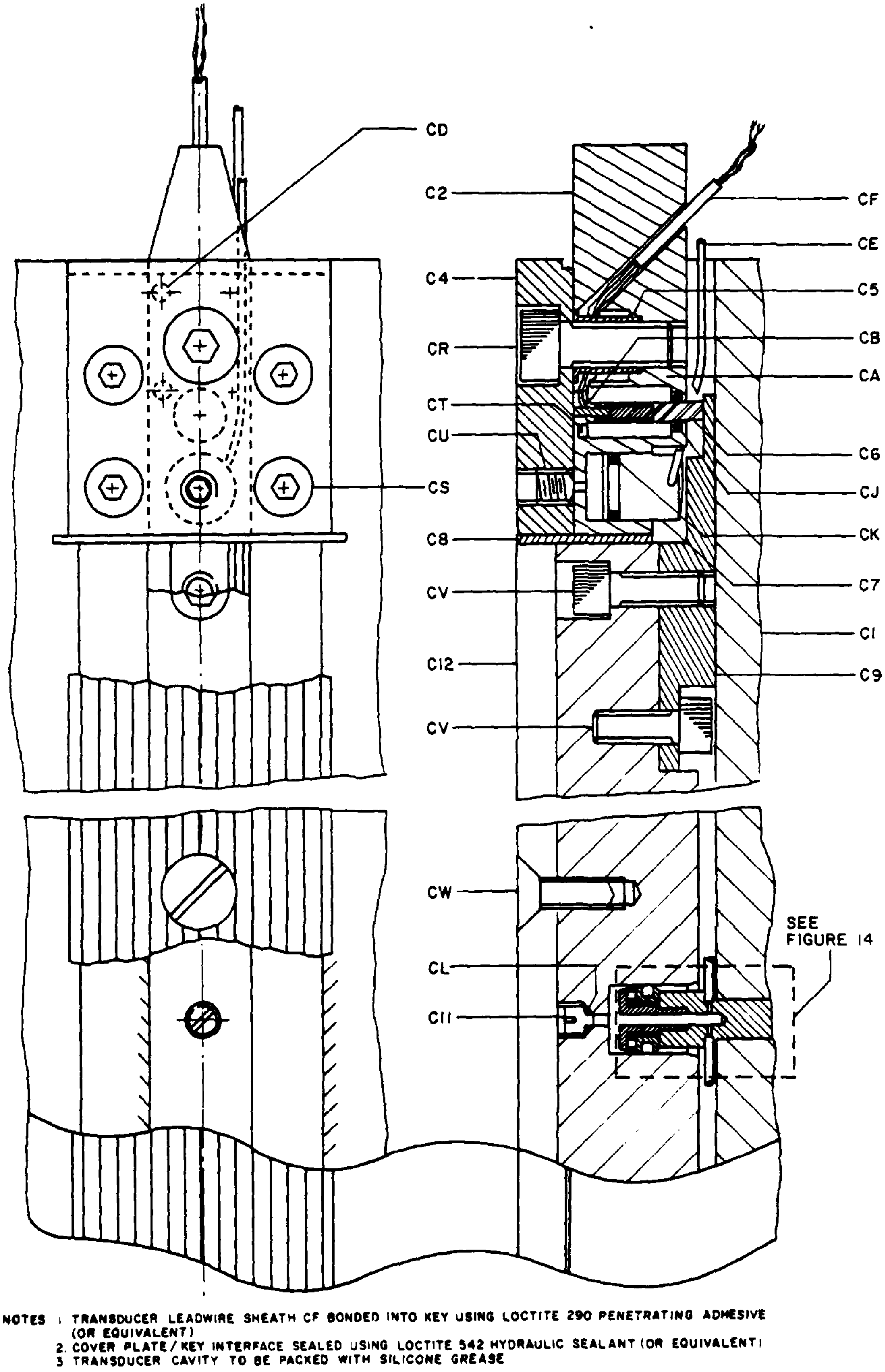
"B" HOLE 1·0mm Ø DRILLED BLIND TO INTERSECT A HOLE AS SHOWN-
COUNTERBORED 1·59mm Ø x 3·0mm DEEP

Figure 16. Retraction piston (component C7).

the cover plates is stepped at the bottom. This feature, together with the outer face of the fixed key, forms a slot when the units are assembled. A lever can thus be used to pr, the fixed key and cover plate apart against the retaining action of the spring pins. During final assembly the interface between the two components is treated with a hydraulic sealant and gasket forming compound (Loctite 542 or equivalent) to seal the displacement transducer cavity.

Also housed in the upper and lower fixed keys are the linear variable differential transformer (LVDT) type transducers for measuring radial displacement of the moving keys. As shown in Figure 13, the transducers are located in a recess which is machined into the body of the key between the central retaining bolt and the retraction actuator cylinder. The plunger of the transducer senses displacement of a heel feature attached to the key which is arranged to extend past the retraction piston. An O ring recess machined into the bottom of the transducer bore allows sealing of the system by acting against the front face of the transducer and the moving core carrier assembly. The magnetic core of the transducer is housed within a plastic carrier which is spring loaded from the cover plate. To ensure minimal friction in the plunger action and to provide additional protection against the ingress of water, the displacement transducer installation is coated and packed with silicone grease during assembly.

In order to route the transducer lead wires past the central retaining bolt to their point of exit from the fixed key body, a special compartment is arranged within the key. A top hat shaped bushing (C5 on Fig. 17) located in the bottom of a recessed section of the bolt hole, is arranged to fit flush with the upper face of the key. The annular compartment so formed around the bolt fully encloses the transducer cables and allows them to lead past the bolt into their drilled passageways in the key without risk of damage



NOTES 1 TRANSDUCER LEADWIRE SHEATH CF BONDED INTO KEY USING LOCTITE 290 PENETRATING ADHESIVE (OR EQUIVALENT)
2. COVER PLATE/KEY INTERFACE SEALED USING LOCTITE 542 HYDRAULIC SEALANT (OR EQUIVALENT)
3 TRANSDUCER CAVITY TO BE PACKED WITH SILICONE GREASE

Figure 17. Assembly of the expansion retraction system (upper fixed key).

KEY TO FIGURE 17

ITEM	QUANTITY	DESCRIPTION
C1	1	Coupling Cylinder Core.
C2	12	Upper Fixed Key.
C4	24	Cover Plate.
C5	24	Top Hatted Bushing.
C6	24	Core Carrier.
C7	12 + 12	Retraction Piston.
C8	2	Thrust Washer.
C9	24	Shear Coupling Key Extension.
C11	60	Bleed Screw.
C12	12	Shear Coupling Shoe.
CA	24	LVDT Transducer (moving key displacement), $\pm$1.0 mm stroke type SM4. Supplier - Sangamo Transducers, North Bersted, Bognor Regis, West Sussex PO22 9BS, UK.
CB	24	Stainless Steel Compression Spring (LVDT core return), 12.7 x 3.05 x 0.41 wire dia (0.152 Kgf/mm). Supplier - Lee Spring Limited, No. 6 Fishponds Estate, Fishponds Road, Molly Millars Lane, Wokingham, Berkshire RG11 2QJ, UK.
CD	96	3 mm dia Spring Pins (cover plate/fixed key retaining), 3 dia x 15.
CE	-	Specification 316 Seamless Stainless Steel Tubing, 1.63 mm (1/16 in) OD x 0.4 mm (0.016 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
CF	-	Specification 316 Seamless Stainless Steel Tubing, 2.38 mm (3/32 in) OD x 0.2 mm (0.008 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
CJ	24	O Ring (transducer core carrier (CG) seal), 3 ID x 1.5 section, ref 206-003.

KEY TO FIGURE 17 (continued)

ITEM	QUANTITY	DESCRIPTION
CK	24	O Ring (retraction piston seal), 10 ID x 1.5 section, ref 206-010.
CL	60	O Ring (bleed screw seal), 1.42 ID x 1.53 section, ref 003.
CR	24	Socket Cap Screw (cover plate/key fastening), M8 x 20 SS.
CS	96	Socket Button Screw (cover plate retaining), M6 x 15 SS.
CT	24	Slotted Machine Screw - Cheese Head (LVDT retaining), M3 x 5 SS.
CU	24	Socket Setscrew - Cone Point (retraction piston bleed screw), M6 x 8 SS.
CV	48	Socket Cap Screw (shear coupling key - heel retaining), M6 x 15S.
CW	120	Slotted Machine Screw - Countersunk Head (shear coupling shoe retaining), M6 x 16 SS.

during assembly or dismantling. As mentioned earlier, the transducer lead wires are sheathed in thin-walled stainless steel tubing (BF, Fig. 8) for the portion of their route from the fixed key assemblies to the services manifold. This encasement emerges directly from the rear of the lower set of fixed keys, and via an inwardly inclined passage of the upper set of keys. The inclined entry for the upper set of keys is incorporated because these units do not fit flush with the end of the coupling cylinder as do the lower set. The keys are tapered in the section leading axially out of the coupling cylinder. This is required to implement the special wedge coupling described in Section II-3a. After leaving the inner face of the keys, the sheathed lead wires bend upwards to join the cables from the lower section. These cables are routed through the coupling cylinder in the bottom of the keyway grooves and emerge from underneath the upper fixed keys. As shown in Figure 18, the cable sheaths from the

Figure 18. Routing of the LVDT lead wire sheaths into the lower fixed key assemblies

lower set of keys emerge from the bottom of the coupling cylinder and are bent back to pass below the fixed keys. The cable sheaths for both the upper and lowe fixed keys are sealed and bonded into the bodies of the units using a penetrating adhesive (Loctite 290 or equivalent). This compound is also used during final installation of the top-hatted bushing and central retaining bolt assembly to prevent water entering from around the bolt hole.

c. The Soil Particle Screen The soil particle screen is required in order to prevent the ingress of particles into areas of the BSD probe where there are moving parts. The most likely source of problems in this regard is the coupling cylinder section where the surface of the probe is comprised of discrete segments. These segments, the shear coupling shoes, are free within certain bounds to move both radially and to tilt. Therefore, there must necessarily be working clearances in the system. If the screen is to be effective, these clearances must be minimized and controlled for all conditions or states of expansion of the coupling cylinder. It should be noted that the soil particle barrier system need not, and indeed as outlined here is not, watertight. The system allows the surrounding groundwater to permeate all regions of the coupling cylinder, the principal object being only to prevent abrasive particles from entering the assembly and damaging machined surfaces or causing siezure of the expansion/retraction systems.

For the ends of the moving keys, the axial load thrust washers are used to implement the particle barrier. As described in Section II-4b the specially shaped pair of plastic washers are fitted over the BSD core, the inner contour being shaped to locate with the keyways in the core, and the outer of the washers being circular to correspond with the cylindrical profile of the probe. The retraction system limits the maximum radial outward displacement of each moving key to 2.5 mm, thereby ensuring that the undersides of the shear coupling shoes do not approach the contracted position of the outside of the probe. As a consequence, there is always a portion of the end of the shoes in contact with the thrust washers to affect a particle barrier. The axial fit of the shoes is arranged to be moderately tight in order to maintain the seal. The durable and deformable nature of the plastic from which the washers are manufactured (Delrin)* is used here to advantage.

The arrangement shown in Figure 19 has been implemented for the portion of the barrier along the edges of the shear coupling shoes. Here, the screen is formed by a length of stainless steel strip inserted into grooves machined in the edges of the shoes. The strip is free to float within the grooves so that it does not interfere with the jacking-out and retraction functions, but at the same time, a certain amount of contact is made between the strip and the shoes. The combination of this contact, plus the inherent barrier properties of the turn-corner arrangement, are utilized to form the screen. During manufacture of the screen (Fig. 19) difficulties were experienced in machining the grooves in the edges of the shear coupling shoes. The cutting tool's small size and the difficulties in holding the relatively awkward shape of workpiece frequently caused the cutter to break. To overcome this problem an alternative method of producing the slot was devised where the underside of the shoes are machined with a double stepped profile. As shown in Figure 20, the slot is formed when a special insert strip is positioned to replace part of the removed metal. The strip is located laterally by the first step, and the width and depth of the slot is controlled by the second step. In

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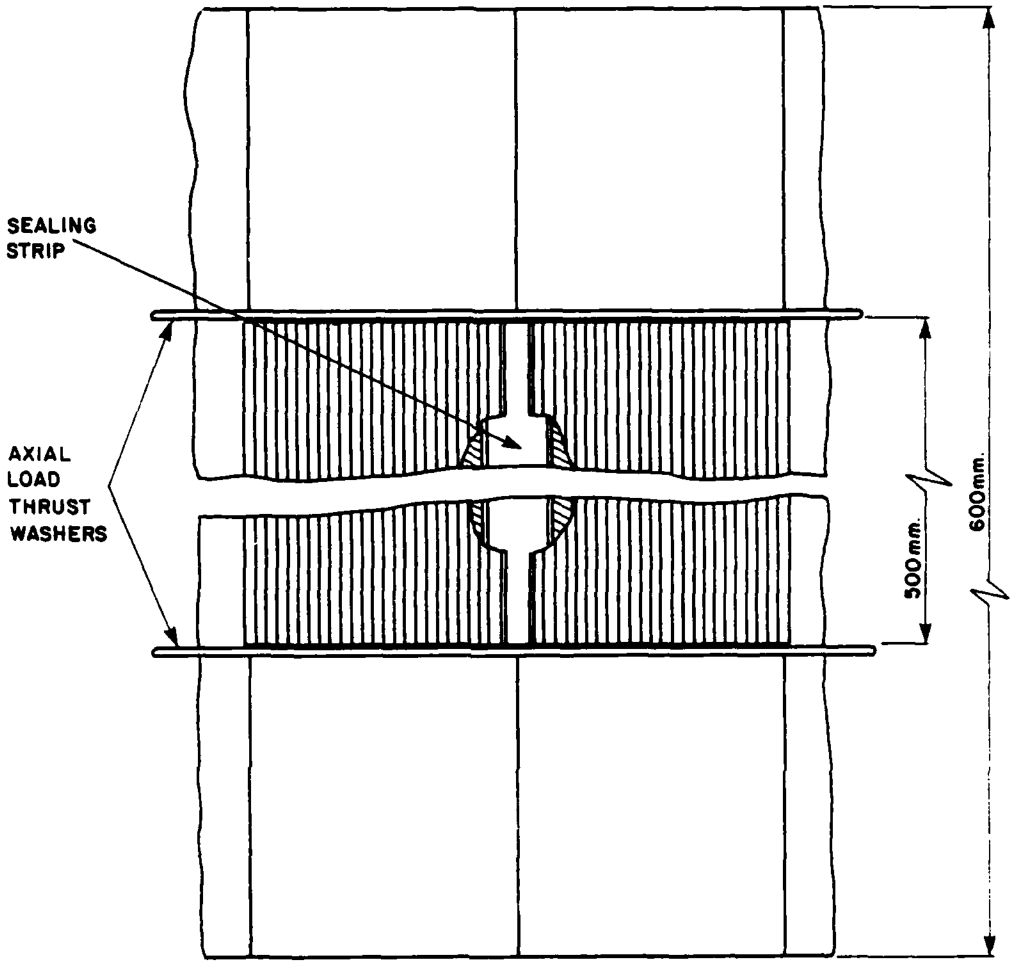
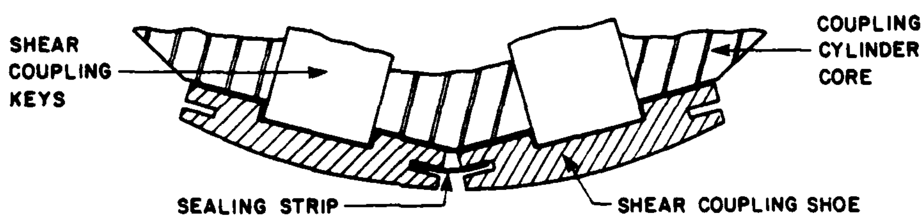
*Delrin is a proprietary name for a synthetic product similar to nylon but with improved mechanical properties.



a) View of the shear coupling shoes



b) Section through the shoes

Figure 19. The soil particle screen.

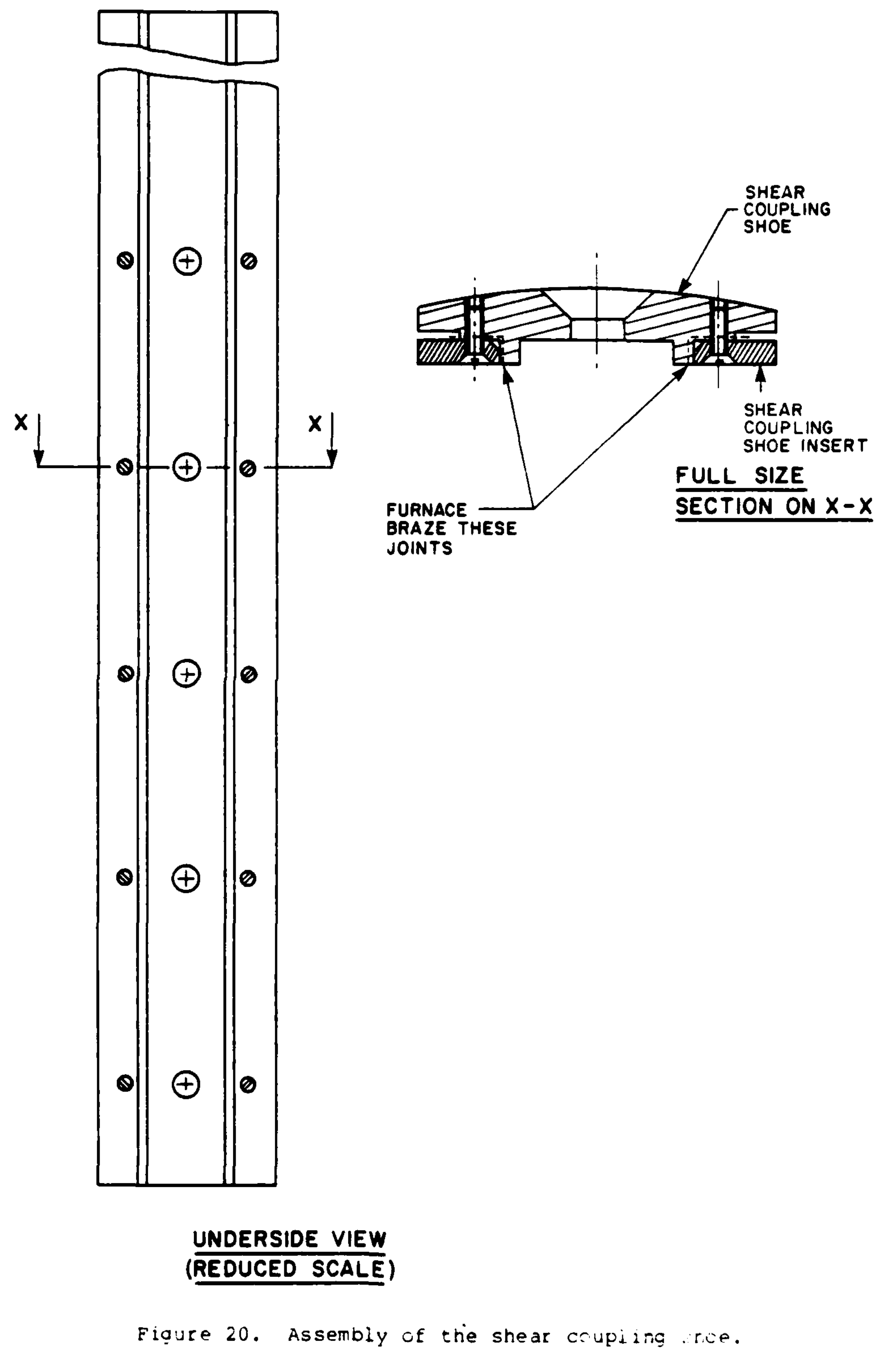


Figure 20. Assembly of the shear coupling [illegible]nce.

order to achieve maximum strength for the shear coupling shoe assembly, the inserts were brazed into place. This was done in an inert atmosphere furnace and, to ensure that the assembly remained stable during the process, the strips were initially jigged into position by a series of retaining screws.

As a further step towards increasing the effectiveness of the screen and in minimizing interference with the jacking and retracting functions, the grooves in the shear coupling shoes are filled with silicone grease. The lubricating properties of this material, although not optimum, are adequate for this application. The high viscosity, and the inherent inertness of the substance were the chief reasons for the choice.

5. LOWER PROBE SECTION

a. The Self-Boring Head The self-boring head is the principal feature of the lower section of the BSD probe. The heart of the system is a special cutting head that (acting on applied normal load) channels or moulds the removed soil into a throat section. The basis of the method is that the stresses induced in to the remoulded material by the applied load act to balance the in situ stresses in the undisturbed soils. It is therefore an inherent part of the process that an axial or push-down load be applied to the system while emplacement is in progress. During retrieval of the probe from surrounding soil, the head must also undergo an upward or pull-out load to overcome the frictional effects at the soil/probe interface. While the design of a system purely to withstand these loads presents little difficulty, it is a different matter when the requirements of the rotation measurement system are considered. Here, as outlined in Section II-1, advantage is taken of the fact that the head is emplaced beyond the region of influence of the coupling cylinder and, hence, can be used to provide measurement data. The principal design problems for this part of the probe arose from this requirement, and were centred around the search for a system which could withstand the reversing axial load, yet provide smooth rotational freedom between the coupling cylinder and the self-boring head.

The final design for the lower probe section is shown in Figure 21. Structural attachment to the core of the coupling cylinder occurs with the upper bearing platform (D1, Fig. 21). This unit, though bolted to the core of the coupling cylinder, provides a machined recess over which an upper bearing (DF, Fig. 21) is permanently fitted. Push-down load is transferred through this bearing to the outer sleeve component (D2, Fig. 21). The cutting shoe (D4, Fig. 21) is screwed into the outer sleeve and compressive load transfer occurs at the machined mating faces of this joint. Running concentrically within the lower probe section is a specially machined auger tube extension section (D5, Fig. 21). This component, as the name implies, forms the auger encasement for the lower probe section, but also serves to couple tensile loading into the upper bearing platform. The tensile load arises from pull-out reaction and it is coupled to the cutting head from the extension tube via the lower bearing platform and the lower bearing D9, DG. The upper and lower systems of bearings are maintained in preload by the action of the retaining nut DH on the extension tube and the preload is coupled between the bearings as tension in the extension tube, and compression in the outer sleeve. A series of O ring piston seals (DM and DQ, Fig. 21) at the threaded junction of the outer sleeve and the cutting shoe provide watertight sealing of the transducer compartment. Here, freedom for relative rotation between the extension tube and the cutting head is ensured by the allowance of generous clearance for the inner sealing ring DR.

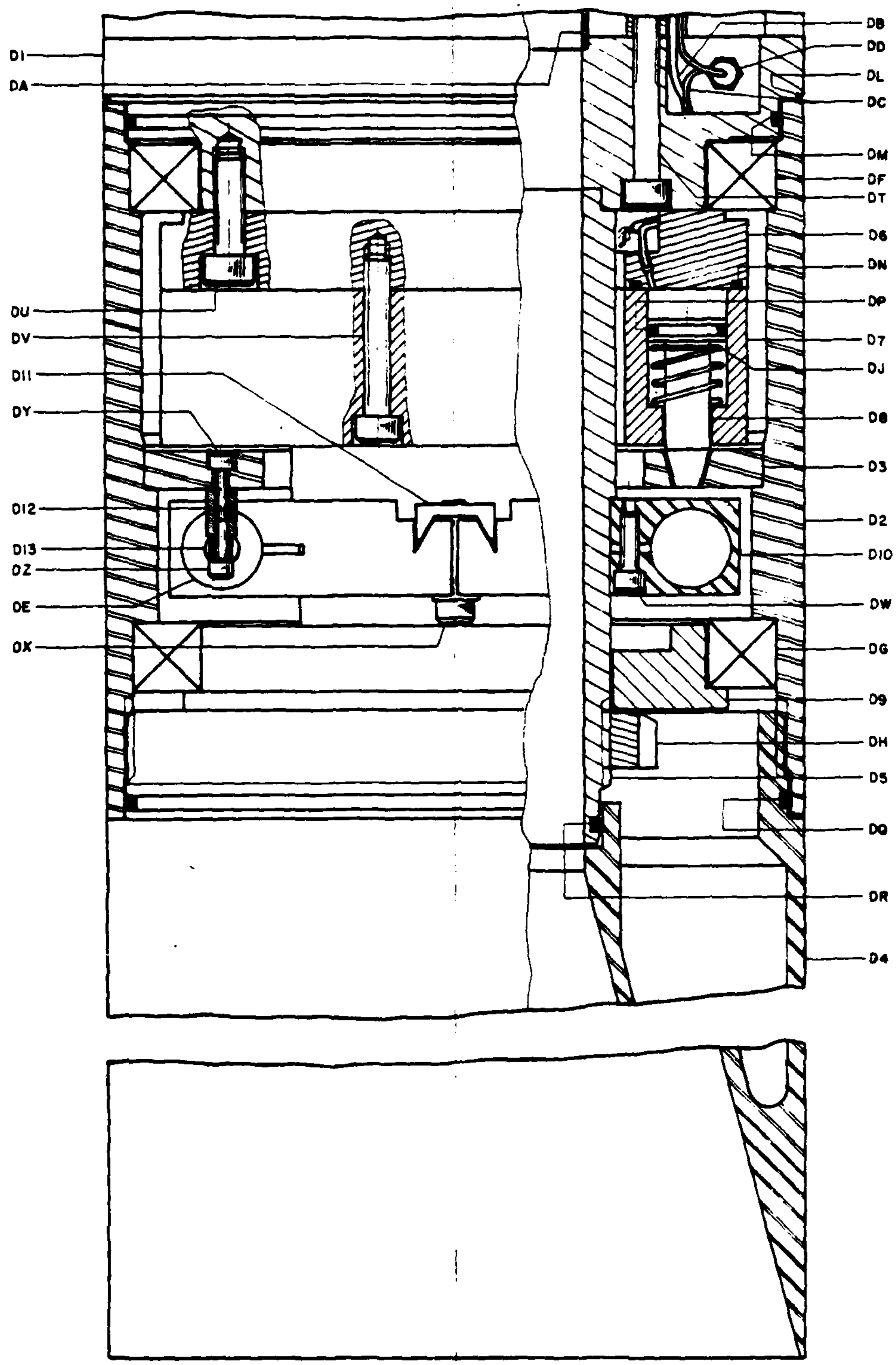


Figure 21. Assembly of the lower probe section.

KEY TO FIGURE 21

ITEM	QUANTITY	DESCRIPTION
D1	1	Upper Bearing Platform.
D2	1	Self-Boring Head Outer Sleeve.
D3	1	Datum Locking Plate.
D4	1	Self-Boring Head Cutting Shoe.
D5	1	Auger Tube Extension.
D6	1	Cylinder Mounting Plate.
D7	1	Cylinder Block.
D8	4	Locking Piston.
D9	1	Lower Bearing Platform.
D10	1	Transducer Mounting Collar.
D11	1	Wedge Clamping Piece.
D12	2	Core Mounting Collar.
D13	2	Core Rod End.
DA	1 x 605 mm length	Specification 316 Seamless Stainless Steel Tubing (auger encasement in coupling cylinder section), 66.68 mm (2 5/8 in) OD x 1.63 mm (1/16 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
DB	-	Specification 316 Seamless Stainless Steel Tubing (transducer lead wire sheath), 2.38 mm (3/32 in) OD x 0.2 mm (0.008 in) wall. Supplier - Tubesales (UK) Limited West Bay Road, Southampton SO9 5HQ, UK.
DC	-	Specification 316 Seamless Stainless Tubing (hydraulic pressure supply), 1.63 mm (1/16 in) OD x 0.4 mm (0.016 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
DD	1	Hydraulic Compression Fitting (pressure supply coupling), 1.63 mm (1/16 in) type "Gyrolok" 1U. Supplier - Hoke Incorporated, One Tenakill Park, Cresskill, N.J. 07626, USA.

KEY TO FIGURE 21 (continued)

ITEM	QUANTITY	DESCRIPTION
DE	1	LVDT Transducer (static rotation measurement), ±1.27 mm (0.05 in) stroke, type 050 HR. Supplier - Schaevitz Engineering Incorporated, PO Box 505, Camden, New Jersey 08101.
DF	1	Deep Groove Ball Bearing (upper support bearing), 130 x 165 x 18, SKF ref 61826.
DG	1	Deep Groove Ball Bearing (lower support bearing), see DF.
DH	1	Bearing Retaining Nut (lower bearing platform retaining), M80 x 2, SKF ref KM16.
DJ	4	Compression Spring (locking piston return spring), 19.05 mm (3/4 in) dia x 25.4 mm (1 in) x 2.94 mm (0.116 in) dia wire, Entex ref 131. Supplier - Entex Springs Limited, Glaisdale Drive, Bilborough, Nottingham NG8 4JY, UK.
DL	1	O Ring (outer particle seal ring), 173 ID x 1.78 section - Dowty ref 562 (can be made from 1.78 section cord).
DM	1	O Ring (outer sleeve/upper bearing platform piston seal), 158 ID x 2.62 section, international size ref 164 (can be made from 2.62 section cord).
DN	4	O Ring (locking actuator cylinder chamber seal), 24 ID x 2.0 section, ref 204-024.
DP	4	O Ring (locking actuator piston seal), 16 ID x 2.0 section, ref 204-016.
DQ	1	O Ring (outer sleeve/cutting head piston seal), See DM.
DR	1	O Ring (cutting head/auger extension tube piston seal), 69.5 ID x 2.62 section, international size ref 148.
DT	12	Socket Cap Screw (upper bearing platform retaining), M6 x 50 SS.
DU	4	Socket Cap Screw (cylinder mounting plate retaining), M8 x 25 HTS.

KEY TO FIGURE 21 (continued)

ITEM	QUANTITY	DESCRIPTION
DV	6	Socket Cap Screw (cylinder block retaining), M6 x 45 HTS.
DW	1	Socket Cap Screw (transducer wedge clamp-locking), M6 x 25 HTS.
DX	1	Socket Cap Screw (transducer mounting collar-retaining), M4 x 16 HTS.
DY	2	Socket Cap Screw (core mounting pillar-retaining), M4 x 10 HTS.
DZ	2	Socket Cap Screw (rod end block-retaining), M4 x 10 HTS.

Since the geometry of the cutting head is determined by soil type, it was felt that the cutting head should be easily replaced or changed. As mentioned earlier, this is achieved by screwing the shoe into the outer sleeve component. The outer sleeve is fully supported on the upper and lower ball bearings and provides axial restraint for the shoe as well as helping to establish the rotation datum. The cutting shoe is a one-part machined component which is internally relieved to reduce weight. It is proposed that this component be assembled or removed by applying opposing torques to the cutting shoe and the outer sleeve via a pair of friction strap wrenches. In order to gain access to the transducer compartment or the datum locking mechanism, the upper and lower support bearing system has been designed to be demountable. Locating fits are arranged both between the outer ring of the upper bearing and the outer sleeve, and the inner ring of the lower bearing and the lower platform. By this arrangement, the bearing retaining nut and the lower bearing platform are the only structural components that are required to be removed prior to withdrawal of the outer sleeve component. The lower bearing comes away with the sleeve being positively retained in its seating recess, and the lower bearing remains in position located and secured onto the upper bearing platform. The outer sleeve is piston sealed with the bearing platform by O rings (DM and DL, Fig. 21). This dual ring arrangement was incorporated to provide watertight sealing of the transducer compartment and to prevent soil particles from being urged into the clearance space by the lateral earth pressures.

b. <u>The Datum Locking Mechanism</u> The datum locking mechanism function is twofold:

(1) To provide repeatable mechanical centering of the self-boring head/coupling cylinder system.

(2) To prevent gross relative rotation occuring between these two systems if the soil yields during testing.

The need for mechanical centering of the cutting shoe arises from the characteristics of one of the rotation measurement systems required for instrumenting the BSD test. As mentioned earlier, static rotation of the coupling cylinder is detected by its movement relative to the self boring head. Any arbitrary offset or shift in the relative positions of the coupling cylinder and the cutting shoe will reduce the available range of measurement by introducing an offset into the output of the transducer (see Section II-7c). The centering mechanism overcomes this problem by locking the self-boring head at the center of the transducer measurement range during emplacement. Since the system is spring released it requires pressure only to engage center and to maintain lock. This is achieved from a single hydraulic supply line tapped into the coupling cylinder retraction system.

As shown in Figure 21, the datum locking mechanism achieves the centering action by hydraulically driven taper plugs (D8, Fig. 21) The plugs are arranged to align with conical recesses in an annular plate (D3, Fig. 21) attached to the outer sleeve of the cutting shoe. When driven home in the seating recesses, the plugs can apply considerable torque to establish and maintain centering. The cylinders for the hydraulic actuators are machined into a special cylinder block component (D7, Fig. 21) which is attached via a mounting plate (D6, Fig. 21) to the upper bearing platform (D1, Fig. 21). Jacking forces developed by the centering system act axially with respect to the probe and flow around the loop bounded at the top by the upper bearing platform, and at the bottom by the bearing retaining nut. When the rotation measurement system is required, supply pressure is isolated from the centering system and the hydraulic circuit is connected to drain into the reservoir. The hydraulic locking plungers are spring loaded and, hence, can provide the necessary force to displace oil from the system and return to their retracted position at reasonable speed. In this position the taper plunger still protrudes into the seating plate, but this time a clearance occurs in the system. This clearance allows rotation of the self-boring head but is arranged so that this movement can only slightly exceed the full displacement range of the transducer. If any further relative rotation occurs, the taper plungers engage with the cone seatings in the datum locking plate, thereby applying torque to the cutting head. By this means the head follows the coupling cylinder with a backlash action and overranging of the transducer, with the resultant risk of damage to the core mounting components, is avoided.

To maintain watertightness for the transducer compartment, the transducer lead wires and the hydraulic supply pipe for the locking function are routed through a special countersink seal arrangement which is configured in the top face of the mounting plate D6. To supply the actuators, oilway ports are drilled through the mounting plate to break into the cylinder bores. Two ports are provided for each cylinder and the cylinders are interlinked by a chain of bridging pipes. The ports are counterbored to accept the pipes, which are then silver soldered into the mounting plate. The cylinder block is secured in position by the retaining bolts (DV, Fig. 21) which provide restraint for both the hydraulic pressures acting on the locking pistons, and the spring retraction forces. The actuator cylinders are sealed by O ring face seals (DN, Fig. 21) machined into the mounting block D6.

c. <u>The Rotation Transducer Assembly</u> It has been mentioned previously that two rotation measurement systems are proposed for use in the BSD test:

(1) A static displacement sensing system where the relative rotation of the self-boring head with respect to the coupling cylinder is sensed by an LVDT type transducer.

(2) A dynamic measurement where an inertial transducer senses absolute rotation of the coupling cylinder.

The arrangement for the rotation measurement systems in the lower probe section is shown in Figure 21. The system is housed within the lower probe section in a compartment between the cutting shoe of the self-boring head and the datum locking mechanism. The entire compartment is hermetically sealed to protect both the electrical measurement system and the ball bearings used to provide rotational freedom for the self-boring head. Sealing is achieved by O ring piston seals acting both statically (DQ, Fig. 21), and dynamically (DM and DR, Fig. 21). The upper bearing platform (D1, Fig. 21) forms the barrier between the wet regions of the coupling cylinder section and the sealed lower section. Services to the lower section (i.e. the hydraulic supply and the transducer lead wires) are led through countersink seals in a manner similar to that for the services manifold baseplate. The seals are configured in the cylinder mounting plate (DG, Fig. 21) and are activated by the clamping pressures provided by the retaining bolts (DV, Fig. 21). After passing through the countersink seal arrangement the services enter specially provided cavities in the mounting plate. These cavities, formed by radial slots machined into the lower face of the plate, provide space for routing and for connecting the hydraulic supply to the locking actuators. The transducer lead wires are diverted through certain cavities into the annular clearance between the mounting block and the auger tube extension. In this position they pass through the locking system to their destination in the lower compartment.

Rotation of the coupling cylinder for both the static and dynamic systems is sensed by measuring the movement of the central auger extension tube component D5. This tube is bolted to the coupling cylinder core and by extending down to the throat of the self boring head, transfers the rotational movement of the cylinder through the datum locking section into the transducer compartment. Mounting for the transducers is provided by a special collar (D10, Fig. 21) seated on a machined recess on the auger extension tube. This collar features a novel wedge driven clamping arrangement which minimizes disturbance to the transducer setting during final tightening of the assembly. As shown in Figure 21, the collar has two lateral through holes for mounting the measurement devices, each of which is provided with a split clamp arrangement for positive retention. For the static rotation (LVDT) transducer, the core of the device is mounted via spacers D12 on the datum locking plate (D3, Fig. 21). A pair of spacers provide secure mounting for both ends of the core, and the unit is located within the transducer by a stainless steel threaded rod attached to the core rod end blocks (D13, Fig. 21). The arrangement provides a stable mounting for the transducer core and, by the use of the threaded rod, allows fine adjustment of the core position. Assembly or dismantling of the system is assisted by ensuring that the rod end blocks are able to pass through the central aperture in the transducer. This permits assembling the transducer into the mounting collar before the core is fitted. The core assembly can thus be assembled away from the probe requiring relatively little in situ fitting to attach it to the locking plate. Following final adjustment of the core position (performed with the datum locking mechanism activated), the lock nuts on the core carrier rod (Fig. 22) are tightened home to retain the setting.

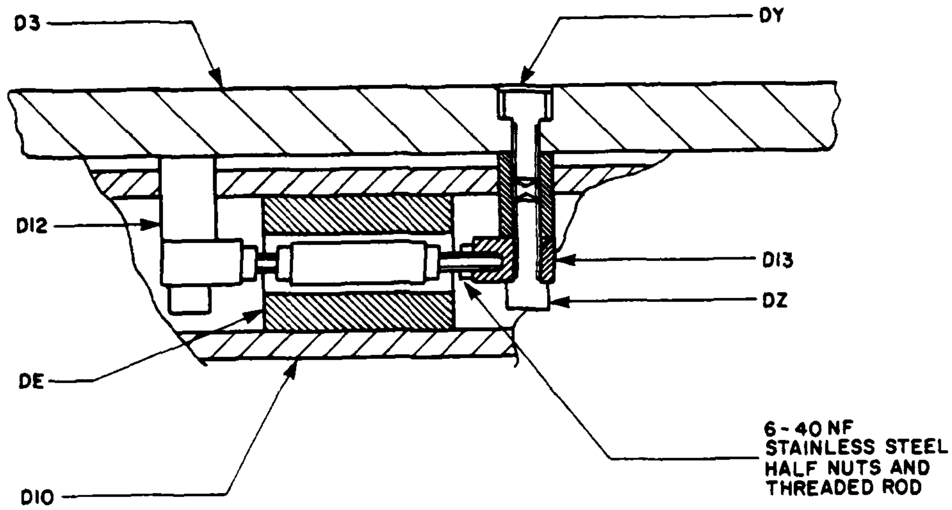


Figure 22. Assembly of the LVDT core mounting.

KEY TO FIGURE 22

ITEM	QUANTITY	DESCRIPTION
D3	1	Datum Locking Plate.
D10	1	Transducer Mounting Collar.
D12	2	Core Mounting Collar.
D13	2	Core Rod End.
DE	1	LVDT Transducer (static rotation measurement), $\pm$1.27 mm (0.05 in) stroke, type 050 HR. Supplier - Schaevitz Engineering Incorporated, PO Box 505, ref 148.
DY	2	Socket Cap Screw (core mounting pillar-retaining), M4 x 10 HTS.
DZ	2	Socket Cap Screw (rod end block-retaining), M4 x 10 HTS.

Both rotation transducers are mounted the same, i.e., the lateral through holes machined into the mounting collar component. This is primarily to house the LVDT device. The through hole diameter is capable of directly accepting the recommended transducer. However, some form of adaptor will be required for mounting the inertial transducer. It is suggested that the adaptor be a cylindrical plug with the same diameter as that of the LVDT transducer. If a standard piezoelectric accelerometer of nominal sensitivity in the region of 25 $pC/m/s^2$ is used, the mounting arrangement for the device will almost certainly be a central stud with a standard thread such as 10-32 Unified Screw Thread, Fine Series. The cylindrical plug can be correspondingly threaded to provide a mounting for the transducer, and at the same time be used to electrically isolate the device from the mounting collar. This may be found to be a desirable feature since the more common types of piezoelectric accelerometers often do not possess isolated cases. With these devices it is necessary that an insulated mounting be provided in order to avoid problems with earth loop effects.

6. SELF-BORING

a. General Discussion In the Phase I feasibility study for the Borehole Shear Device it was proposed that the system should utilize self-boring emplacement, and that the soil cuttings produced by the self-boring head should be transported through the probe by a helical flight auger. Perhaps the most commonly known self-boring geotechnical tool is the Cambridge Camkometer which uses water flush to raise the soil cuttings. The possibility of similarly using water flush for the BSD was considered, but it was felt that for sands or highly permeable media, there was an undesirable risk of disturbance to the in situ soils by erosion. Early on in this second phase of the BSD contract, the PM In Situ company, geotechnical consultants and commercial operators of the camkometer equipment, were invited to collaborate with Dames & Moore to consider the problem of designing a self-boring head for the BSD probe for use in sand. PM In Situ were found to be in favour of the water flush technique for removing the soil cuttings and, following their preliminary study of the problem, produced a schematic design using the water flush technique. However, no conclusion was reached as to the relative merits and potential problems of the originally proposed flight auger method. Consequently it was decided, in view of the practical simplicity of the method and the lack of problems associated with soil erosion, to attempt some experimental work to determine the feasibility of using a flight auger to raise soil cuttings through the BSD probe.

b. Experimental Work on Soil Lifting The first experiment was a test to assess the feasibility of using pressure induced in the soil at the throat of the cutting head to produce the required lift. The apparatus designed for this test simulated the BSD probe with two concentric steel tubes headed with a cutting edge and a conical throat section. A special drive rod running in the center of the apparatus was fitted with an array of protruding pins, the object being to continuously break-up any structures that might form in the sand. The central rod was manually driven and push-down load was applied via a system of dead weights. The soil sample (a medium-fine building sand) was contained within a bin around which a simple structure was erected to support the apparatus. The result of the test implied that little success was likely to be achieved with this simple method of soil lifting. After only a small head of sand was developed in the central bore of the model probe, the breaker rod siezed. It was thought that the sand, under the shearing action of the central rotating rod, was locking the system due to a dilating action. The conclusion was that to prevent this happening the sand should not be allowed to deposit in the central passageway in any significant density. It was felt that the turbine action of a helical flight auger might be successful in achieving this, and accordingly, a further experiment was made.

For t second experiment, a scale model of the BSD apparatus was constructed. As shown in Figure 23, the sand sample was contained within a stout cylindrical bin, and a guide frame was constructed over the bin to support the probe and to maintain alignment. A yoke arrangement across the top of the probe was coupled to the soil bin via a pair of hydraulic pulling cylinders to the soil bin thus enabling a pushdown load to be applied to the probe. The auger used is a commercially available 60-mm soil drilling unit suitable for use in the actual BSD probe. In the experiment the auger was powered by a heavy duty single phase electric drill which was rated at 1 kW maximum output at 470 r/min. The test was conducted using a fine, partly rounded micaceous sand, 100 to 300 BS size sieve (0.05 to 0.1-mm) single sized.

The experiment (Fig. 24) using the model self-boring probe (Fig. 25) was noteably successful. No problems with siezure or fouling of the auger were experienced, and the rate of flow of sand upwards through the 2-m length of lift was continuous and steady. Moreover, the hydraulic pulling facility was not required since the weight of the probe (approximately 180 lb) was adequate to maintain penetration into the sand.

A summary of the testing procedure follows:

(1) The soil bin was filled to approximately 0.6 m height, and the probe was hoisted into place and rested on the surface of the sand.

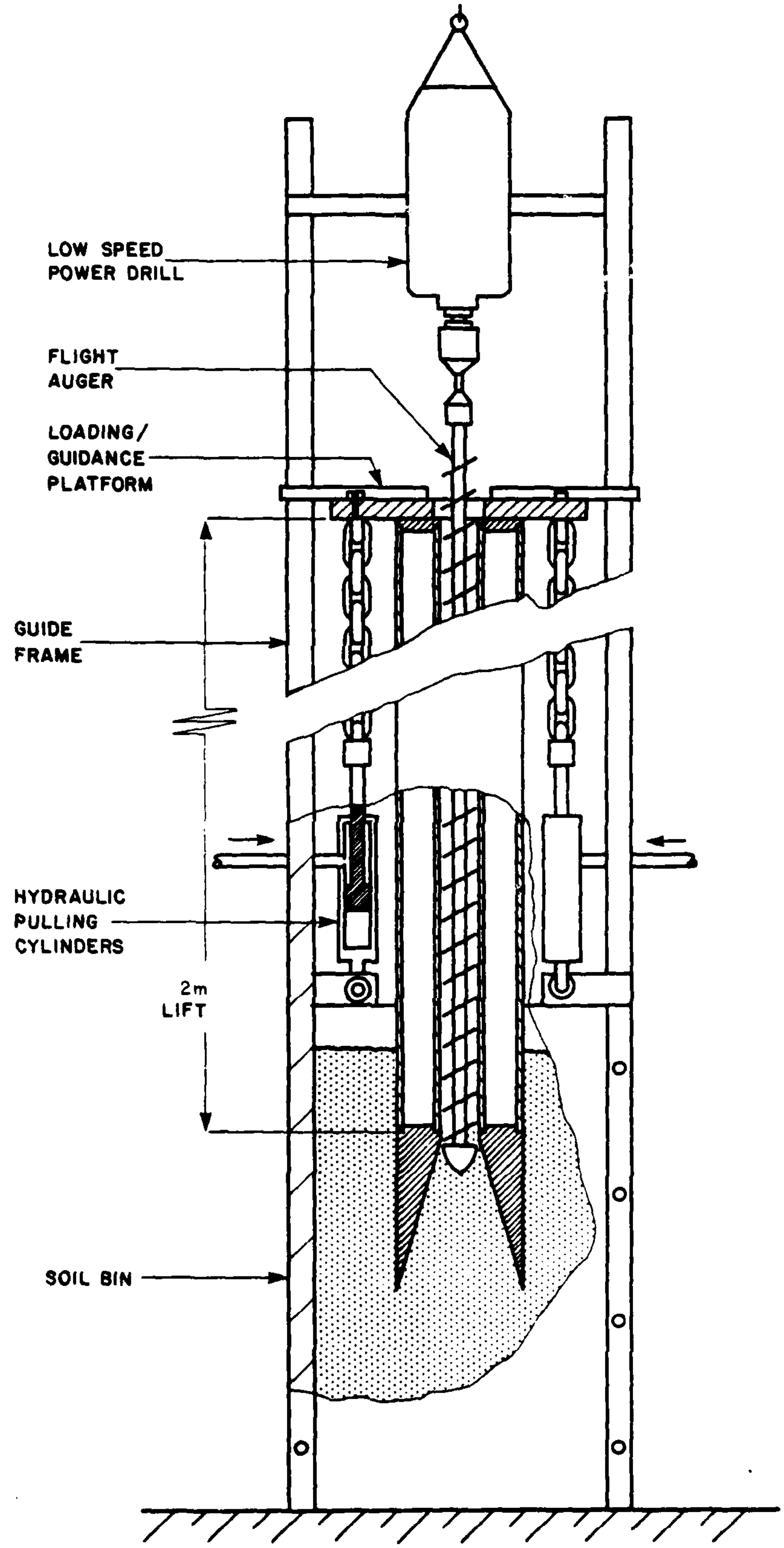


Figure 23. Schematic of the auger assisted soil lifting experiment.

Figure 24. General view of the soil lifting experiment.

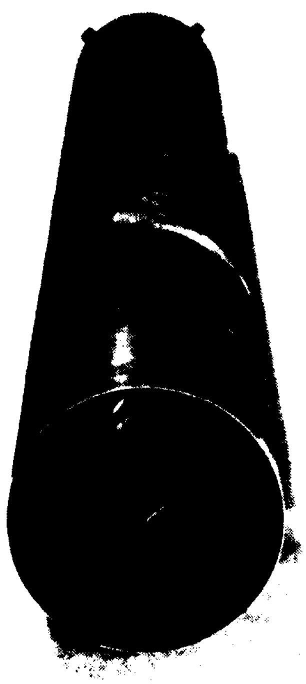


Figure 25. View of the model BSD probe.

(2) The guiding system was attached to maintain both the top and lower end of the probe roughly concentric with the soil bin.

(3) The auger (Fig. 26) was lowered into the central tube of the probe, and attached to the power drill. The drill in turn was attached to a hoist to provide controlled lowering and to react against the downward pull of the auger (Fig. 27). The guidance frame was used to provide torque reaction for the drill.

(4) The auger was turned on and gradually lowered into the throat of the probe.

(5) As the probe bored into the sand, the auger was lowered to follow it and maintained a steady flow of displaced sand (estimated at 5 l/min).

By following the above procedure, the model probe penetrated the sand progressively and smoothly. The drill auger head used to stimulate self boring was approximately midway into the conical section of the cutting shoe. This would not necessarily be the optimum position for a true self-boring emplacement since the relative position of the region where soil breaking occurs would be a critical factor affecting the stress state of the undisturbed soil. However, the experiment did amply demonstrate that an auger can be used to raise noncohesive soils through the probe, and that for these soil types water flushing is not necessary.

c. Implementing Self-Boring in the BSD Probe It will be evident from the various discussions presented in this report that the self-boring head has been regarded as a fundamental feature of the BSD apparatus, and that the design and development work has been, where appropriate, directed towards accommodating a practical scheme into the hardware. The experimental work on

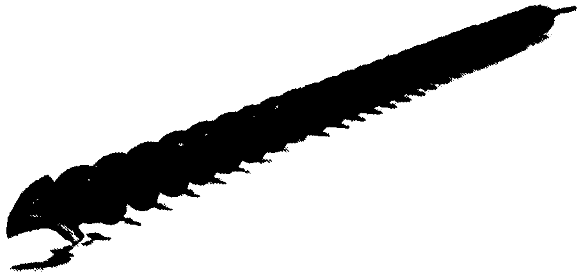


Figure 26. The auger used in the experiment.

Figure 27. The soil lifting experiment in progress.

soil lifting described earlier has been conducted under realistic conditions using an auger which is suitable for use in the final design for the probe and it is felt that, as well as validating the basic method, the testing has provided some useful system design data. Included in the designs produced for this second phase contract effort is that for a self-boring head or lower probe section. In this system the cutting shoe is detachable, thereby permitting use of units designed for specific soil types. It can be thus appreciated that the steps involved in the production of a fully operational self-boring system for the BSD are relatively straightforward. This section of the report deals with the part of the self-boring system for which it is difficult to recommend or propose a uniquely suitable design; this is the arrangement for mounting and driving the auger. Basically, there are two options for this part of the system, one being simple and hence rapid to implement, the other being more sophisticated and offering greater convenience of use. The choice between these systems is, it is felt, best made at the time of drawing up the plans for the field trials of the apparatus. If emplacement of the BSD probe is critical, then the sophisticated system would be an asset. If on the other hand, control of BSD probe emplacement is not regarded as critical, the simple drive system would be the obvious choice.

The alternatives for the auger drive system are:

(1) Mount the drive unit within the torque tube string and at the point where the auger encasement emerges from the catch tank section of the torque tube.

(2) Drive the auger from an uphole source via a system of extension rods.

Alternative (1), requires a downhole power source. As shown in Figure 28 a motor (probably a direct drive low speed hydraulic motor), would be mounted at the end of the string of auger tubes. The number of auger tube extensions used depends upon the depth of self-boring emplacement (i.e., the volume of soil to be stored in the torque tubes), and the torque reaction for the auger drive (estimated to be approximately 10 N·m/m of auger) would be provided by the final auger tube extension socket. Axial loads on the auger could be absorbed either by the motor shaft bearings or by an additional thrust bearing and would again be coupled into the torque tube string by the auger socket. No difficulties would be experienced in accommodating a hydraulic motor of the rating discussed here within the torque tubes, and the hydraulic supply could easily be routed to it *via* flexible hoses from an uphole source. A hydraulic motor would be an ideal choice for the auger drive system from a number of viewpoints:

(1) The power to volume ratio is high compared to electric motors.

(2) Both torque and speed are controllable by relatively simple hydraulic circuitry. Powerful starting torque is available if required.

(3) The units are inherently watertight, impervious to solid matter, and extremely rugged.

(4) The units are intrinsically safe.

(5) Suitable hydraulic supplies are frequently found on site.

Mounting the auger drive system uphole on the other hand, is potentially a simpler expedient. Here, the drive unit could again be coupled directly to the auger, providing axial restraint and torque reaction from uphole. In the case of a deep emplacement, the drive to the auger would be by a simple system of extension rods, although it is likely that this would

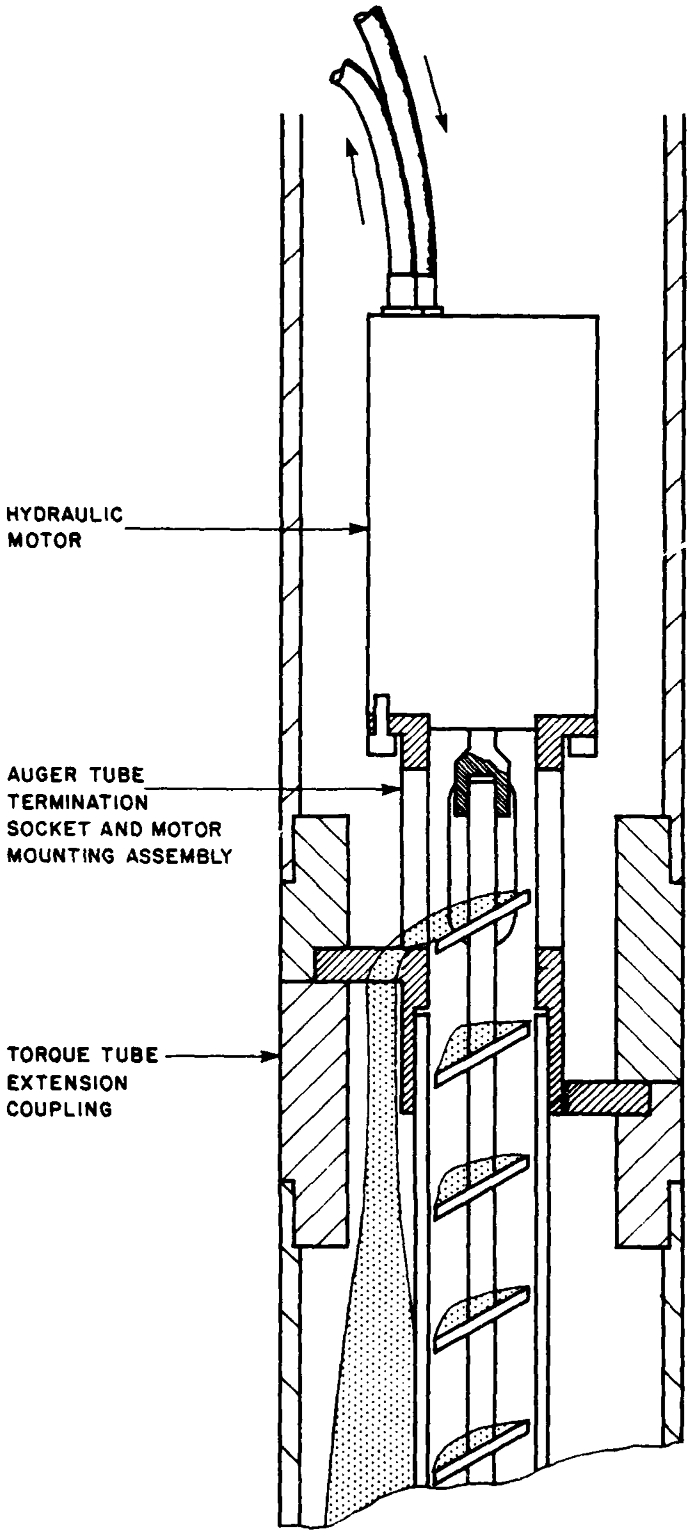


Figure 28. Schematic of a downhole auger drive motor system.

require stabilizing by using bearings at given intervals. The chief drawback is that it could interfere to a certain extent with the arrangement used to provide push-down emplacement loading. Clearly these two systems must be in operation simultaneously, and the need for direct overhead access to the central bore of the torque tube string might lead to difficulties for certain configurations of the vertical load system. The downhole motor system would not present problems since the auger drive power could be routed along with the services conduit through a side entry slot in the cap of the torque tube string.

The above discussions summarize the extent of the work conducted on self-boring emplacement of the BSD probe during this second phase development contract. It is felt that the viability of the method has been established, and that the detailed design of the remaining hardware for two alternative schemes, one suited to a simple field trial, the other to a more sophisticated testing programme, should be straightforward.

7. INSTRUMENTATION OF THE BSD PROBE

a. General Discussion The fully instrumented BSD probe would feature the following measurement systems:

(1) Applied torque.

(2) Rotation of the coupling cylinder.

(3) Applied normal load (emplacement load).

(4) Lateral jacking pressure.

(5) Displacement of the segments comprising the surface of the coupling cylinder.

The torque and rotation measurements are, of course, fundamental to the BSD test, and are the systems requiring the greatest dynamic range of measurement; of particular importance in this respect is the rotation measurement. This parameter, corresponding to shear strain in the soil, extends over a considerable range as the deformations in the soil range from those of elastic response up to the values associated with yielding.

The normal load measurement has been included to allow monitoring and control of the emplacement process. The measurement provides information on the push-down forces being applied to the coupling cylinder and lower probe sections, and monitors the dead load acting on the system. A valuable additional benefit of this is that the normal load measurement transducer uses the same sensing element as the torque transducer. Being necessarily a relatively highly stressed component, the transducer is liable to be damaged if overloaded. By continually monitoring the applied axial forces during emplacement and during retrieval of the apparatus, this risk can be substantially reduced.

The lateral jacking pressure and radial displacement measurements are incorporated into the design for assessment of the quality of the emplacement. A particular emplacement might be unsuccessfull for one of two reasons:

(1) Lack of control or continuity in the operation of the equipment.

(2) Attempting to emplace the probe in irregular or inhomogeneous soils.

The data produced by the BSD probe will be characteristic of a relatively small volume of soil and it is essential that the material surrounding the probe is uniform and representative of the overall soil mass. By measuring the initial stess/displacement characteristics of the segments of the coupling cylinder surface, irregularities in the surrounding soil can readily be detected. It is possible, in addition, that a useful geotechnical evaluation of the soil can be attained from the data produced by these measurements.

b. The Torque/Normal Load Measurement The torque normal load transducer is located in the upper probe section. The device utilizes a thin walled gauge tube stressed in simple shear to sense applied torsion, and in tension/compression to sense axial or normal load. Elastic strain induced by these loads is sensed by strain gauges bonded to the inside of the gauge tube. A complete bridge configuration is recommended for each measurement resulting in a minimum of eight gauges on the inside of the tube. The orientation of the gauges is dependant on the strain required to be sensed, and it is important that the array of gauges are properly connected in the bridge circuit. A detailed treatment showing the configuration of the gauges in the transducer and giving an analysis of the transducer constants for the system is given in Appendix A. With a conventional bridge arrangement it is shown that for a given gauge tube geometry a certain transducer sensitivity results. This sensitivity cannot be increased by using a greater number of active gauges in the system since the bridge configuration would average the outputs of any additional strain gauges. However, this effect can be used to improve the overall tranducer characteristics, particularly in regard of the susceptibility of the device to misaligned loads and its ability to cope with unwanted bending, shear, or compression/tension loads. By using an eight gauge bridge configuration, an improvement on the basic four gauge system will result. To achieve such a system, the four gauge configuration is duplicated in a manner that achieves regular symmetry, and the gauges (responding in the same manner) are connected in series in the bridge circuit. In view of the dual application of the gauge element of the torque/normal load transducer, it is recommended that an eight gauge configuration be used for both measurements.

c. The Rotation Measurement This is perhaps the most critical measurement in the system. The rotation of the cylinder, as shown in the first phase design study, is directly related to the maximum shear strain in the surrounding soil, and the relationship is such that the angular rotations are of similar magnitude. (In Section III of the Phase I report it is shown that the peak shear strain; ie., the strain in the soils at the coupling cylinder/soil interface, is numerically equal to twice the angular rotation of the cylinder in radians.) In the domain of elastic response of the soil, the rotational movements to be measured are small. For the measurement system to be able to resolve these movements the datum or zero drift of the transducer, expressed as an equivalent angle, must be held well below the smallest angle to be measured. In general, the limits to the unwanted component of a transducer output (ie., that not due to an input) is expressed as a fraction of the full range output of the device. These outputs can be random noise, temperature induced offsets, and the effects of nonlinearity and hysterysis in the transducer. It is important to distinguish between these effects because some of them can, to a limited extent, be overcome to improve the measurement resolution, and others cannot. For example, random noise superimposed on a transducer output results in instantaneous short-term errors in the measured value. If the process being measured produces a signal which is stationary long enough to average or filter the transducer output, the effects of the noise can be reduced in proportion to the filter time constant. With a hysterytic characteristic on the other hand, resolution cannot be had below the intrinsic hysterysis level and averaging, while it may superficially improve the appearance of the signal, will not contribute to the accuracy or resolution of the measurement.

The temperature environment for the measurement system within the BSD probe is stable and it is likely that the major sources of error in this regard will arise from self-heating within the transducer under the effect of energisation. Thus, providing these effects are comparable for a given choice of devices, the crucial factors affecting the choice of rotation transducer appear to be those of sensitivity and hysterysis. As mentioned earlier on in Section II-5c the transducer type chosen for the basic rotation measurement (ie., the static relative rotation between the self-boring head and the coupling cylinder) is an LVDT type. The LVDT transducer has a number of features which make it suited to this measurement:

(1) The devices are noncontacting. Mechanical linkages are not required and operating forces are not encountered. Hence, there are no mechanical sources of frictional or hysterytic behaviour.

(2) The devices feature virtually infinite electrical resolution.

(3) Being of simple and robust construction, the characteristics are extremely stable.

(4) The sensitivity or transducer gain is inherently high.

(5) In their basic form the units are ac energised and produce an ac output. It is thus a relatively simple matter to apply external gain to enhance the sensitivity of the system.

The particular transducer chosen for the static rotation measurement is a low displacement range high sensitivity unit. LVDTs are generally rated for sensitivity in percentage units of output with energisation for a unit displacement input (i.e., millivolts output per volt of excititation per millimeter input). This parameter is found to vary quite widely from unit to unit, depending somewhat on the application, i.e., large bore clearance units, short stroke to transducer length units etc., it is also found to be dependent on the full-scale displacement. It is, perhaps surprisingly, not optimum for the smallest stroke transducers, there being a definite range of full-scale strokes within which the parameter peaks. The transducer chosen for the static rotation measurement is optimum in this respect, and the overall stroke available ($\pm$1.3 mm), since it is not required to be as great, is effectively a bonus to the system.

As described earlier, the transducer senses rotation as a linear displacement at the end of a radius arm, and so to calculate the overall system sensitivity, both the length of the radius arm and the LVDT constant are required. For the system described here, these values are 60.0 mm and 248 mV/V/mm respectively, resulting in a measurement constant of 14.88 V/V/rad. Given a nominal energisation voltage of 5 V rms the overall system constant is thus $\pm$74.4 V rms/rad.

In order to express a change of sign in the measured variable the LVDT output windings are usually connected in such a way as to produce an overall output which changes in both magnitude and phase through the zero value. In this respect, the differential output of the LVDT resembles that of an ac energised bridge system (the difference being the generally higher output levels produced by the LVDT). It was mentioned earlier that ac signals have the advantage that considerable gain can be applied. This is true because of the significant difference in the inherently stable gain characteristics of a correctly designed ac amplifier compared to the problems in achieving high and stable dc gain. Furthermore, ac signals are immune to such effects as thermal electromotive forces (emfs) and have considerable potential for improvement in dynamic range of measurement if synchronous demodulation or lock-on amplification is considered. With this method the signal is chopped under the control of a reference signal source. This reference is derived from the same excitation source that feeds the transducer and, hence, the chopper operates only at the frequency of the desired signal component in the transducer output. The chopper has the effect of rectifying only this synchronous component in the input signal. All other components remaining sinusoidal and, hence, of zero average value. By averaging or filtering the chopper output the ac components are rejected and the synchronous component emerges as a steady dc level. For noise with spectral components close to the sychronous frequency the chopper produces slow beats and, hence, long averaging times are required to reject them from the output signal.

Using synchronous demodulation, it is possible to retrieve and measure signals in conditions where the signal-to-noise ratio is 60 dB (where the noise level exceeds the signal level by a factor of 1000). Hence, providing the measurement system is an ac carrier type, there is scope for extending the dynamic range considerably by the use of lock-on amplification. As an example, assume a lock-on amplifier unit to have a full-scale sensitivity of 10 μV (a typical value for a good quality unit). If the amplifier affords 1 percent resolution at this gain, the resolution of the LVDT based rotation measurement system (coupling the transducer output directly to the amplifier) is an impressive 1.3×10^{-9} rad. It must be remembered that with synchronous demodulation, the phase of the reference or chopper driving frequency is important. The system produces a dc output which is directly proportional to the mean of the modulus of the coherent signal input, times the cosine of the phase angle between it and the reference. With LVDTs there is often a considerable phase shift between primary and secondary voltages and this must be taken into account when using lock-on amplifiers to detect their output. Figure 29 shows a possible configuration of the rotation measurement system.

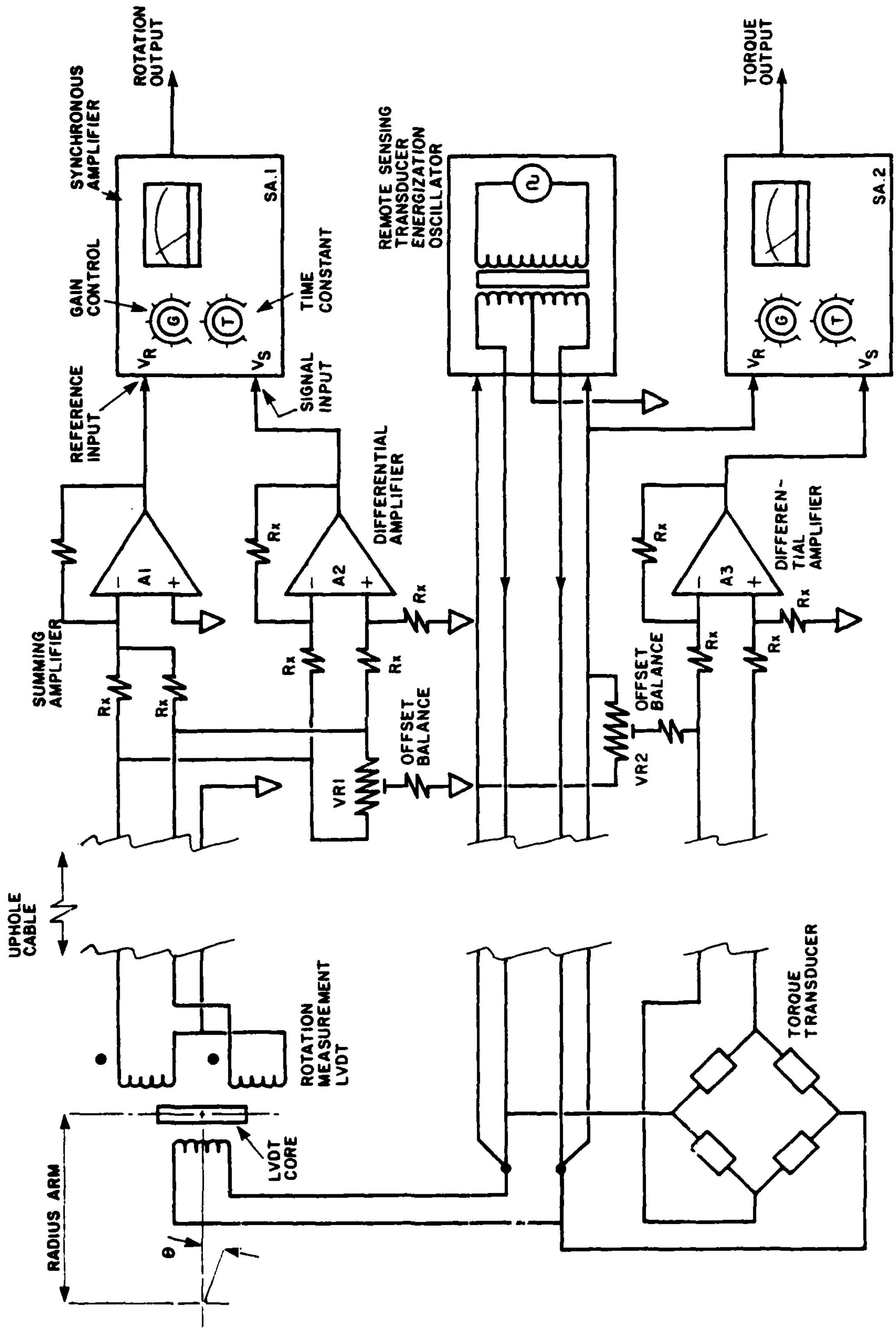


Figure 29. Schematic of the LVDT based (static) rotation measurement system.

Here the problem of phase shift in the LVDT is overcome by taking the reference frequency signal directly from the secondary winding via the summing amplifier A1. The differential signal output of the windings is converted to a single ended output by the differential amplifier A2. Both the reference and differential signal amplifiers provide balanced loading of the LVDT preserving the electrical symmetry of the system. This is an important consideration since it is partly due to inherent electrical and mechanical symmetry that the LVDT achieves its superior performance.

In order to resolve extremely small signals, the gain of the measurement system must be high. This in turn means that to achieve a high dynamic range of measurement, the mean value about which the small measured variations occur, must be at or near zero. Any residual or net offset in the output from the measurement transducer will limit the amount of useful gain that can be applied. In the design of the rotation measurement system described in Section II-5, it is mechanically locked in the zero position during emplacement of the probe. Thus, with the transducer set so that its electrical zero position corresponds to the mechanically locked positon, the entire working range of the transducer is available for the subsequent measurement and the output of the transducer at the start of testing is sensibly near zero. Since the locking mechanism cannot be expected to be perfectly repeatable there will be some zero imbalance, and to resolve small anglular movements, an electrical zeroing system which can be manually adjusted will be required. As shown in Figure 29, this is achieved by a shunt load across the secondary windings of the LVDT. The relative degree of loading of the secondary windings is controlled by a multiturn potentiometer which is the zero adjustment control, and the coarseness of the control (or the resolution of zero attainable) is determined by the ratio of the shunt resistances applied to the LVDT secondary windings.

With this arrangement zeroing range is directly traded for zeroing resolution as the value of shunt resistance is reduced.

There is inevitably a limit to the dynamic range attainable from any measurement system. For the LVDT/synchronous amplifier configuration described above the limit is likely to be brought about by shifts or drifting of the electrical zero. The electrical zeroing system must, as shown in Figure 29, be implemented at the point where the ac signals from the transducer are accessible. This means that the ac output leads from the transducer must be incorporated into the uphole cable telemetry system. In this situation there is the possibility that coupling can occur between the LVDT lead wires and that at low levels unpredictable offsets or shifts could be created. This state of affairs will depend greatly upon the manner in which the lead wires are routed uphole, but it is almost certain that the nonconstant offsets that do occur due to this mechanism will be difficult to compensate for and will vary as a result of a number of influences. Other factors that could introduce drift into the zero of the rotation measurement are:

(1) Self-heating of the LVDT due to the energisation power absorbed by the device.

(2) Temperature induced variations in the ratio of the offset potentiometer and the absolute value of the shunt resistance.

By the use of synchronous detection enhanced signal conditioning about an electrically zeroed LVDT transducer output, a resolution of 1 part in 10^6 should be achievable for the rotation measurement. For the system described in this report the full-scale angular range of the LVDT rotation measurement is 0.021 rad (1.21 deg), giving a system capable of resolving to shear strains of about $\gamma = 4 \times 10^{-8}$ absolute. This implies a good measurement accuracy for strains in the elastic range $\gamma < 10^{-4}$ absolute.

For measurement of linear soil response in the low strain domain an alternative technique to the LVDT system described above exists, which can offer freedom from the effects of zero offsets. This is the use of inertial sensing of the rotational movement of the coupling cylinder. If, instead of a displacement transducer, a velocity or acceleration sensing device is used, registration of static displacements will not occur and the gain of the measurement system is correspondingly not limited by this factor. Inertial sensing of the rotational movement does require dynamic (steady state oscillatory) torsional drive of the BSD probe, but this provision exists in the dynamic actuator unit which is part of the equipment developed under this second phase contract. The inertial transducer would be a linear sensing device mounted, as in the case of the LVDT unit, to detect tangential movement at the end of a radius arm. The unit would not however, require coupling to a reference datum, but would register velocity or acceleration about a mean reference point. The output from the torque and rotation transducers would be sinusoidal functions and, since a reference frequency exists (i.e., the frequency applied to the dynamic actuator drive system), both functions can be synchronously amplified to yield high measurement resolutions. A schematic of an inertial measurement system based around an acceleration sensing transducer is shown in Figure 30. Here a charge amplifier is used to condition the output of the accelerometer and a lock-on amplifier, functioning as in Figure 29 as a high gain carrier amplifier/demodulator unit, conditions the output of the ac energised torque transducer. The two steady state signals at the frequency of drive of the dynamic actuator are then processed by the synchronous amplifiers SA3, SA4.

For a steady state rotation θ of the coupling cylinder:

$$(t) = \theta_o \sin 2\pi ft$$

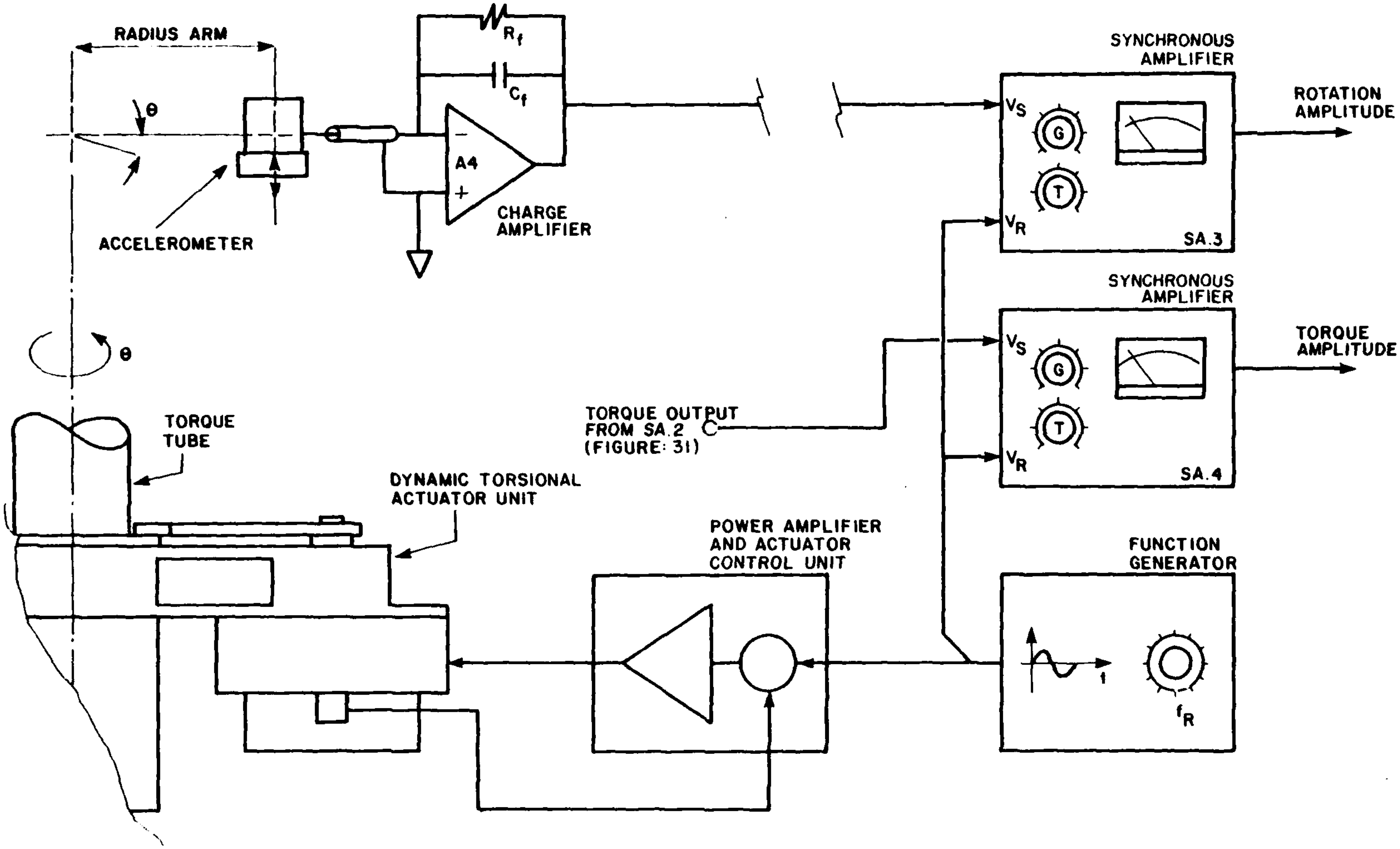


Figure 30. Schematic of the inertial rotation measurement system.

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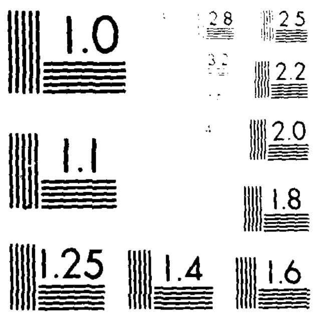
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2.8
2.5
3.2
2.2
2.0
1.1
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1.6

where f = torsional excitation frequency (hertz)

t = time

θ_o = peak angular excursion

The tangential acceleration x(t) at the end of the radius are R is then:

$$\ddot{x}(t) = -4\pi^2 f^2 R\theta_o \sin 2\pi ft$$

The corresponding output e_o from the accelerometer/amplifier system is then:

$$e_o(t) = -4\pi^2 f^2 RK_1K_2\theta(t)$$

where K_1 is the accelerometer sensitivity ($pC/m/s^2$), and K_2 is the charge amplifier gain (V/pC). The synchronous demodulator detects the coherent component of the applied input signal and converts it to a voltage proportional to the mean of the modulus of the waveform. For a sinusoidal function this value is equal to twice the peak value divided by π. Hence the rectified dc output $\bar{e}_o$ of the lock-on amlifier system shown in Figure 30 is:

$$\bar{e}_o = 8\pi f^2 K_1 K_2 R\theta_o.$$

In order to assess the order of sensitivity attainable from such a system, assume the following nominal system parameters:

(1) Accelerometer sensitivity K_1 = 25 $pC/m/s^2$

(2) Charge amplifier gain K_2 = 0.004 V/pC (corresponding to a feedback capacitance of 250 pF)

(3) Radius arm length R = 60 mm

which, for a torsional excitation frequency of 20 Hz, gives a sensitivity of:

$$\frac{\bar{e}_o}{\bar{\theta}_o} = 60.3 \text{ V dc/rad pk.}$$

If, as for the case with the LVDT system, a full-scale sensitivity of 10 μV and a nominal resolution of 1 percent is assumed for the synchronous amplifier, the inertial system is found to be potentially capable of resolving angular movements down to 1.7×10^{-9} rad (a figure close to that calculated for the LVDT based system). However, it would be unlikely that this sensitivity would be achieved in practice since ambient seismic noise at the site would probably induce linear accelerations into the measurement system exceeding the values equivalent to the angular resolution quoted above*. It is worth mentioning here that this effect would likely be considerably less noticeable with the relative (static) rotation system, since the seismic noise is unlikely to introduce a significant amount of the type of motion to which to which the system is sensitive. In summary, the advantages of the inertial system lie in the fact that low strain testing can be conducted with freedom from the effects of offsets in the system, and furthermore that the output of the system is particularly easy to handle being a steady dc level corresponding to amplitude for both the torque and rotation measurements.

It should be noted that the use of th^ synchronous detection system is different for the absolute and relative rotation sensing systems. With the absolute rotation sensing or inertial system, the reference frequency is the torsional rate at which the BSD probe is driven. The output of the torque and rotation synchronous amplifiers will therefore be a steady dc level corresponding to the mean of the modulus of the functions. Only amplitude and relative phase can be deduced from this measurement. Hence, it is suitable

*This affect could be reduced to a certain extent by using a pair of transducers mounted diametrically opposite on two radius arms, and arranged so that their outputs were summed. With this arrangement the tangential movements due to rotation would add, but the net linear displacements would tend to cancel.

only for low strain elastic response measurement where nonlinearities are not present or are insignificantly small.

The above discussions outline two basic rotation measurement systems suitable for use with the BSD probe. Possessing different and complementary characteristics, the systems will allow use of the probe over a very broad range of strains ranging from elastic response to yield. The LVDT based measurement system is the most useful being suitable for use with both static and dynamic torsional drive to the BSD probe. It is suggested that as a first step in instrumenting the probe, this would be the most versatile system and therefore should be installed and tested before the inertial system. Both measurement systems can be accommodated within the lower section of the probe and can be operated simultaneously during testing.

d. <u>Instrumentation of the Radial Expansion Function</u> As described earlier in Section II-4 measurement of the radial displacement of the moving keys of the coupling cylinder assembly is achieved using a system of LVDT transducers. These transducers, housed within sealed chambers in the upper and lower fixed key units, sense displacement via a spring loaded plunger which acts on a heel extension of the moving keys. The particular devices chosen for the application are miniature units, offering a good stroke to body length ratio. As shown in Figure 17, the core of the transducer is mounted within a special carrier component (C6, Fig. 17) which positions the core approximately midway through the electrical stroke when the moving keys are midway through their mechanical displacement. With the moving keys fully re[illegible], the LVDTs are positioned at approximately 80 percent of their electrical s[illegible]oke in the core's inward direction. The rated nominal sensitivity of the units is 120 mV/V/mm at 2.5 kHz. This gives a total output voltage change (progressing through electrical zero) from the basic unamplified transducer with 5 V rms energization, of 1.50 V rms for the full 2.5 mm stroke of the moving keys. Using a conventional high quality carrier amplifier/demodulator unit, resolutions approaching 2.5 x 10^{-4} mm should be attainable with relative ease from this system.

For the fully instrumented probe there are 24 radial displacement measurements resulting in a total of 120 conductors (5 per LVDT -- 2 for the primary winding and 3 for the center-tapped secondary) leading into the services manifold compartment. It is recommended that all transducers in the BSD probe (with the exception of the inertial rotation measurement device), both the strain gauge bridge type and the LVDT units, be energized from a common source capable of supplying a nominal 5 V rms at 2.5 kHz. With the relatively large number of transducers in the BSD probe, the downhole lead loss of the conductors carrying the energizing potential will be appreciable and will therefore require compensation. To achieve this, it is recommended that a feedback controlled energization oscillator (shown in Figure 29) be used. Here the energization potential at the end of the downhole cable run (at the distribution point in the services manifold), is sensed and fed back to the oscillator unit. A control device within the oscillator adjusts the output to maintain the remotely sensed potential constant. In order to reduce the number of signal carrying cables uphole, it is recommended that an electronic multiplexing telemetry unit be incorporated into the services manifold. This is discussed in detail later in this section.

While not specified as a contract requirement, a number of radial displacement LVDTs have been installed in the BSD probe. This was done primarily to assist limited laboratory testing conducted at Kings College, London, but has also been of value to the project as a whole in validating the proposed design for this part of the measurement system. Six transducers are installed to instrument each end of three moving keys which are symetrically displaced around the probe (positions 3, 7, and 11)*. The units are installed within the coupling cylinder section of the probe to full operational specification, and are consequently ready for use. Only temporary termination of the transducer lead wires in the services manifold compartment, however, has been affected. The lead wires are attached to terminal pads which are bonded to the central auger tube above the point in the manifold base plate where the cables enter. Depending on the amount of instrumentation to be included in the subsequent field trails this arrangement may require subsequent modification.

A miniature semiconductor strain gauge pressure transducer is proposed (Kulite Semiconductor Products Model XTMS-1-190-2000) to sense radial jacking pressures. The unit, rated to 13.8 MPa (2,000 lb/in^2) full-scale, is a miniature metal diaphragm type suitable for working with standard hydraulic fluids. As described in Section II, the unit is located within the services manifold assembly mounted on a special transducer block.

e. Uphole Telemetry The following is a list of the measurement devices in the fully instrumented BSD probe:

(1) Torque - one 350 Ω strain gauge bridge

(2) Normal load - one 350 Ω strain gauge bridge

(3) Jacking pressure - one 650 Ω strain gauge bridge

(4) Rotation measurement, static - one LVDT, primary impedance 430 Ω

(5) Rotation measurement, inertial - one piezoelectric accelerometer, (self-generating signal source)

(6) Moving key displacement - 24 LVDTs, primary impedance 175 Ω

* These positions are identified by engravings on the probe.

It is recommended that a common ac excitation source be used for the strain gauge bridges and LVTD transducers. Because of the relatively large number of transducers involved, the excitation power required will be significant, and a means of compensating for the downhole cable potential drop will be required. As shown in Figure 29, a feedback controlled remote sensing transducer energization oscillator is used to sense the potential at the downhole distribution point with a separate pair of conductors. These conductors carry very little current and, hence, negligible loss occurs in them. The action of the feedback loop is to control the output of the oscillator in such a way that the remotely sensed potential remains constant. At 2.5 kHz, the total excitation load impedance of the transducers [(1) through (6) above] is calculated to be 4.8 Ω at a power factor of 0.6. For an energizing supply of 5 V rms, this requires a downhole cable current of 1.04 A, and a power delivery capability for the excitation oscillator a little in excess of 5 W. Remote sensing excitation oscillators of the type discussed here are readily commercially available. The units are frequently found in multichannel installations where a large number of transducers require energizing. They are usually incorporated into modular systems which are designed to accommodate a variety of signal conditioning units.

The need for a multiplexing telemetry unit in the BSD probe has been discussed elsewhere in this report. There are a relatively large number of transducers in the system, the outputs of some of which need not be recorded or observed simultaneously, and many of which produce data which are only slowly varying. Additionally, there is the problem of space in accommodating the uphole cables in the services conduit, and the electrical connectors in the services manifold compartment. These factors, despite the increased system complexity and the relatively inconvenient geometry necessitated by the layout of the services manifold, strongly suggest that the overall system would benefit from the incorporation of a signal multiplexing system. The following summarizes of the potential advantages of such a system:

(1) The number of uphole conductors required for the fully instrumented BSD probe can be reduced from 64 to 25, resulting in considerably more manageable cable and connector assemblies.

(2) The multiplexing unit reduces the number of parallel signal outputs and, hence, the number of channels required in the data recording system. The unit automatically provides channel code information which can be recorded along with the transducer data to identify the signal sources.

(3) With a reduced number of uphole signal lines, it becomes feasible to buffer the multiplexed data lines before feeding the cable. This gives the system tolerance to various cable types and length of runs. It also assists in producing a less noise or interference prone telemetry link.

The schematic of a system designed to achieve the above and, therefore, recommended for incorporation into the BSD probe is given in Figure 31. The signals from the strain gauge transducers and the rotation measurement systems are routed directly into the uphole cable link. The signals from the moving key displacement transducers, however, are all fed to the array of analogue switching units (AS1 to AS6, Fig. 31). There are a number of reasons for not multiplexing the torque and rotation measurements:

(1) The maximum possible dynamic range is required from these measurements. When using electronic analogue switches, a certain amount of interchannel mixing or crosstalk occurs. Although of insufficient

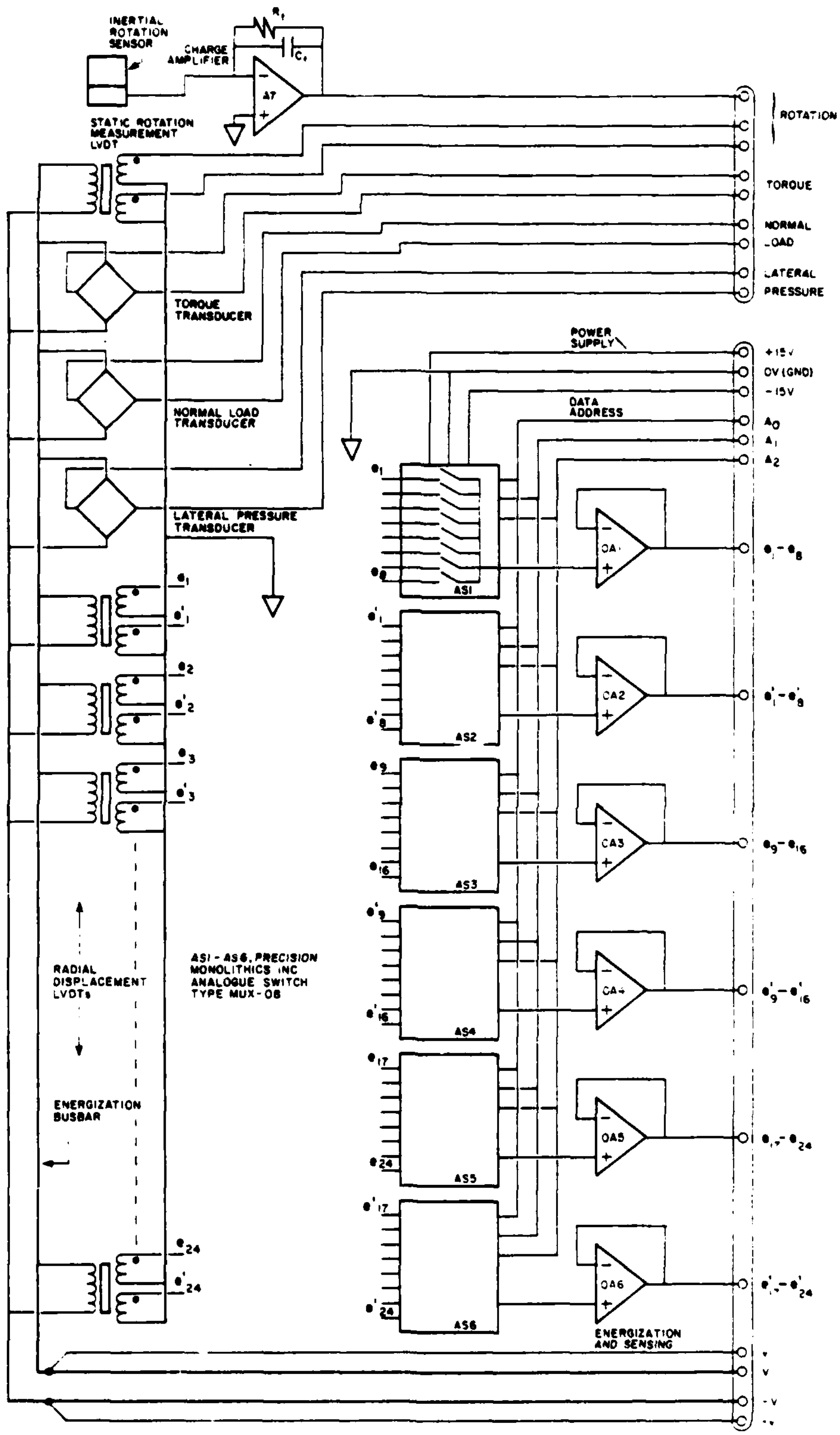

Figure 31. Schematic of the multiplexer and telemetry unit.

magnitude to disturb the less important signals, there will be a limit of dynamic range imposed on the data. (For the recommended switching devices, crosstalk and off-channel isolation are specified as better than 100 dB or 1 part in 10^5 at the carrier frequency of the measurement system.)

(2) As discussed in Section II-7c, there is the requirement to manually control fine zeroing the torque and static rotation measurement channels. This is implemented as shown in Figure 29 by precision multiturn potentiometers and shunt resistances connected directly to the transducer leadwires. This passive type of zero circuitry is essential for a stable system at high gains. It follows that, unless the zero potentiometers are located downhole and remotely servo driven, the channels requiring this type of zeroing or offsetting cannot be multiplexed. There is little benefit to be had from the servo zero system other than a possible further reduction in the number of uphole conductors. This would, however, be only a small reduction and would be gained at the cost of a relatively complex and costly system.

Thus the outputs from the rotation measurement systems and the strain gauge bridge transducers are fed directly uphole. It is recommended that these signals, as shown in Figure 31, be routed in a separate uphole cable to the other main system cable. The signals in the main cable will be buffered and many of them will be of high level (the radial displacement LVDTs are in general not operating close to electrical zero), meaning that they are relatively immune from interference from the transducer excitation supply. This is not the case for the torque and rotation measurements.

The multiplexer system is comprised of an array of six analogue switch units (Precision Monolithics Inc. eight-channel analogue multiplexers type MUX-08). These units are electrically equivalent to an 8-way selector switch, the particular channel being selected (or the switch position being determined) by a 3-bit binary code which is applied to a set of digital inputs. The 8-way switch units are connected to the axial displacement LVDT transducers in such a way that they act in three groups of paired units to select three differential transducer outputs for a given digital control code. In this way a pair of multiplexer units serve eight transducers, reducing the signal outputs to a single differential pair. The full complement of 24 radial LVDTs is, correspondingly, handled by three pairs of multiplexers resulting in three differential pair outputs. The multiplexer outputs are fed into unity gain follower amplifiers. These units serve to buffer the multiplexer and transducer outputs to provide a low impedance cable drive and to eliminate the effects of variation of resistance on the analogue switch units. Thus, as the 3-bit binary control word advances through its eight states (000 to 111), the 24 radial LVDT channels are selected in groups of three and appear as buffered differential ac signals at the outputs of amplifiers (OA1 to OA6, Fig. 31). Uphole, these outputs can be fed directly into conventional LVDT signal conditioning (demodulator) units. Control of the multiplexing function can be achieved in a number of ways:

(1) By an asynchronous (free running) clock: here an 8-state counter is driven by a clock and the resultant data are recorded continuously along with the counter states.

(2) By synchronous control where an external data logging system advances the counter states.

(3) By manual step-through control.

The measurement systems described in this section have been designed with the data recording problem in mind. For the fully instrumented BSD probe equipped with the multiplexing telemetry unit, there are nine channels to be recorded (eight data channels plus one multiplexer address code). However, depending on the nature of the testing to be undertaken, it is unlikely that all data channels will need to be recorded simultaneously. The modes of operation of the BSD probe are potentially diverse, ranging from inertially driven dynamic operation to a high torque quasi-static mode of testing. The effect of this range of usage of data on the acquisition and recording system is significant. Maximum flexibility has been the aim in designing the measurement systems for the probe. However, this philosophy cannot realistically be persued with the data acquisition and recording system. The cost of a generalised system capable of encompassing the full complement of recording channels, the highest data rate, and the maximum resolution required would be prohibitive. It could, therefore, hardly be justified at such an early stage in the development of the BSD probe as a geotechnical testing tool. The following section was added to assist in selecting a suitable data acquisition and recording system. The basic modes of testing are set out, and their specific characteristics and requirements are discussed from the data acquisition and recording viewpoint.

f. Data Acquisition and Recording In order to explore the requirements of the data acquisition and recording system, two basic modes of operation of the BSD apparatus are considered:

(1) A low-strain cyclic, high-frequency excitation phase in the 20 to 50 Hz range. Here, the dynamic inertial actuator described previously

is used to provide steady-state torsional excitation to the apparatus at a number or range of predetermined torque amplitudes. The soil is not excited to the yield point in this phase of the test, but is excited at strains corresponding to elastic deformation. Torque and rotation signals are derived from instrumentation within the probe (using the strain gauge torque/normal load transducer and the inertial rotation measurement system).

(2) A high-strain, quasi-static loading test. During this phase, a ground-anchored or statically restrained actuator is employed to load the BSD up to full rated torque to yield the surrounding soils. The actuator produces cyclic or monotonic loading histories, applies torque to the coupling cylinder, and records the resultant rotation. However, the inertial rotation measurement system would not be suitable for this type of actuator and the static LVDT based system would be used.

For the low-strain dynamic excitation phase, the degree of resolution required of the entire instrumentation and data recording system will depend on the requirements of the data analysis. If the sole purpose of the low-strain testing is to determine an effective modulus for the in situ soils at a number of strains, then only one amplitude parameter (e.g., mean-modulus, rms, peak, etc.) need be determined for each recorded variable at each excitation level. Under these circumstances it would be wasteful of data storage-capacity and inconvenient from the point of view of subsequent data processing, to record the variables with time as simple periodic functions. It would be more efficient to preprocess the signals to derive static or averaged amplitude parameters, digitize these results with the required resolution, and record them together with the excitation frequency and any other relevant test information. In this way, the amount of data storage capacity required is considerably reduced. This is one of the prime reasons for suggesting that synchronous detection amplifiers be used as shown in Figure 30. The preprocessing method also offers the advantages of low data acquisition rates. The averaged parameter is stable for the duration for which excitation is applied to the system, meaning that high resolution measurement or digitization can be conducted at low speeds. This both substantially reduces the cost and broadens the choice of commercially available systems.

If on the other hand, a more complex assessment of the soil response than an effective modulus is required, the data acquisition and recording system must be capable of retrieving unprocessed test data. It must then operate at a considerably higher speed. Consider, for example, the problem of assessing damping properties. This requires the ability to retrieve a description of the hysterytic behaviour of the soil. At low strains, hysterysis is small and is evident from small differences in the paths of the load response as the direction of loading changes. To quantify these small differences from the test data, a high resolution of measurement is required. Clearly, a certain minimum number of data points are required to define the stress/strain characteristic of the soil, and given this additional parameter, the total data rate (i.e., data word length times measurement rate) can be determined. If there is a need to use the dynamic actuator in determining the more complex soil behaviour, the high loading speed of the device and the need for high resolution recording, requires that a relatively high performance data acquisition system be used. If, in addition, the change in soil response under sustained loading, or over a number of load cycles is required, a proportionately larger data storage capacity would be required. Given the large capacity and high data rate it would be difficult to provide temporary or buffer storage for the incoming data. In this extreme case the mass storage system (probably a high performance instrumentation tape recorder) would have to operate at data acquisition speed.

The data acquisition requirements imposed by the quasi-static actuator, in contrast to the dynamic device, are straightforward. It is inherent in this phase of the test that the soil is loaded up to failure and that a detailed description of the nonlinear stress/strain characteristic is required. Signal recovery or averaging techniques cannot be employed since the loading or response functions are, in general, not periodic or repetitive. Thus, for a given sensitivity of the measurement systems, the more resolution attainable from the data acquisition system, the greater the dynamic range of strain over which the characteristic of the soil is measured. The torque and rotation signals for the quasi-static phase of the test would be derived from the strain gauge torque/normal load and LVDT transducers. The LVDT transducer measures rotation between the coupling cylinder and the self-boring head of the BSD probe. The maximim resolution attainable from these transducers depends upon the type of detector or signal conditioning systems used. If the transducer signals are amplified and demodulated by conventional high quality signal conditioning units, the likely dynamic range attainable will be about four orders of magnitude. However, if as shown in Figure 29, synchronous detectors operating at the transducer excitation frequency are used, resolution can be improved beyond this figure. The improved dynamic range of the system is achieved by gain ranging the synchronous amplifiers (either manually or by using autoranging units) and, therefore, some method of inputting gain information along with the test data would be required. Apart from the considerations of data acquisition discussed later, the rate at which the test is conducted in the quasi-static mode is limited by a number of factors:

(1) The response time of a synchronous amplifier (or any noise rejection system) to a change in the input variable is finite. The rate of change of the input function must be limited so that the amplifier is allowed to settle to the desired resolution.

(2) Any digitized process will be affected by the rate of change of the analogue data. For a conventional analogue to digital converter this is determined by the front end sample/hold system. If the input is constant to a degree approaching the system resolution for the sampling time of the converter, the measurement will not be degraded.

Of particular importance to the BSD test are the effects of the sampling rate and resolution on the integrity of the data. The object of the test is the study of the stress/strain characteristics of the soil. The conventional materials testing approach of directly plotting these variables one against the other has certain inherent advantages over a system which utilizes a data logging or digital data acquisition system. If the values of the stress/strain functions are measured sequentially (as in multiplexed input data acquisition systems), a time displacement between the measured parameters is introduced. As the loading rate, or speed of testing increases, this effect becomes more apparent. If the time displaced readings are interpreted as simultaneous coordinate points on the stress/strain curve, the effect is a distortion of the characteristic. This is illustrated in Figure 32 and below for a function of negligible hysterysis.

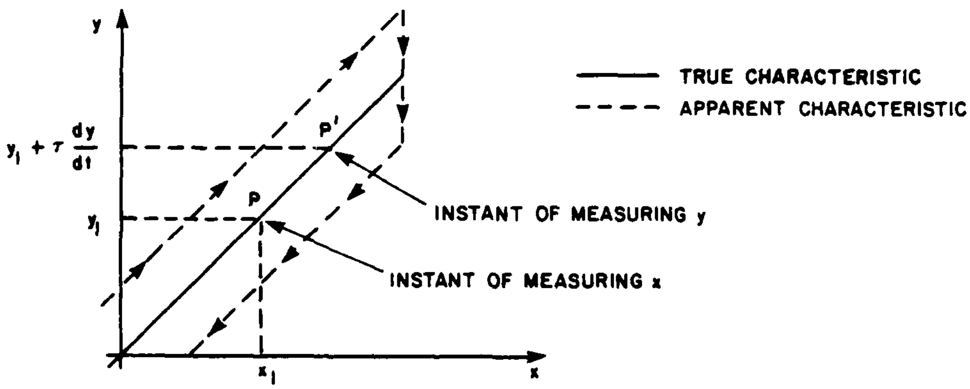

Figure 32. Sequential sampling of data points on a stress/strain curve.

Let the focal point of the curve be moving at a constant velocity, and let the values of x and y be read x first, followed by y after delay τ. The apparent position of the point if replotted using the staggered data is:

$$P' = P'(x_1,\ y_1 + \tau \frac{dy}{dt})$$

The error term $\tau \frac{dy}{dt}$ is seen to depend linearly on the interval between samples and the time derivative of the delayed variable. If the curve is traced in the reverse direction the error term changes sign and an apparently hysterytic function results. As an example, let the straight-line function in Figure 32 represent the load/deflection characteristic of an elastic body. Let the body be subjected to a linear strainrate cyclic loading function between the values plus and minus x_1, whereby the velocity of the focal point of the loading curve between the stationary points is constant. If the period of the complete loading cycle is T seconds, the velocity of the vertical component is given by:

$$\frac{dy}{dt} = \pm \frac{4y_1}{T}$$

and, hence, the error term in the y coordinate value due to the sampling interval is $\pm \frac{4y_1\tau}{T}$. The strain energy stored in the body at deflection x_1 is $\frac{y_1 x_1}{2}$. Upon returning to zero load from deflection x_1 the apparently hysterytic function will show an energy dissipation of $\frac{8y_1 x_1 \tau}{T}$. Thus, for this load half-cycle the ratio of apparent energy dissipated to maximum strain energy stored, or the apparent damping ratio, is given by:

$$\tau = \frac{16\tau}{T}$$

For a continuous sequential sampling process, N samples/cycle of loading at interval seconds,

$$N = \frac{T}{\tau}$$

and the apparent damping introduced into the otherwise entirely elastic load response is;

$$\zeta = \frac{16}{N}$$

The above analysis is elementary. It does show, however, that for low values of hysterysis, the already difficult problem of measurement due to the need for high resolution is directly aggravated by sequential sampling in the data acquisition system. For instance, in order to avoid introducing an apparent hysterysis of greater than 1 percent, 1600 sequentially sampled stress/strain data points per cycle of loading are required; that is, unless the system can be modified to sample adjacent channels rapidly in comparison to the scan interval.

There are a number of solutions to this problem:

(1) Sample sequentially at a rate high enough to make the time difference negligible.

(2) Sample and digitize the data from the two channels simultaneously with parallel and synchronized input channels.

(3) Knowing the data acquisition system sampling rate, (this would have to be recorded with the data), the true coordinate points of the stress/strain curve can be determined from subsequent processing of the data. An algorithm derived from the above analysis could be used.

Option (1) requires a fast input multiplexer and analogue-to-digital converter. To reduce the error introduced by sequential sampling to an acceptable level, it would seem reasonable to make the sampling interval equal to the loading cycle time multiplied by the resolution of the readings. For a 1 Hz cyclic load and two data channels of 12-bit (0.025 percent) resolution each, the interval between reading data points should be no greater than 0.25 ms. This does not imply a continuous data word rate of 4 kHz since this would produce an excessive number of data points. The number of coordinate points required by the analysis to define the stress/strain characteristics may vary, but would almost certainly be less than that produced by continuous sampling at the maximum allowable channel-spacing interval. Thus, with this arrangement only a certain portion of the data stream would be recorded.

In option (2), with parallel sampling, each data channel has a dedicated analogue to digital converter. The converters are controlled so that they operate synchronously and the data stagger problem is eliminated. The converters operate at a speed determined by the number of coordinate points required per load cycle, and the loading rate. The conversion speed is additionally reduced by the fact that more than one converter is used. The digitized data so produced can be multiplexed and interleaved so that they can be recorded serially in exactly the same manner as for the sequential system discussed earlier. Despite the increased amount of hardware, the parallel input system need not be excessively costly compared to the sequential one. Assisting here would be the fact that the lower speed converters would be less expensive and, furthermore, that digital multiplexing would be an easier hardware problem to deal with than the switching of analogue signals.

Option (3) uses sequentially digitized readings and eliminates the stagger interval by postprocessing of the data. As previously stated, the data acquisition system scanning rate would have to be recorded along with the data from the measurement transducers. The data processing algorithm would generate the error term by calculating the rate of change of the delayed parameter and would modify the data accordingly. The degree of correction attainable by this process depends upon the complexity of the data and on the amplitude and time resolution of the data stream. It would certainly be possible to generate staggered data from plots of realistic stress/strain behaviour in order to test the correction routines and to determine their accuracy.

Finally, in this discussion of data acquisition, it is worth considering the problem of data processing and manipulation. Only a very limited amount of high resolution data can be analyzed or processed manually. If the purpose of the data acquisition system is to retrieve and store any significant bulk of data, then the compatibility of the data with a computer processing facility is an important consideration. Most dedicated or portable data acquisition systems employ minimal formatting of the data in order to optimise the use of storage space and speed of recording. Such systems often record data directly onto the storage medium in binary or binary-coded-decimal form with primitive codes to identify channel numbers, file blocks, time readings, etc. While this is desirable from many standpoints, problems can arise with subsequent manipulation and processing of the data. Often the manufacturers of the smaller data acquisition systems do not provide more than what is essentially a data replay system. These units typically produce an analogue output for chart recording or a printed list of the contents of the data store. In such a form, few computer installations will accept the data directly, and interfacing can be a complex task. It is worthwhile considering these matters in some detail before deciding on which acquisition system to chose for the BSD test.

III THE ACTUATOR SYSTEMS

1. GENERAL DISCUSSION

In the Phase I BSD study, the matter of drive power required by the system was considered. It was found that while the power required by the in situ coupling cylinder was relatively small, the power absorbed (or more correctly, stored and exchanged) by the elastic twist of torque tube string could be appreciable and could, therefore, affect significantly the design of the actuator system for the apparatus. Added to these considerations are the effects of frequency of operation of the system. At low frequencies, i.e., where the inertia of the system is not appreciable, the drive power requirements are determined primarily by the torsional stiffnesses of the in situ probe and the torque tube string. Neglecting hysteretic or inelastic soil behaviour, mechanical energy is stored almost totally as strain energy in the system. At a given amplitude of torsional drive, the power exchanged between the source and the system is directly proportional to frequency of operation of the actuator. As drive frequency increases, however, a point is reached at which the inertia of the system (this comprising both the rotational inertia of the downhole system and of the actuator system itself) dominates. This effect increases with the square of the operating frequency and acts to absorb the drive torque developed by the actuator in accelerating and decelerating the rotational inertia in the system. The result is that angular drive output becomes limited and, correspondingly, the useful torque (i.e., the reaction torque due to twist in the torque tube) is reduced. It can be appreciated from these considerations that the need to operate the BSD apparatus at high drive torques conflicts with that of using the device at high operating frequencies.

For high torque operation the actuator is highly stressed and is, therefore, necessarily bulky. Furthermore, the range of angular drive required is relatively large. Both of these features, as mentioned above, tend to reduce the upper frequency of operation at which maximum torque can be developed.

The above matters were given detailed consideration during the initial period of the Phase II development contract, the problem being to produce a system capable of providing the high maximum design torques required by the apparatus to investigate inelastic soil behaviour, yet to satisfy a contract requirement to produce a system capable of operating at frequences in the range of 20 to 50 Hz. Ultimately, Dames & Moore proposed that two actuators should be considered:

a. A quasi-static high torque actuator for monotonic or low frequency cyclic loading up to the full design torque.

b. A dynamic unit employing inertial reaction to provide high frequency capability but at the cost of reduced maximum operating torque.

The high torque or static actuator as the name implies is a low frequency device. Being statically restrained by a deadweight/friction system or by the use of ground anchors the unit employs an arrangement of hydraulic motors and a worm and wheel reduction gear to drive the torque tube. The static actuator can provide controlled drive (either controlled torque or angular displacement) up to the full BSD design torque and features unlimited output rotation. Practically any type of loading history can be developed by the device provided that a certain limiting angular velocity (corresponding to an upper bound imposed by the maximum flow rate in the hydraulic supply) is not required.

The dynamic actuator, in contrast to the high torque device, is not statically restrained, but relies on inertial reaction to allow it to develop an output torque. This reaction is provided by the structure of the actuator itself and, hence, the unit requires mechanical attachment only to the torque tube string. Being derived purely from inertial reaction, the torque developed by the system is of zero average value. The system can only sustain a momentary torque in a given direction and it achieves this by accelerating a seismic or self-inertia in an opposite direction to the angular drive output. For a given type of mechanical power source or prime mover, the effectiveness of such an actuator is determined by:

a. The amount of self-inertia it possesses in comparison to the load.

b. The combination of both the self-inertia and the operating frequency in determining the effective drive stiffness of the unit.

The fact that the reaction structure is dispensed with is a significant advantage of the system. For the scale of the BSD apparatus and for the range of operating frequencies required, it was felt that problems might be experienced in the design of a resonance free and mechanically stable structure for such an application. During the course of the preliminary study of the dynamic actuator, a number of various design concepts for the device were considered. The study progressively narrowed until two basic concepts were isolated as being potentially capable of leading to a workable system:

a. A system where the prime mover or mechanical power source was continuously rotating and used to maintain a source of rotational kinetic energy in a flywheel. Oscillatory torques would be produced by a special linkage system between the torque tube and the rotational power source, and operating frequency would be directly proportional to the speed of rotation of the source.

b. A system employing a fast responding power source capable of directly producing the required oscillatory or reciprocating motion. Being fast responding, the prime mover would have a relatively low force or torque output and, hence, some form of low inertia gearing would be required.

An example of a concept that was investigated for the continuously rotating prime mover actuator is shown schematically in Figure 33. Mechanical power is provided in the system by a pair of electric motors and a system of balanced flywheels is employed to ensure a relatively steady rotational speed against the accelerating and decelerating affect of the oscillatory torque output. Crank pins on the rotational drives are coupled via special hydraulic connecting rods to radius arms which are in turn connected to the torque tube. Being formed from double acting hydraulic cylinders the connecting rods are of variable length. If the two halves of the cylinders, are interconnected by a flow control valve, the reaction force exerted by them in response to the oscillatory input drive from the crank pin can be controlled. In the arrangement of Figure 33, alternate chambers of the opposing cylinders are directly interconnected to ensure equal system pressures and hence a balanced torque couple. The flow control valve shunts the two circuits and, hence, allows bypass flow directly between the opposing chambers. With the valve closed off, the pressures developed in the cylinders create opposing forces and the torque developed by the system (or the angular excursion of the radius arm plate) is a maximum. As the valve is opended, the bypass flow

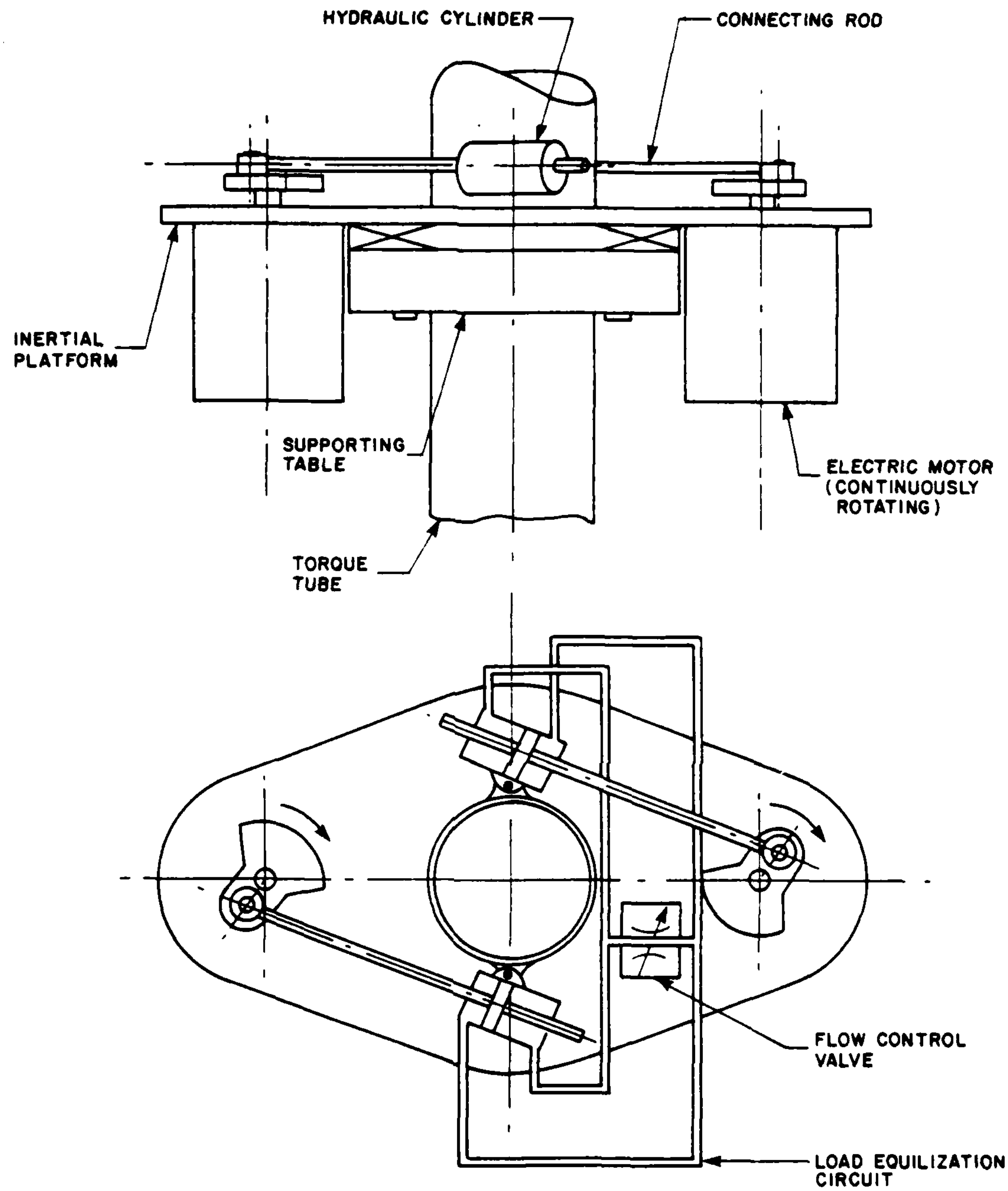

Figure 33. Schematic of an early design concept for the dynamic actuator.

reduces the effective stiffness of the connecting rods and the output torque is reduced. However, detailed study of a practical implementation of this system revealed that for the higher operation frequencies the flow velocities in the hydraulic circuit would be particularly high. This would result in a limited control output range for the system and, perhaps worse, an inability to control the output down to low levels. In addition to these problems, it was realized that the shafts of the drive motors would be required to rotate with exact synchronization. No convenient or easy means to solve either of these problems was found and it was decided to abandon the concept. Similar experiences with various other configurations of the rotating prime mover concept eventually resulted in the final design of the actuator being based around the second basic principal above, that of an oscillatory power source. To achieve the required high frequency performance without inordinate penalties in power consumption or production and development costs, a compact design using high power electromagnetic servomotors was decided on. While he use of these devices affords the system a number of desirable operational characterisitcs, it is inherent that the basic actuator cannot develop torques approaching the maximum design specification for the BSD. The actual output of the dynamic actuator in a given test configuration will depend on a number of factors:

a. The stiffer both torque tube string (i.e., the shorter the length of the string) and the in situ probe, the smaller is the angular excursion required of the drive system given torque. This will result in higher torques since, as mentioned earlier, the amplitude of angular excursion that the actuator can produce becomes limited at the higher frequency by inertial effects.

b. The self-inertias of the BSD probe, the torque tubes, and the actuator will serve to modify or influence the resultant drive characteristics or the response of the dynamically actuated system. These effects will be strongly frequency dependant and may result in resonant behaviour. This may serve to reduce or enhance the drive torque applied to the probe.

Details of the final design for the dynamic actuator and the associated drive and control system are discussed in the next section of this report. Following that, a section describing the design for the static actuator appears. Assembly and component detail drawings for the dynamic and static actuators are available on request from AFWL/NTES.

2. THE DYNAMIC ACTUATOR

a. General Description It was mentioned in the previous section that the final design chosen for the dynamic actuator was based around a high power electromagnetic servomotor drive arrangement. The actual units used in the design are dc servomotors which, being of the ironless disc rotor type, offer high performance and fast response. A number of advantages arise from the use of this type of electromagnetic actuator:

(1) Using disc rotor motors results in an inherently low inertia at the crucial point in the system (before the reduction gearing).

(2) Series operation of the units can be employed to balance drive current and, hence, help to produce a pure torque couple.

(3) Servocontrol to achieve static positioning of the system (see Section III-2f) is easily achieved.

(4) The system can be driven from standard electronic power amplifiers.

(5) The inherently high linearity of the motors and the driver amplifiers allow the system to produce a relatively pure, harmonic free drive output. This can only benefit the measurement systems associated with the testing in which the actuator is used (particularly the inertially sensed rotation measurement described elsewhere in this report).

A schematic diagram illustrating the principal of operation and showing the basic layout of the final design of the dynamic actuator is given in Figure 34. The unit employs two high-power dc servomotors displaced symmetrically about the central torque tube and mounted on a platform or structure. This structure is suspended on the torque tube via bearings and expanding friction couplings. The couplings effectively lock a central tube onto the torque tube and bearings, while transmitting dead load from the structure of the actuator, allows the structure complete rotational freedom about the locked tube. The servomotors drive the torque tube via an eccentric pin and connecting rod, and gearing or mechanical advantage is achieved by the three arm linkage so formed. The connecting rods are attached at the torque tube end to a central tube (termed the inner sleeve) which is attached to the torque tube by a friction locking coupling. This coupling is used to transmit the axial deadload and the torsional drive load developed by the system. Reaction for the applied drive torque is provided by the self-inertia of the overall structure of the actuator.

Without any form of static attachment or restraint, the mean position of the structure relative to the torque tube and, hence, the alignment of the three arm linkage system, is not maintained. The gearing ratio provided by the linkages (being the ratio of the effective torque tube radius arm length to the

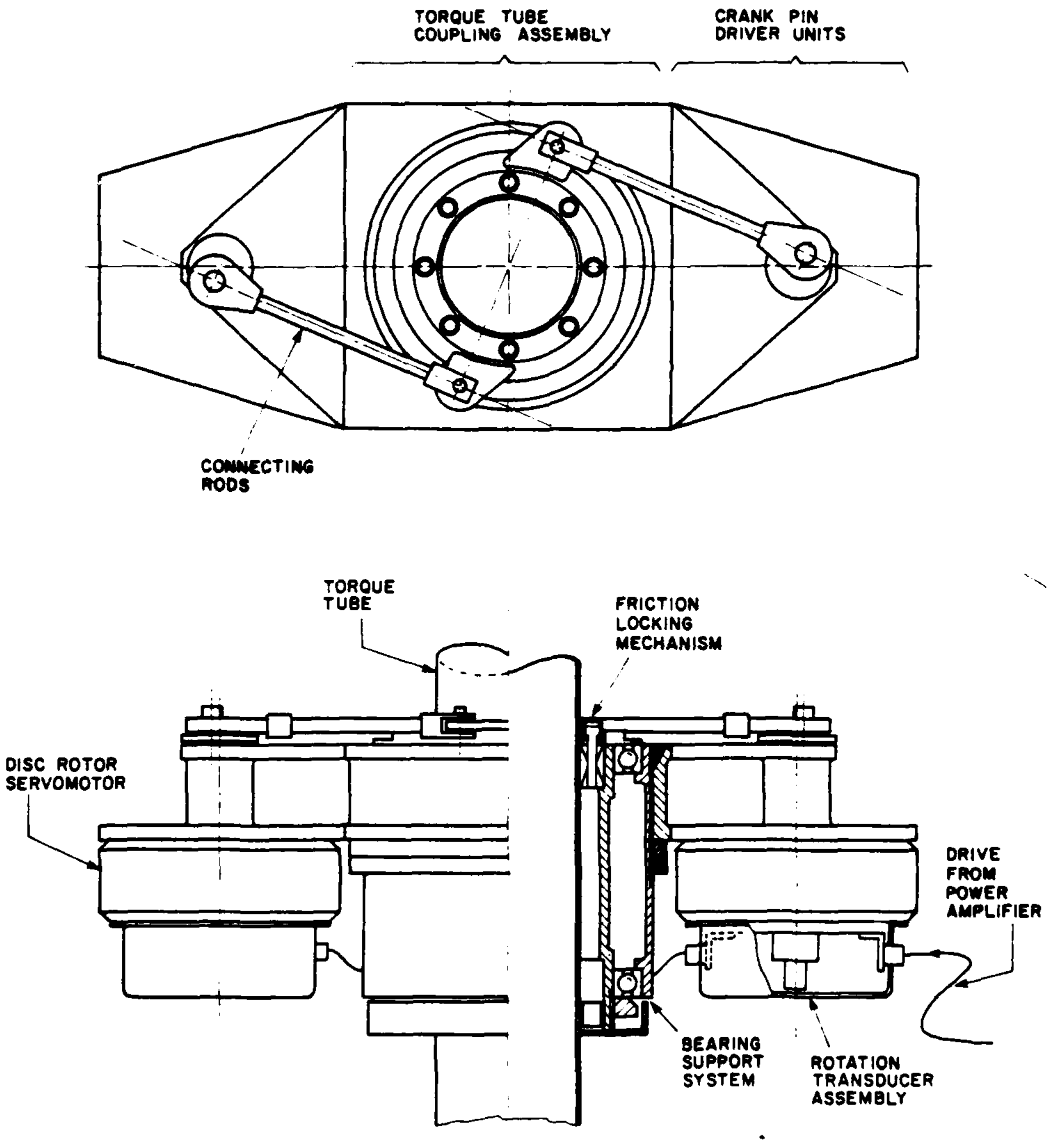

Figure 34. Schematic of the final design for the dynamic actuator.

radial offset of the crank pins) is maximum when the radius arms are at right angles to the axis of the connecting rods (see Fig. 35). This position is not self-maintained by the oscillatory loading in the actuator and, hence, some form of static positioning system for the linkages is required. As shown in Figure 36 this is achieved by a low frequency or slow acting control system which senses the angular position of one of the drive motor shafts (and, hence, the position of the crank pin) and compares it with a fixed reference. The control loop feeds the difference between the shaft position signal and the reference (i.e., the error signal) via a low pass filter into a summer unit where it is mixed with the oscillatory torque command signal for the drive amplifier. The effect of this arrangement is that the drive amplifier, being dc coupled, is made to handle a composite input where the fluctuating component is the desired or preselected dynamic drive control signal, and a slowly varying dc component (which causes net rotation of the servomotors) which is the static position control signal for the linkage arrangement. The function of the low pass filter is to prevent the position control system from attempting to correct for the high frequency changes in the linkage positions. Without the filter the position control loop would interpret the oscillatory drive of the motors as instantaneous position errors and would attempt to cancel the input.

b. The Torque Tube Coupling Assembly The torque tube coupling assembly is the name given to the section of the dynamic actuator structure which houses the friction locking mechanism and the system of rotation isolating bearings. As shown in Figure 37 frictional coupling to the torque tube, both to support the deadload of the actuator and to transmit the oscillatory torque output of the unit, is achieved by a pair of wedge type couplings (FA and FB, Fig. 37).

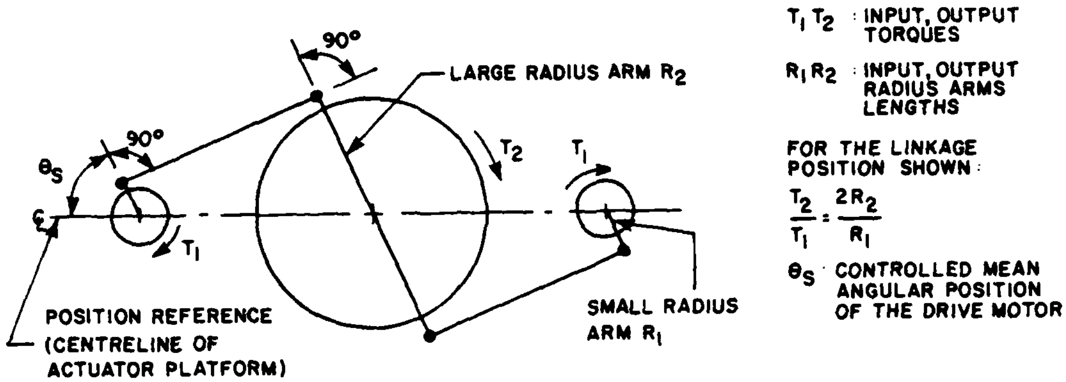

Figure 35. Schematic of the three arm linkage of the dynamic actuator.

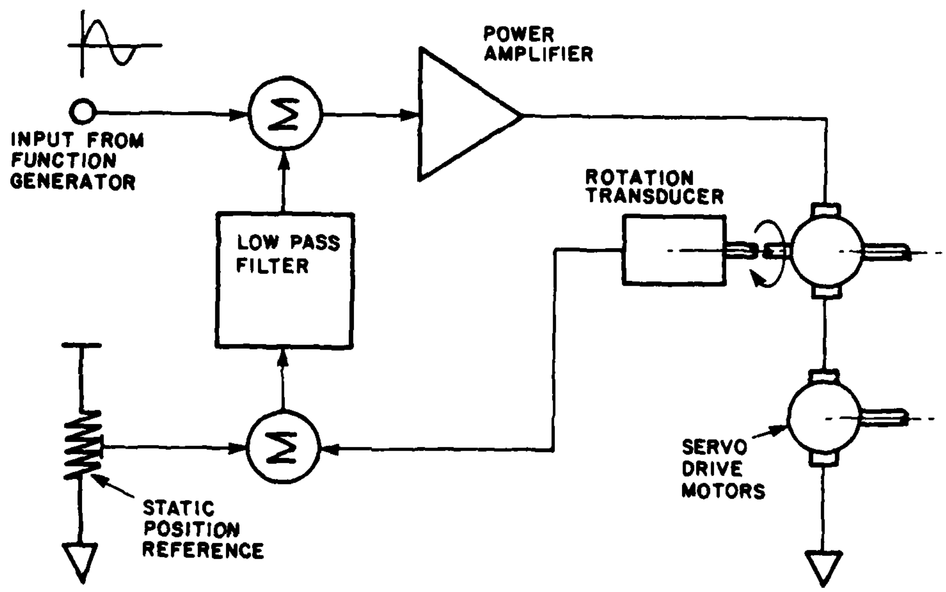

Figure 36. Schematic of the position control and power drive system for the dynamic actuator.

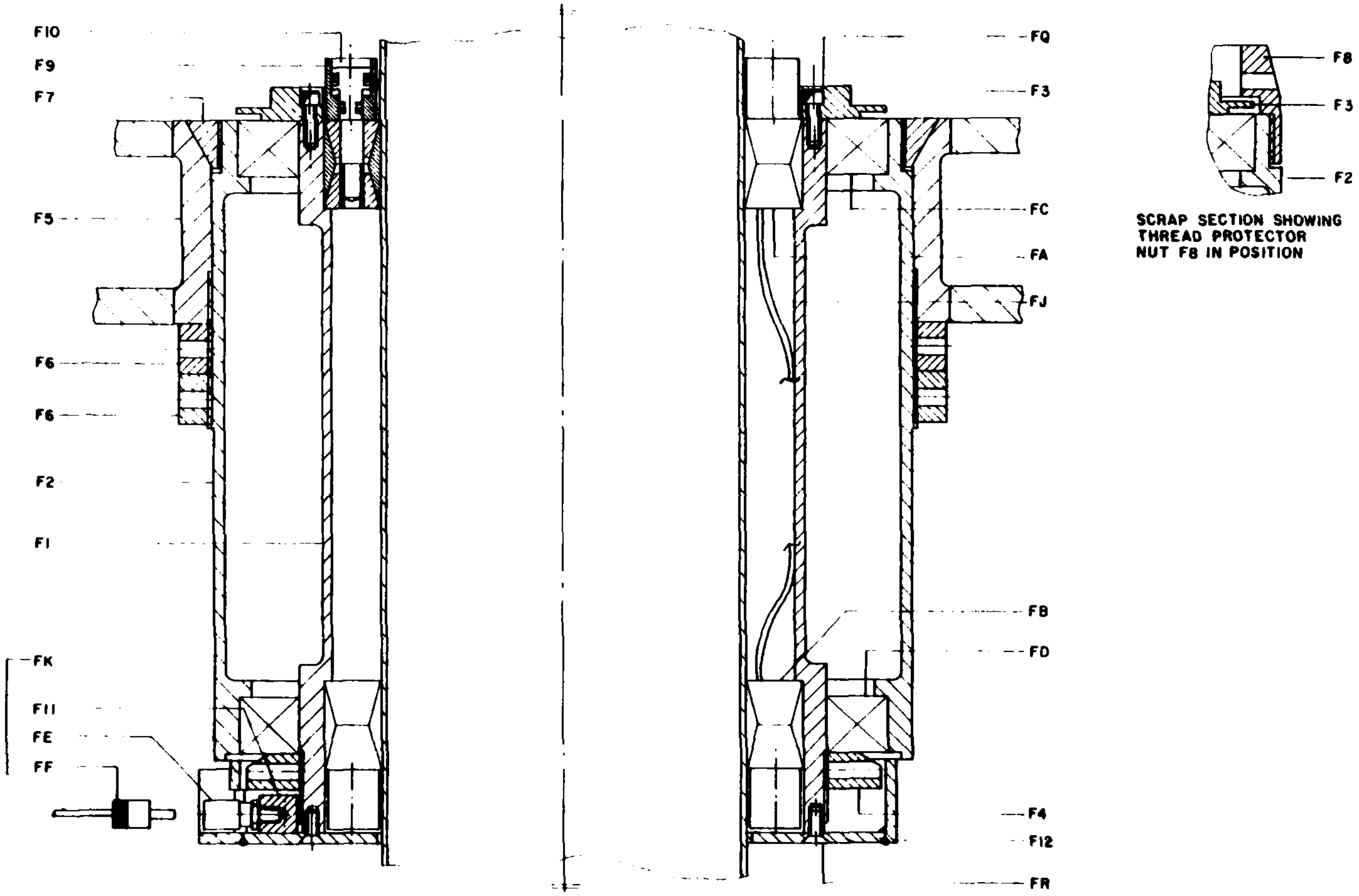

Figure 37. Assembly of the torque tube coupling system - section.

KEY TO FIGURE 37

ITEM	QUANTITY	DESCRIPTION
F1	1	Inner Bearing Sleeve.
F2	1	Outer Bearing Sleeve.
F3	1	Bearing Retaining Plate.
F4	1	Bearing Retaining Nut.
F5	1	Hub Housing.
F6	2	Lock Nut.
F7	1	Taper Nut.
F8	1	Thread Protector Nut.
F9	16	Hydraulic Actuator Cylinder.
F10	16	Hydraulic Actuator Piston.
F11	1	Hydraulic Port Block.
F12	1	Bottom Cover.
FA	1	Expanding Friction Coupling (upper locking mechanism), 180 x 235, type RFN 7012. Supplier - Ringfeder Limited, Forum Drive, Midland Industrial Estate, Rugby, Warwickshire CV21 1NT, UK.
FB	1	Expanding Friction Coupling (lower locking mechanism), see FA.
FC	1	Deep Groove Ball Bearing (upper support bearing), 260 x 320 x 28. Supplier - SKF ref 61852.
FD	1	Deep Groove Ball Bearing (lower support bearing), see FC.
FE	1	Hydraulic Quick-Connect Fitting (manifold bulkhead connection), 1/16 NPT to quick-connect female type "Swagelok" QM2-B-1PM. Supplier - Crawford Fitting Company, 29500 Solon Road, Solon, Ohio 44139, USA.

KEY TO FIGURE 37 (Continued)

ITEM	QUANTITY	DESCRIPTION
FF	1	Hydraulic Quick-Connect Tubing Fitting (supply tube to manifold block connector), 1/16 compression to male quick-connect type "Swagelok" QM2-S-100. Supplier - see FE.
FJ	-	Specification 316 Seamless Stainless Steel Pipe (hydraulic interlinking tube), see BH.
FK	-	Specification 316 Seamless Stainless Steel Pipe (hydraulic supply pipe), see BH.
FQ	12	Socket Cap Screw (bearing retaining plate/inner sleeve fastening) M6 x 12 HTS.
FR	6	Socket Countersunk Head Screw (bottom cover retaining) M6 x 12 HTS.

Based around a modified component known as a Ringfeder coupling, the arrangement is hydraulically actuated by an array of pulling cylinders (see Fig. 38). The friction couplings act to drive wedges between a pair of split rings, and in this way, lock both ends of the inner bearing sleeve (F1, Fig. 37) of the assembly onto the torque tube. The Ringfeder unit is normally used with an array of draw bolts to drive the wedges. However, for the dynamic actuator application this method was rejected and the system of hydraulic pulling cylinders substituted for the bolts. The hydraulically actuated system was preferred for a number of reasons:

(1) An inherently uniform distribution of load in the coupling is obtained by using an array of hydraulic actuators connected to a common pressure source.

(2) The controlled force characteristic of the actuators prevents damaging crushing loads from being applied to the relatively thin walled torque tube.

(3) The time required to attach the actuator is considerably reduced.

(4) Site personnel are not required to access fastenings underneath the assembly, hence, there is no inherent risk of accident or injury in the event of failure or slipping of the friction locking mechanism.

The pulling actuators are mounted in an array on the outer wedge rings of the upper and lower friction locking assemblies. There are eight cylinders per coupling and the units are interconnected by microbore high pressure piping. The hydraulic circuit for the pulling cylinders terminates in a special manifold block F11 which is mounted on the bottom cover (F12, Fig. 37) of the torque tube coupling assembly. The manifold block is fitted with a

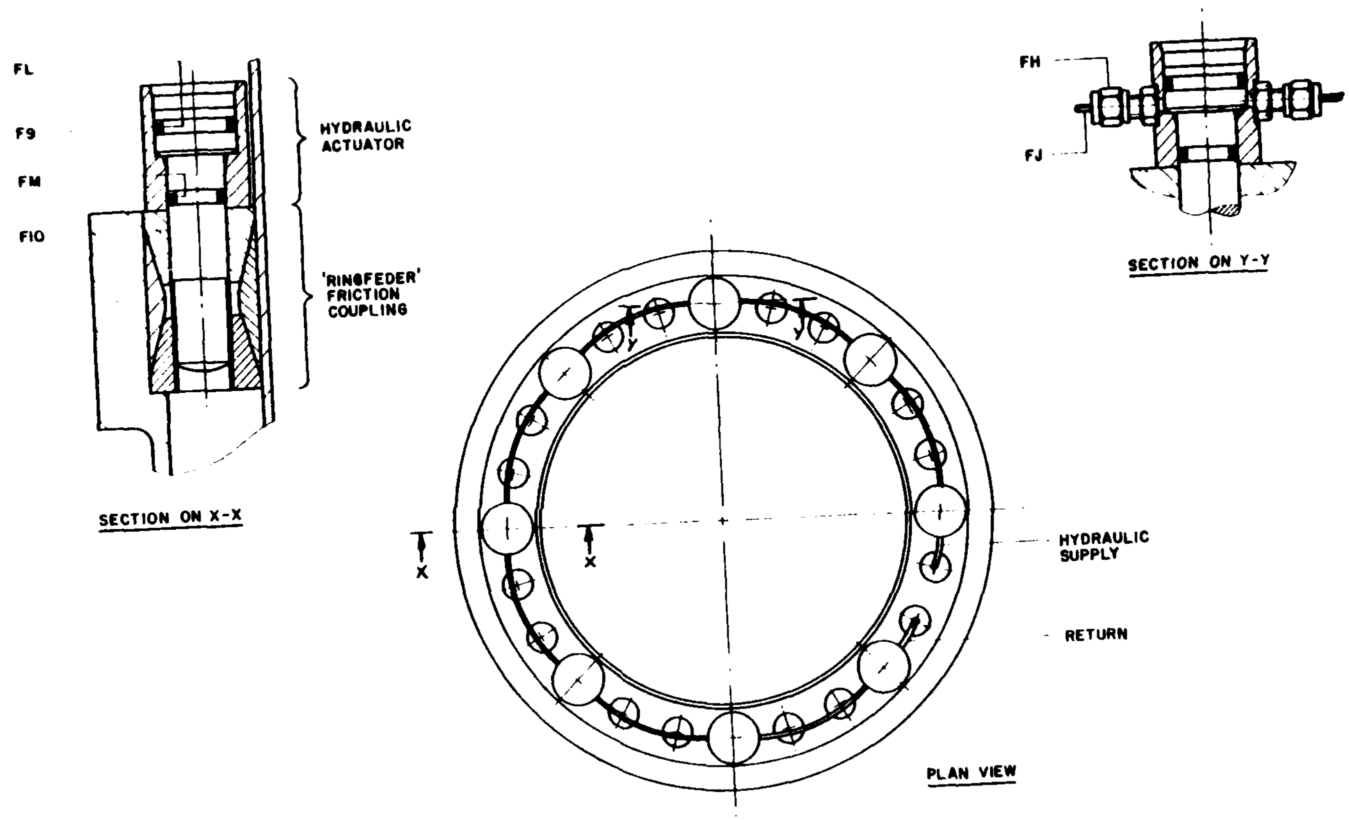

Figure 38. Assembly of the hydraulically actuated friction coupling.

KEY TO FIGURE 38

ITEM	QUANTITY	DESCRIPTION
F9	16	Hydraulic Actuator Cylinder.
F10	16	Hydraulic Actuator Piston.
FH	32	Hydraulic Compression Fitting to Pipe Adaptor (interconnecting fittings for hydraulic actuators) 1/16 compression fitting to 1/8 tube, type IRU. Supplier - Hoke Incorporated, One Tenakill Park, Cresskill, N.J. 07626, USA.
FJ	-	Specification 316 Seamless Stainless Steel Pipe (hydraulic interlinking tube), 1.63 mm (1/16 in) OD x 0.4 mm (0.016 in) wall. Supplier - Tubesales (UK) Limited, West Bay Road, Southampton SO9 5HQ, UK.
FL	16	O Ring plus PTFE Spiral Back-up Ring (hydraulic locking actuator piston head seal).
FM	16	O Ring plus PTFE Spiral Back-up Ring (hydraulic locking actuator piston rod seal).

quick release hydraulic coupling FE for attachment of a high pressure hand pump, and the end of the hydraulic loop is accessible via a standard compression fitting FG for bleeding the assembly.

Rotational freedom for the inner sleeve component with respect to the overall structure of the actuator is provided by the system of upper and lower support bearings (FC and FD, Fig. 37). These components carry the deadload of the actuator and any lateral forces that may arise from out of balance thrusts in the connecting rods. The bearings chosen for this application are deep groove ball types which are well suited to working with combined axial and lateral load. Being available with very slender aspect ratios (ratio of sectional dimension to mean diameter) the units afford a compact design. Preload for the bearings is provided by the retaining nut F4 and is distributed between the two units via tension in the inner bearing sleeve F1 and compressive load in the outer sleeve (F2, Fig. 37). The upper bearing is retained against the preload forces by the plate F3 which is fastened to the inner sleeve. This plate is arranged to overhang the bearing to provide a degree of protection against entry of dust or grit into the assembly.

To attach the drive servomotor assemblies onto the outer sleeve of the torque tube coupling assembly, a special hub housing component F5 is used. This component is threaded onto the outer sleeve and is square in profile to allow attachment of the drive units. The system of lock nuts F6 serve to provide a bottoming face for vertical positioning of the housing, and a special wedge nut threaded onto the end of the outer sleeve is screwed into a cone seating in the upper face. The wedge nut was incorporated into the design as a result of a need for a high degree of overall stiffness within the actuator structure. A significant portion of the loading of the structure is oscillatory and it occurs over a relatively broad range of frequencies; there are correspondingly many vibrational modes in which the structure can respond. If resonance occurs at one of these modes then potentially damaging effects such as fretting, fatigue, or wear could result. To avoid such problems the philosophy regarding the structure or chassis of the actuator was to produce a design of optimum stiffness. Here, the resonant modes could be shifted to the high frequencies where the actuator output would be reduced, and the liklihood of damaging deflection occurring would be small. However, conflicting with the need for an inherently stiff structure, was the requirement for a modular system which could be conveniently assembled or dismantled in the field. As shown in the Figure 37, this is achieved by locating the hub unit over the outer sleeve in a plain bearing or bush arrangement, and securing the assembly by means of a threaded section at the bottom of the sleeve. While this scheme provided a convenient coupling or joint in the system, it suffered from the drawback that the clearances required by the plain sleeve section did little to contribute to the fixity and, hence, stiffness of the joint. The wedge nut is incorporated into the design to compensate for this affect by providing positive lateral restraint for the top of the hub housing against the outer sleeve. Being threaded and locked at the bottom face and wedged at the top the hub is thus firmly secured and, from a vibration standpoint, can be regarded virtually as an integral part of the outer sleeve unit. To allow space for the connecting rods to lie close to the structure, the wedge nut is finished flush with the top of the outer sleeve and the hub housing. Fitting or removal of the nut is accomplished using the specially designed pin spanner F18. To protect the threaded end of the outer sleeve component when the hub housing is not in position a thread protector nut F8 is provided. This nut is finished with the same outside diameter as the plain bearing section of the outer sleeve and, hence, need not be removed prior to fitting the hub. The nut is provided with a lead-in to ease location during assembly and to avoid the possibility of damage to the internal thread in the hub housing. It is recommended that this nut be fitted immediately following removal of the wedge nut.

c. <u>The Crank Pin Drive Units</u> The crank pin drive units are the assemblies which provide mounting for the driver motors and which house a system of support bearings to absorb the thrust loads developed in the connecting rods. The general arrangement of the units is shown in Figure 34 where it can be seen that the crank pin is driven at one end only, i.e., it is loaded in single shear. As shown in Figures 39 and 40, forces in the connecting rod are transferred via support bearings into the body G1 of the assembly. This unit in turn couples these forces back through the structure of the actuator via its points of attachment to the hub housing component F5 of the torque tube coupling system. To yield maximum structural stiffness, the body is machined from solid and is attached to the hub housing by an upper and lower system of face compression joints. The butting faces are located by dowel pins and are retained by a novel through bolt fastening. As shown in Figure 41 the through bolts are set into the flange plates of the bodies by means of an elongated slot into which the bolt hole breaks. Compressive stresses from the bolt are distributed smoothly into the flanges by special saddle washers (F13, Fig. 41). The compression face joint so formed is compact and efficient. Provided the vibratory structural loads do not lead to stresses approaching the static face compression, the scheme results in a stiff and low damping fixture.

The drive motors are bolted to the lower flange of the bodies and are precisely located on their mounting spigots by machined recesses in

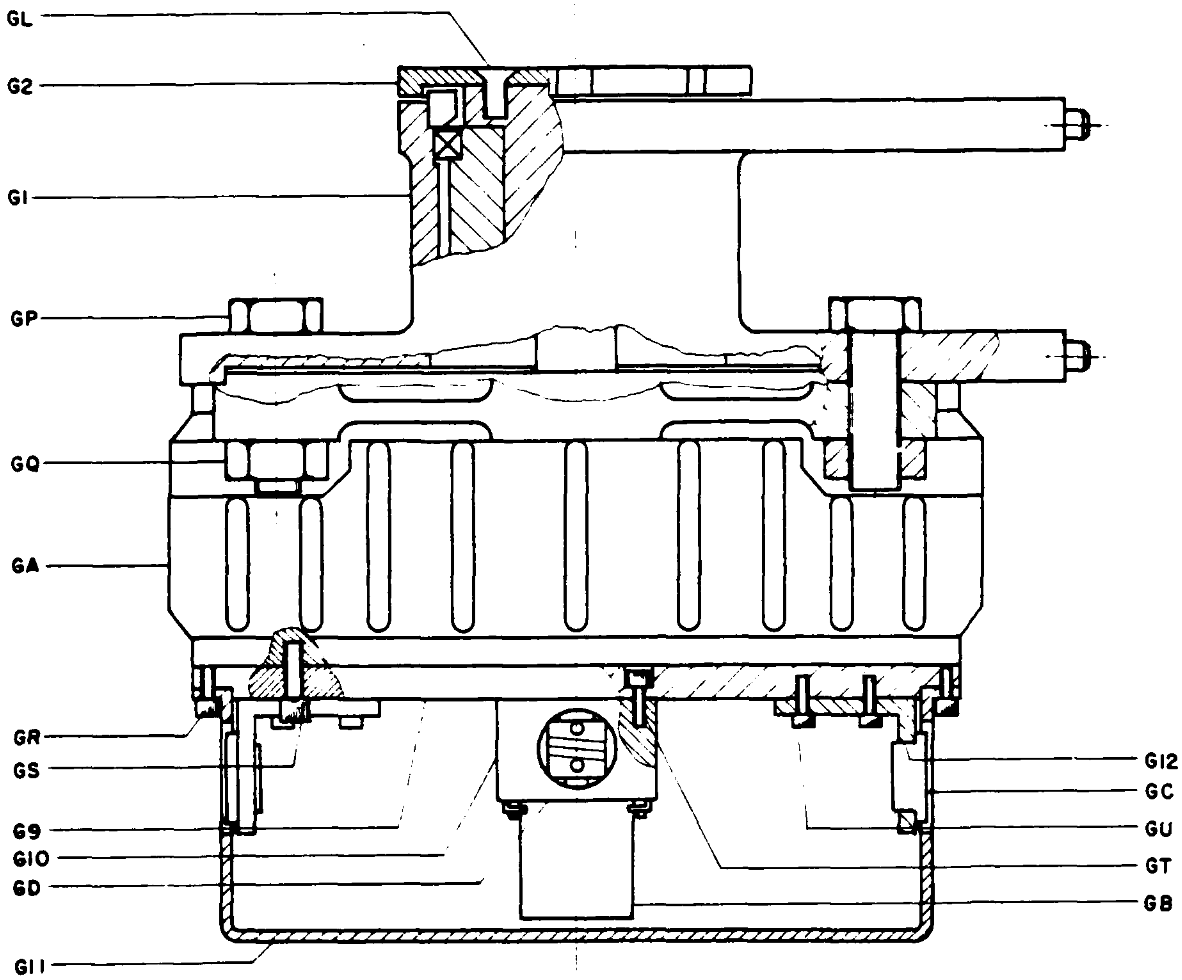

Figure 39. Assembly of the crank pin driver unit.

KEY TO FIGURE 39

ITEM	QUANTITY	DESCRIPTION
G1	2	Driver Unit Body.
G2	2	Bearing Protection Plate.
G9	2	Transducer Mounting Plate.
G10	2	Transducer Mounting Sleeve.
G11	2	Protective Cover.
G12	3	Connector Bracket.
GA	2	DC Servomotor (crank pin drive motor), 3000 watt printed armature motor type MF26. Supplier - Printed Motors Limited, Bordon Trading Estate, Oakhanger Road, Bordon, Hampshire GU35 9HY, UK.
GB		Rotation Transducer (crank pin position transducer), inductive rotary transducer type 3800/290. Supplier - Penny and Giles Potentiometers Limited, Grovely Road, Christchurch, Dorset, UK.
GC	3	Electrical Connector (motor and transducer supply), low loss 20-way connectors complete with cover shells, types: plug 466-040, socket 466-084, cover shell 466-129. Supplier - RS Components Limited, PO Box 427, 13-17 Epworth Street, London EC2P 2HA, UK.
GD	1	Flexible Coupling (transducer to motor shaft coupling), type BS24/0.375/0.1875. Supplier - Reliance Gear Company Limited, Almondbury, Huddersfield, UK.
GL	6	Socket Countersunk Head Screw (bearing protection plate retaining) M8 x 15 HTS.
GP	8	Hexagon Setscrew (servomotor retaining), M14 x 55 HTS.
GQ	8	Self Locking ("Nylock") Nut (servomotor retaining), M14.
GR	12	Socket Cap Screw (protective cover retaining), M5 x 10 HTS.

KEY TO FIGURE 39 (Continued)

ITEM	QUANTITY	DESCRIPTION
GS	8	Socket Cap Screw (transducer plate retaining), M6 x 18 HTS.
GT	12	Socket Cap Screw (transducer mounting sleeve-retaining), M5 x 10 HTS.
GU	9	Socket Cap Screw (connector bracket retaining), M6 x 10 HTS.

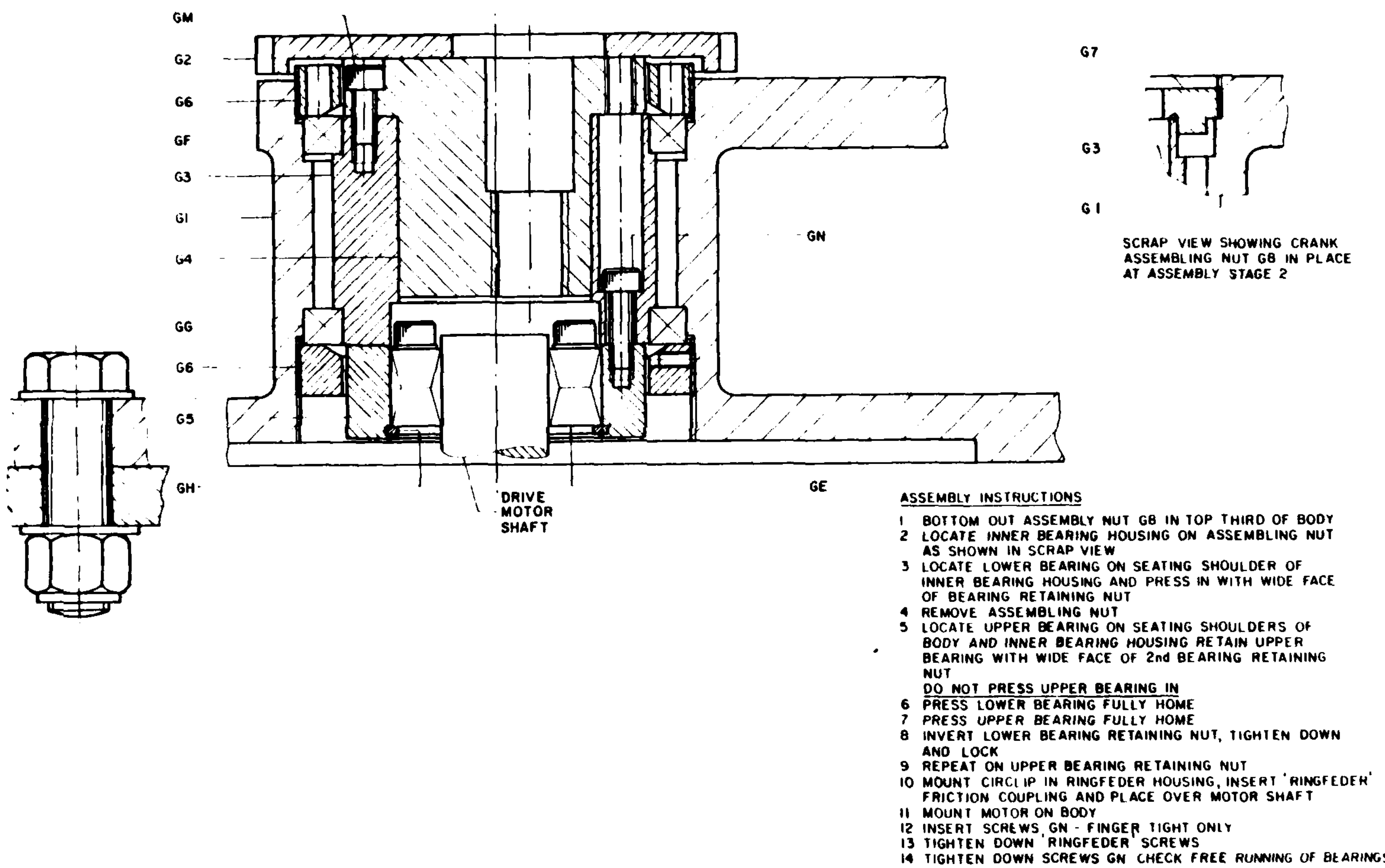

Figure 40. Assembly of the bearing support system.

KEY TO FIGURE 40

ITEM	QUANTITY	DESCRIPTION
G1	2	Driver Unit Body.
G2	2	Bearing Protection Plate.
G3	2	Bearing Hub.
G4	2	Crank Pin Plug.
G5	2	Ringfeder Coupling Sleeve.
G6	4	Bearing Retaining Nut.
G7	1	Assembly Nut.
GE	2	Expanding Friction Company (motor shaft coupling), 55 x 28 ø type RFN 7012. Supplier - Ringfeder Limited, Forum Drive, Midland Industrial Estate, Rugby, Warwickshire CV21 1NT, UK.
GF	2	Deep Groove Ball Bearing (upper support bearing), 80 x 100 x 10. Supplier - SKF ref 61816.
GG	2	Deep Groove Ball Bearing (lower support bearing). Supplier - SKF ref 61816.
GH	2	Circlip (Ringfeder coupling retaining), exterior dia 55, Anderton ref 1300.
GM	12	Socket Cap Screw (crank plug/bearing hub retaining), M6 x 15 HTS.
GN	12	Socket Cap Screw (bearing hub/Ringfeder housing retaining), M6 x 20 HTS.

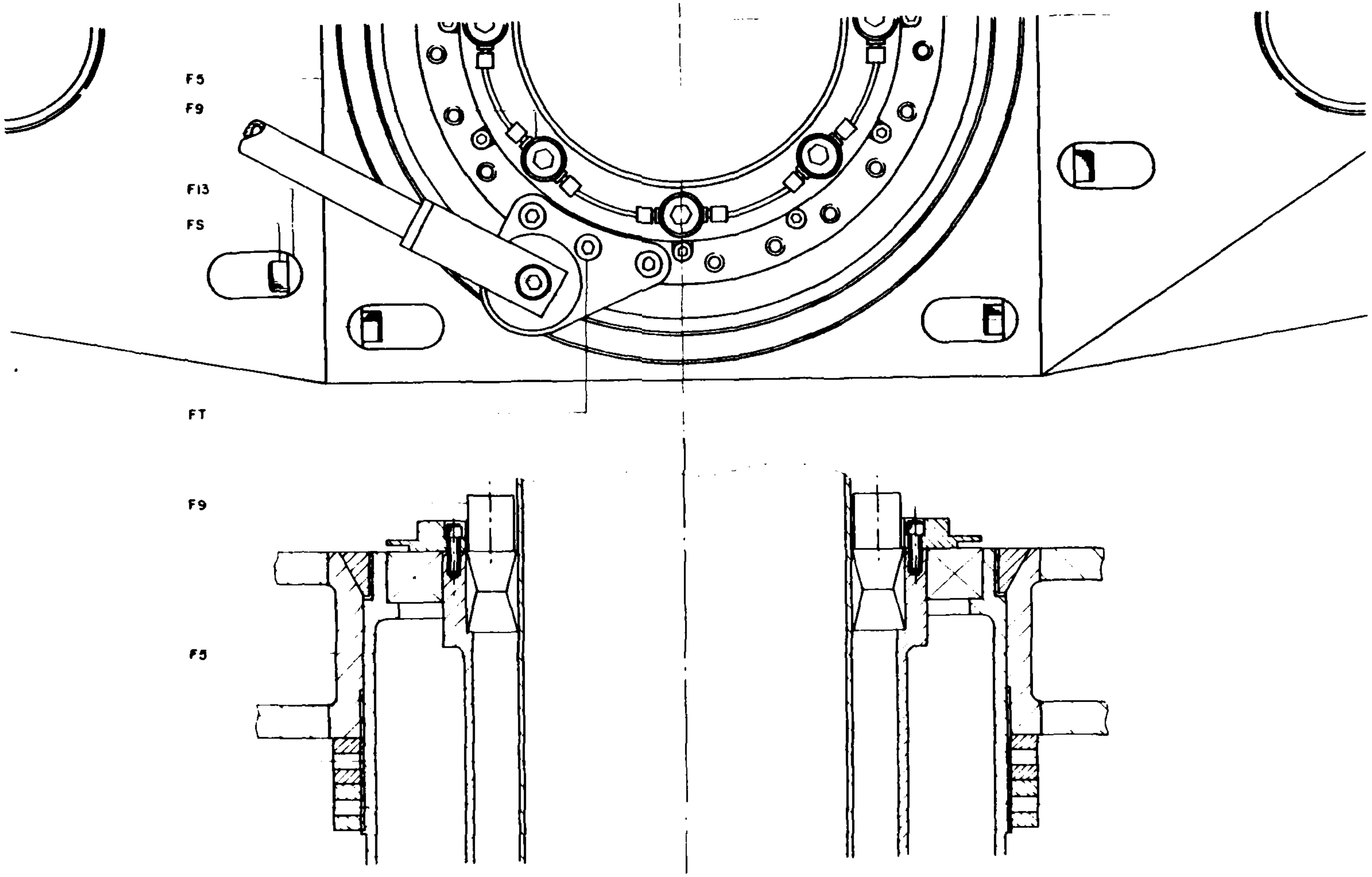

Figure 41. Assembly of the torque tube coupling system - plan.

KEY TO FIGURE 41

ITEM	QUANTITY	DESCRIPTION
F5	1	Hub Housing.
F9	16	Hydraulic Actuator Cylinder.
F13	12	Saddle Washer.
FS	12	Socket Cap Screw (hub housing/crank pin driver units - fastening), M10 x 35 HTS.
FT	6	Socket Cap Screw (connecting rod - torque tube bearing plate to inner sleeve - retaining), M8 x 25 HTS.

the flanges. The system of bearings and couplings in the core of the body perform a number of functions:

(1) The assembly provides a hub to support the crank pin and absorb the loads induced in it from the connecting rod.

(2) The hub is supported in such a way that it provides a rigid yet free running mounting

(3) To avoid large radial shaft loads on the motor due to misalignment with the hub, a self-aligning coupling is configured in the assembly.

A detailed assembly of the crank pin support system is given in Figure 40. Being loaded in single shear, the pin is subject to bending as well as shear loads and, therefore, requires a mounting that provides firm and stiff support. As shown in Figure 39, this is achieved by mounting the pin in a deep bushing and retaining it at the root by of a threaded joint. Lateral support is provided by a close machined fit of the neck of the pin with the plain bore in the crank pin plug (G4, Fig. 40) and the threaded root of the pin is isolated from bending by a generous allowance of length of the bushing. To provide a free running support for the plug and to isolate the drive motor shaft from radial loads, a system of support bearings (GF and GG, Fig. 40) is incorporated into the design. These bearings are located as far apart as possible to maximize the stiffness of support and are preloaded by a pair of retaining nuts G6 to take out radial play in the system.

The friction locking coupling GE is the final link in the assembly of rotating parts and is the means, in conjunction with the sleeve G5 and the bearing hub (G3, Fig. 40) for implementing a self-aligning coupling. The need for this feature in the system arises from the characteristics of the systems being coupled. Both the motor shaft and the support system for the crank pin are rigidly mounted. Under these circumstances there is the risk of large radial forces occurring if the coupling between these assemblies is not precisely aligned (accurate alignment of the rotating components in the housing is ensured by the one piece machined concept of the unit where all seating recesses in the housing are machined at one setting). Figure 40 shows how this problem is overcome by the addition of two independently acting friction couplings into the system. Torsion is coupled from the motor shaft to the sleeve G5 by the expanding Ringfeder coupling (GE, Fig. 40). This unit, being a small scale version of the torque tube friction couplings mentioned in the previous section uses a bolt driven wedge system acting on a pair of split rings to provide the locking action. The second friction joint is formed between the bearing hub (G3, Fig. 40) and the Ringfeder coupling sleeve (G5, Fig. 40) where face compression is achieved by the array of bolts (GN, Fig. 40). These bolts act through relatively large diameter clearance holes in the bearing sleeve and, hence, the collar is allowed within limits to take up any lateral position resulting from the expanding coupling. Access to the collar/ sleeve fastening bolts is through counterbored holes in the sleeve, and to the expanding coupling bolts by removing the crank pin plug (G4, Fig. 40). Overall protection for the rotating components in the housing is provided by an overhanging plate (G2, Fig. 40). This plate, attached to the crank pin plug, features a series of axial slots machined into its outside diameter. These slots are compatible with a commercially available hook spanner (originally intended as a bearing retaining nut spanner) which is used to provide a stabilising torque reaction on the rotating components of the housing during fitting or removal of the crank pin.

A rotation transducer is coupled to one of the drive motor shafts for sensing the position of the crank pin. As shown in Figure 39, this unit is attached via a flexible coupling (GD, Fig. 39) to a rear shaft extension on the motor. The transducer body is mounted on a spacer sleeve which accommodates the coupling, and a lateral through hole in the tube allows access for tightening up the coupling during final assembly. Concentric positioning for the transducer and spacer tube is picked up from a rear spigot on the drive motor body by the mounting plate (G9, Figs. 39 and 40) and transferred to a mounting spigot on the spacer sleeve. The mounting plate is retained by screws GS to the rear of the motor and the tube by screws GT to the plate. The plate also provides mounting for electrical connectors via the brackets (G12, Fig. 39). Overall protection for the assembly is provided by the cover (G11, Fig. 39) which is fastened to a recessed lip on the plate. Only one of the driver assemblies is fitted with a rotation transducer since this is all that is required to sense the position of the linkages in the system. However, to preserve dynamic mechanical balance in the system, both crank drive systems are fitted with the mounting plate spacer sleeve, and protective cover components.

d. The Connecting Rod Assemblies The connecting rods form the center element of the three arm linkage system, the function of which is to enhance the torque output from the crank pin drive units. Coupled between the crank pins and the inner bearing sleeve of the torque tube coupling assembly, the connecting rods are designed so that their axes intersect the radius arms of the linkage system at right angles (see Fig. 35). Figure 42 shows that the connecting rods are based around a steel tubular member (H5, Fig. 42) which transmits the thrust and tensile loads. At the crank pin end the thrust rod terminates in a housing (H1, Fig. 42) which is used to retain the bearing (HA, Fig. 42). This bearing, seated under the hexagon bolt head of the crank pin, is a double-row, self-aligning bearing. The crank pin is captive in the

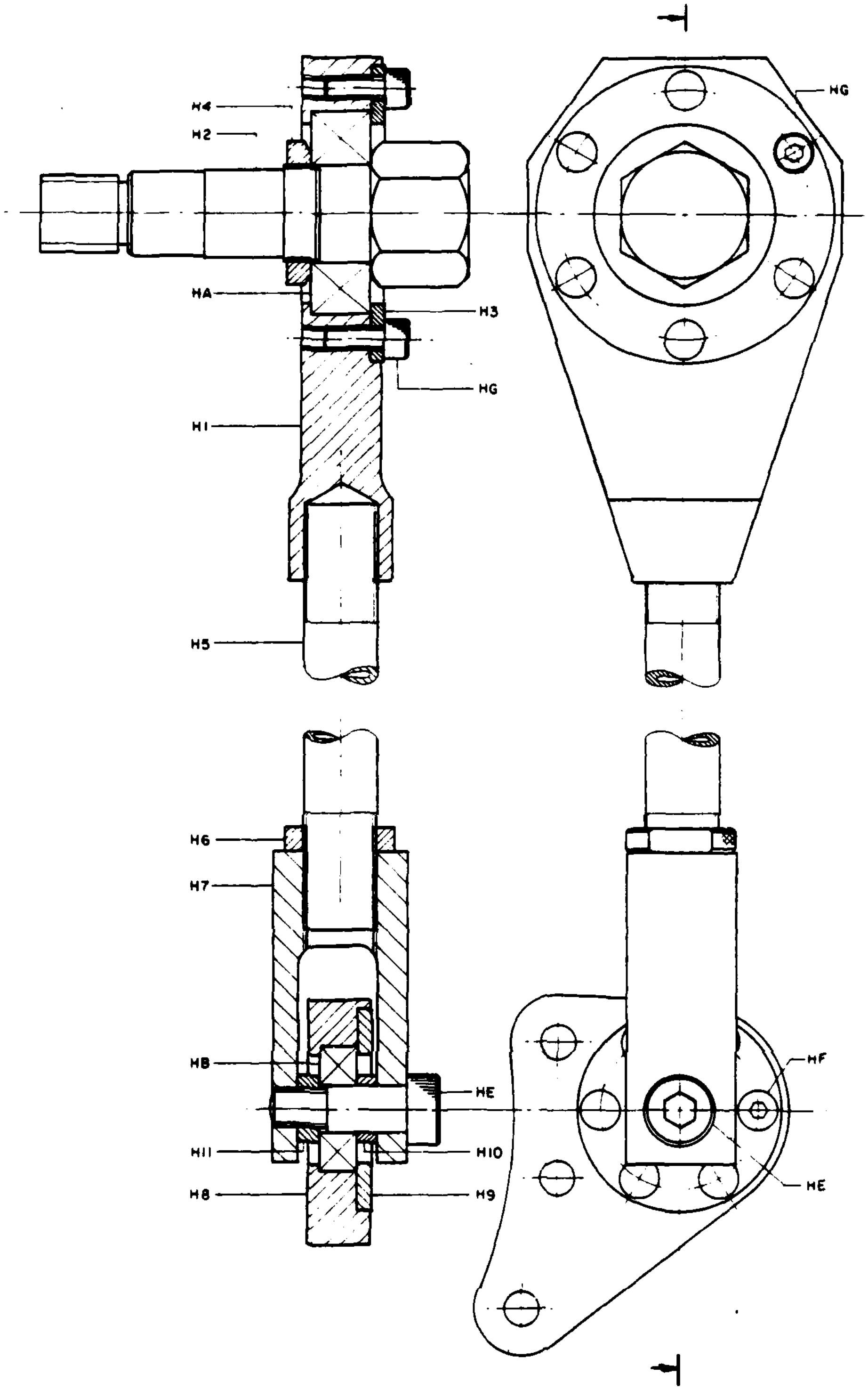

Figure 42. Assembly of the connecting rod.

KEY TO FIGURE 42

ITEM	QUANTITY	DESCRIPTION
H1	2	Crank Pin Bearing Housing.
H2	2	Crank Pin.
H3	2	Crank Pin Bearing Plate.
H4	2	Crank Pin Retaining Nut.
H5	2	Thrust Rod.
H6	2	Lock Nut.
H7	2	Rod End Clevis.
H8	2	Torque Tube Bearing Housing.
H9	2	Torque Tube Bearing Plate.
H10	2	Upper Bearing Spacer.
H11	2	Lower Bearing Spacer.
HA	2	Self Aligning Double Row Ball Bearing (crank pin end bearing), 25 x 52 x 15, SKF ref 1205.
HB	2	Self Aligning Double Row Ball Bearing (torque tube end bearing), 12 x 32 x 9, SKF ref 1201.
HE	2	Socket Shoulder Screw (torque tube bearing pin), 12 (M10) x 20 HTS.
HF	12	Socket Countersunk Head Screw (torque tube end bearing plate - retaining), M5 x 10 HTS.
HG	12	Socket Cap Screw (crank pin end bearing plate - retaining), M6 x 15 HTS.

assembly, being retained by the nut (H4, Fig. 42) on the inner ring of the bearing and by the plate (H3, Fig. 42) on the outer ring. A clevis configuration is used at the torque tube end and thrust loads are coupled into the bearing from the clevis (H7, Fig. 42) via a shoulder screw HE into the inner ring of the bearing (HB, Fig. 42). To allow a certain amount of vertical misalignment in the connecting rod termination points, a self-aligning bearing is again used. The inner ring of the unit is positioned within the fork of the clevis by spacers (H10 and H11, Fig. 42) and the outer ring is retained in the housing H8 by the retaining plate (H9, Fig. 44). The overall length of the connecting rods is adjustable at the torque tube end by the threaded attachment to the clevis. This arrangement allows adjustment in increments of one thread pitch. Lock nut (H6, Fig. 42) is used to hold the setting. At the crank pin end the thrust rod is permanently fitted into the bearing housing by stud locking compound.

The connecting rod assemblies (Fig. 43) have been designed to be conveniently detachable from the actuator unit (Fig. 44). To achieve this the crank pins are threaded into the drive assemblies and a hex head bolt is provided on the pin for use in the assembly or dismantling procedure. Reaction against the torque applied to the crank pin can be applied to the bearing protection plate G2 of the drive units, which are provided with a slotted exterior for attaching a hook spanner. Coupling to the inner bearing sleeve is achieved via a face friction joint between the bearing retaining plate (F3, Fig. 37) and the torque tube bearing housing (H8, Fig. 42). Three bolts (FT, Fig. 41) are used to retain the housing, and the top face of the retaining plate is provided with a regular bolt circle of threaded attachment holes as shown of Figure 41. A special cap (H12, Fig. 45) has been designed for protection of the crank pins. This item should be fitted when the connecting rods are removed from the actuator assembly.

Figure 43. Top view of actuator showing the connecting rods and the upper hydraulic locking assembly.

Figure 44. Removing the connecting rods.

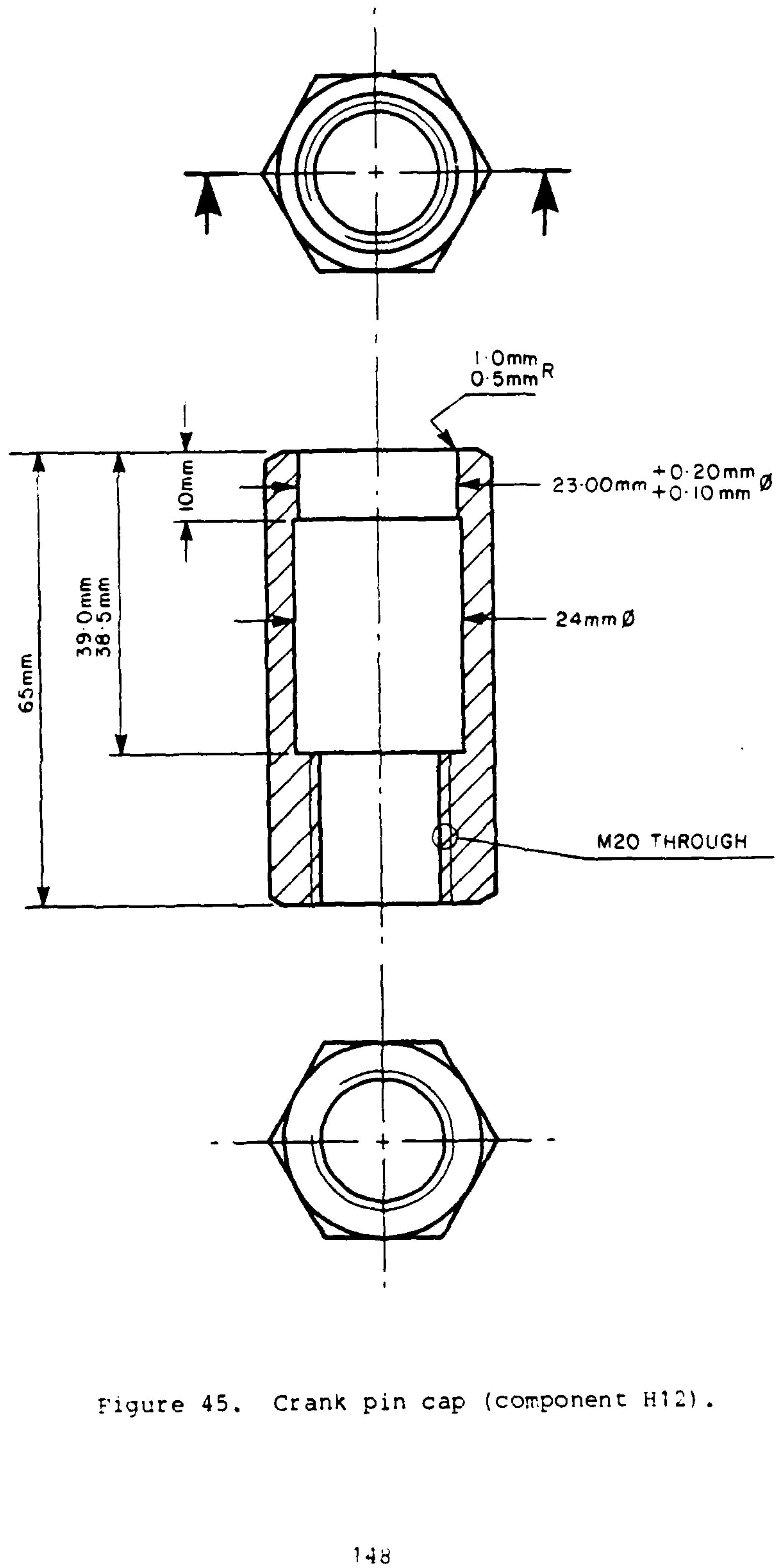

Figure 45. Crank pin cap (component H12).

e. <u>The Electrical Drive System</u> A schematic of the control and drive system for the dynamic actuator (Figs. 46 and 47) is given in Figure 36. It can be seen that the system can be divided into three basic elements or subsystems:

(1) the power drive amplifier

(2) the rotation sensing transducer

(3) the electronic summing circuit

Oscillatory drive torque is achieved by feeding a bipolar (positive and negative swing) drive voltage to the motors, which being dc permanent magnet field types, produce a torque proportional to the drive current. To balance the drive currents to the motors (and, hence, the couple applied to the torque tube by the actuator), the units are connected in series. This might not appear to be the best way of connecting the motors at first, considering that the drive current is an important factor. With the in-series connection, both the back emf (due to shaft rotational velocity) and the armature resistances are added together. To produce a given torque and, hence, armature current, a drive voltage twice that of the alternative parallel configuration is required. However, with the limited angular movement required of the crank pin the drive motors, even at the higher operating frequencies, do not attain significantly high shaft velocities. In addition, the disc rotor motors chosen for the application feature very low armature resistance (0.37 Ω). Thus, the required drive voltages can be obtained relatively easily and the in-series motor configuration, with its inherently balanced drive characteristic, can be used without sacrifice of torque output from the system.

The power amplifier selected for driving the dynamic actuator is a 600 W class AB linear device capable of bipolar output drive (positive and

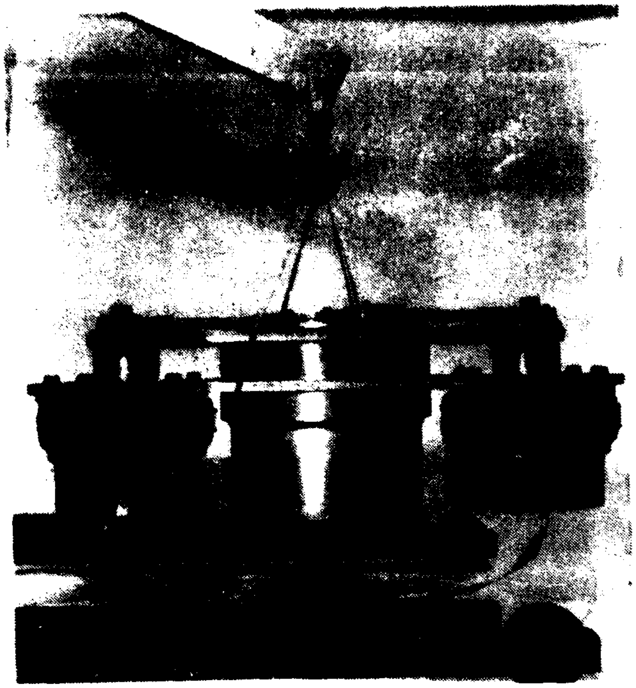

Figure 46. General view of the dymanic actuator.

Figure 47. The actuator, wedge nut spanner and bearing plate hook spanner.

negative output swing) from dc up to frequencies well beyond the range of interest for the dynamic actuator application. It can drive the servo-motors at their full rated load current when not blower cooled (approximately 25 A), and therefore is capable of working the system to its design ratings. The amplifier has some useful features which are well suited to the task considered here:

(1) The drive voltage and current being supplied to the load are indicated on the amplifier's front panel. These may be used to control and monitor the output of the actuator.

(2) Overload and thermal cut-outs are provided to prevent damage to the amplifier and the load caused by incorrect operation of the system.

(3) The overload trip includes a soft start-up interlock where the amplifier can be powered only when the gain control is set to zero. As will be seen later, this feature is put to good use in the linkage position control system.

(4) The amplifier is powered from a single phase ac power source and operates from a 120 or 240 V, 50 or 60 Hz supply. This means that it can be powered conveniently the field and requires no basic modification to operate on American or European standard power sources.

In addition to the above features, the amplifier is fitted with an internal sinewave oscillator source which can be switched into the input and adjusted by amplitude and frequency controls on the front panel. Unfortunately, however, this internal oscillator cannot be used with the dynamic actuator since the external input is disabled when the oscillator is switched into circuit. This means that implementing the static position control function would require modification to the internal circuitry of the amplifier.

Even considering this step, no means was found whereby the soft start-up facility could be used for the complete system (i.e. dynamic drive plus static positioning) when using the internal oscillator. Thus, it was decided to ignore this facility and to use the amplifier switched to operate in the external input mode and connected to a supplementary electronic system. As shown in Figure 48, an oscillator or external function generator is connected to the system to provide the dynamic drive command signals for the actuator.

f. The Static Position Control System The need for an actuator control system to provide static positioning of the linkages was mentioned earlier. Being essentially a dual three-arm-linkage device, the gearing ratio provided by the linkages is dependent on the absolute angular position of the elements (see Figure 35). The dynamic actuator system requires that the radius arms be positioned at near to right angles with the center-line of the connecting rods. This permits the system to develop a maximum useful angular excursion. Also, the variation in effective gear ratio with this excursion is kept to a minimum. However, this linkages position is not an intrinsically stable one and, therefore, some form of position control system is required to maintain it. The scheme designed for this purpose is shown in Figure 36. The position of the linkages is picked up from one of the driver motor shafts by a rotary position transducer. This signal is compared to a static reference potential via a differencing circuit which produces a position error signal. The error signal is then filtered to remove the dynamic component and fed to the drive amplifier along with the externally derived dynamic command input signal. Functioning essentially as integrating elements in the feedback loop, the drive motors do not require a net offset or steady drive voltage to maintain the static or mean position of the linkages. Consequently, the output drive capability of

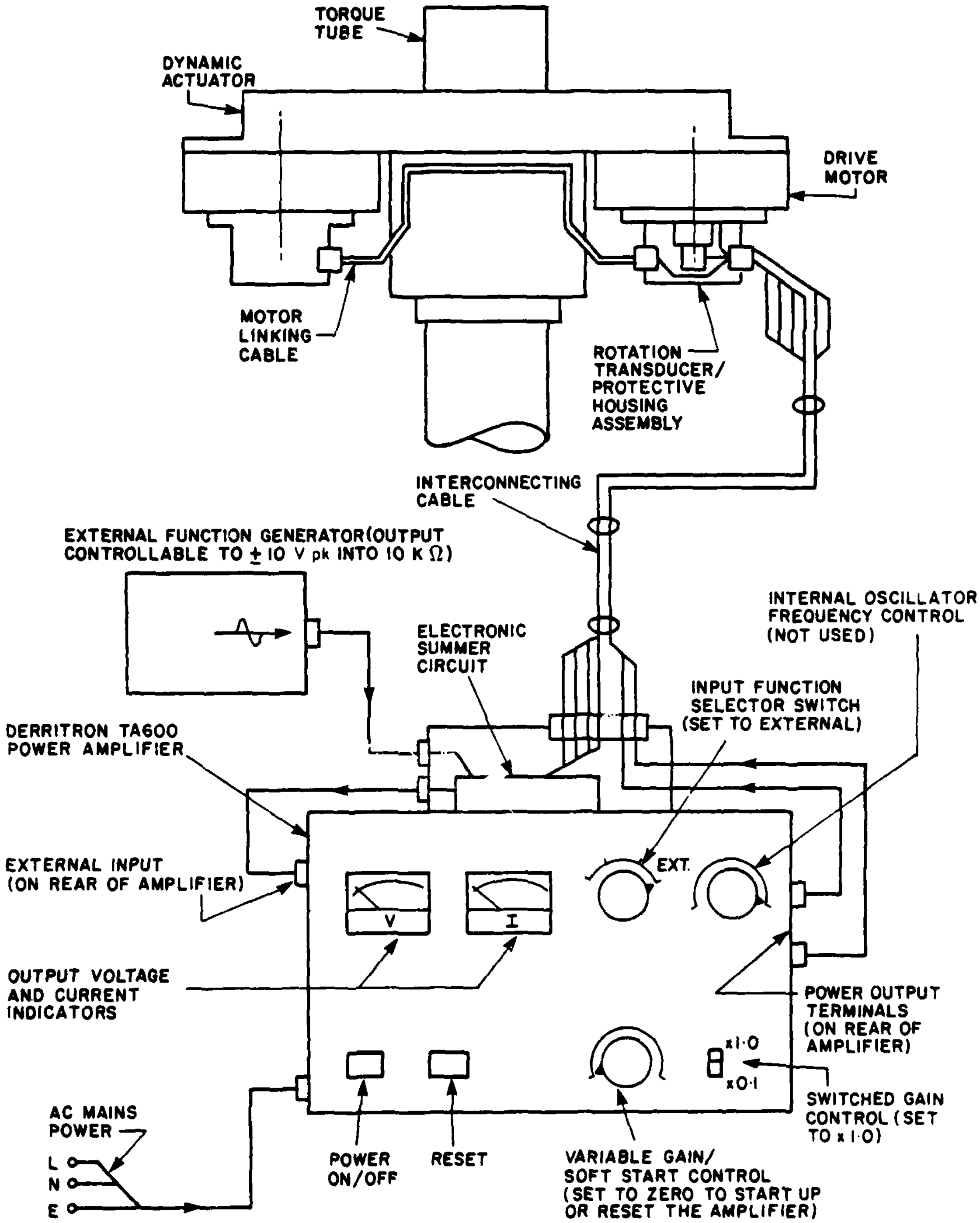

Figure 48. Overall schematic of the dynamic actuator control system.

the amplifier is not impaired by this task and is available in full for the dynamic function.

The rotation transducer (Fig. 49) senses the position of the three-arm linkage by measuring the angular excursion of one of the drive motor shafts. To achieve this, the rotation transducer is mounted at the rear of the motor using a specially designed plate and sleeve arrangement. A flexible coupling attaches it to a shaft extension. The particular device selected for the application (Penny and Giles model 3810/300) is an inductive type and, therefore, does not utilize a wiper or brush type pick-up system. This was thought to be desirable in order to minimize the effects of wear and noise which the rotation transducer must withstand. A built-in dc to ac converter and demodulator circuit enables the transducer to produce a dc rotation output signal from a dc energizing source.

Figures 50 and 51 show the electronic summing circuitry. In Figure 51, the rotation transducer output is compared with a reference potential by amplifier OA1, which integrates the difference between the transducer output and the reference potential. (The integral function is utilized to produce good positioning accuracy for the linkages and to filter out the dynamic component of the transducer signal.) The error signal thus produced is fed via a loop gain control potentiometer VR1 to a second amplifier OA2 which sums it with the dynamic input signal to produce a drive signal for the power amplifier (Fig. 52). The external dynamic drive input to OA2 is decoupled by R7 and C3 in order to prevent any dc component of the external drive signal from altering the static positioning of the linkages. Stabilized supply potentials for the summing circuitry are derived from unsmoothed supplies within the power amplifier, which is the function of the additional circuitry shown in Figure 49.

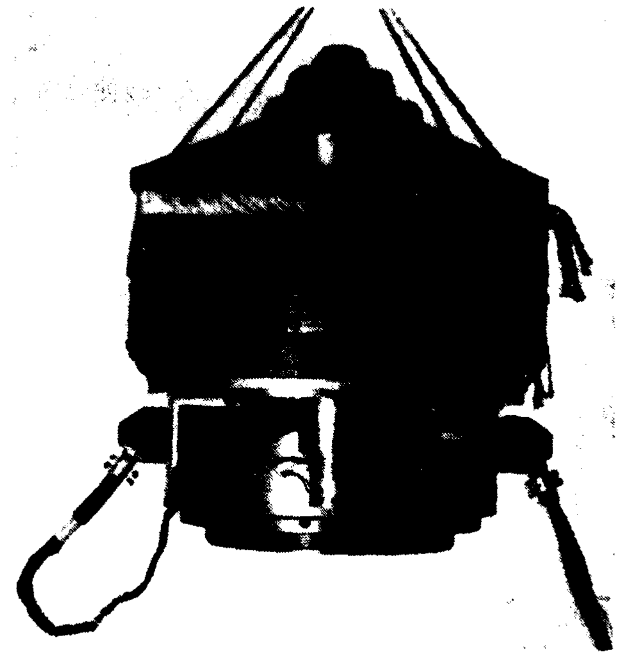

Figure 49. View of the rotation transducer installation.

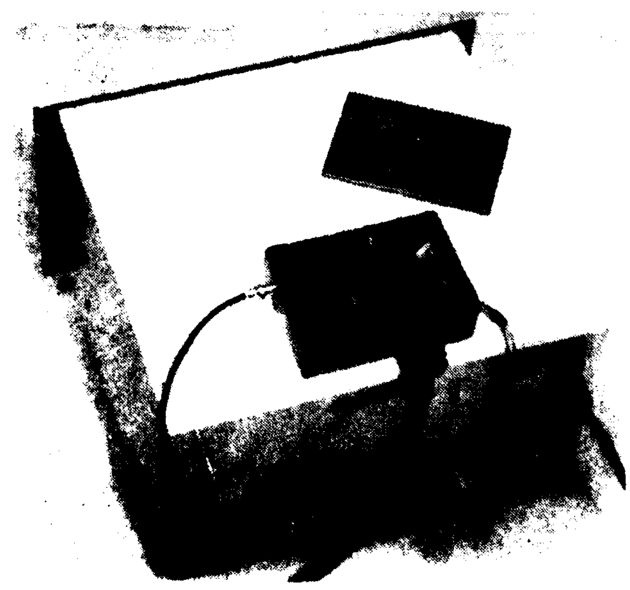

Figure 50. The electronic summing circuitry.

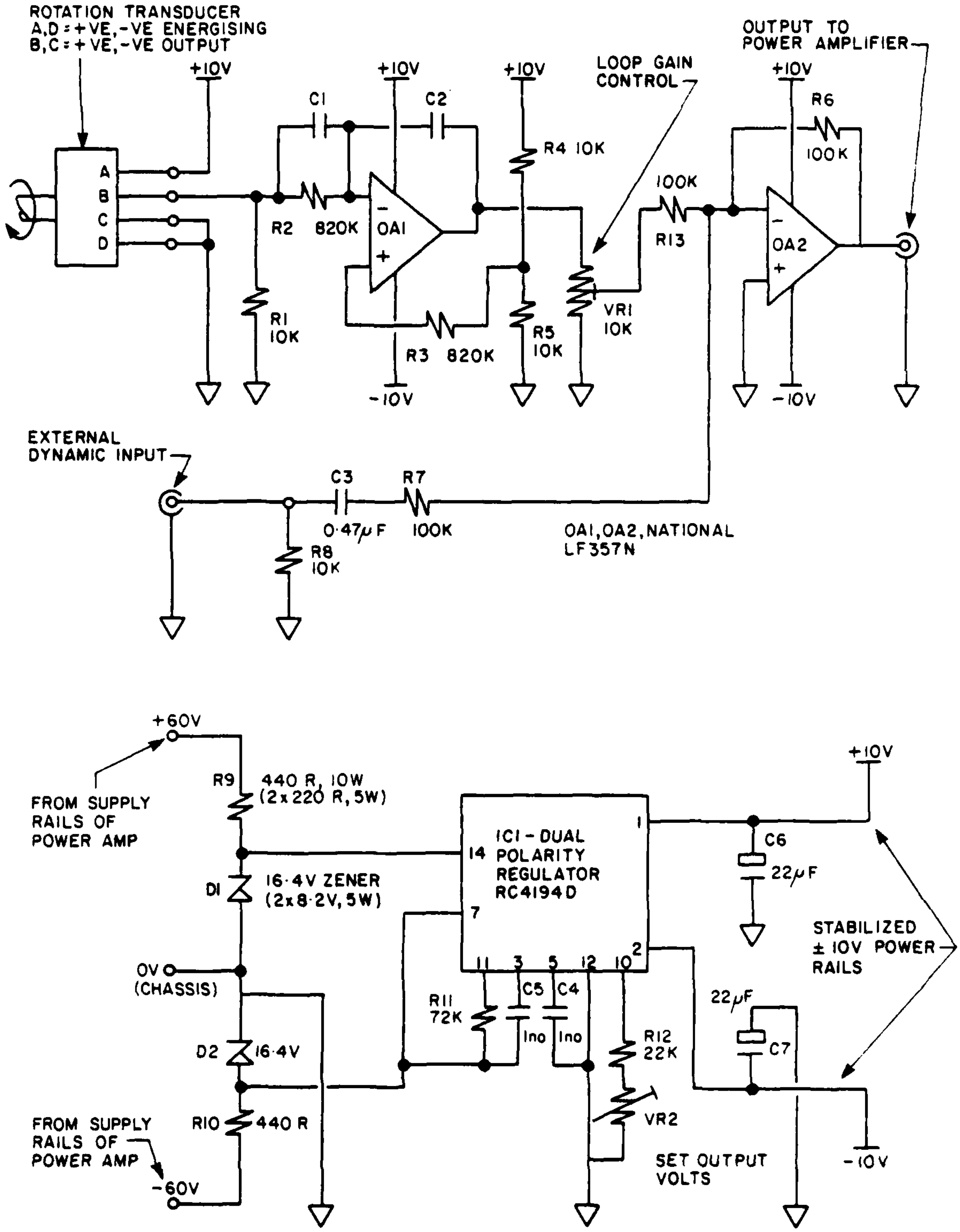

Figure 51. Diagram of the electronic summing circuitry.

Figure 52. The power amplifier and function generator setup

Unsmoothed 60 V power rail potentials (Figure 51) from the amplifier are reduced and partially stabilized to 16 V by the zener diode/resistor networks R9, R10, D1 and D2. These are further stabilized by the *integrated circuit* regulator IC1. This circuit produces precisely controlled $\pm$10 V power supply rails for driving the summing circuitry and for energizing the rotation transducer. As shown in Figure 48, the summing circuitry is mounted in a box attached directly to the power amplifier case. In this way the interconnections between the two systems is permanent and there is a reduced likelihood of their being set-up incorrectly. The electronic summing circuitry and power amplifier are connected to the dynamic actuator by a multiway cable which carries the conductors required for both the motor drive supply and for the rotation transducer.

As mentioned previously, the rotation transducer for the control circuit is mounted on the rear of one of the drive servomotors using a plate and sleeve arrangement. To provide overall protection for the assembly a steel top-hatted cover is bolted to the transducer mounting plate. Demountable connectors are used to provide electrical connection between motor units and to the power drive and control system. These connectors are accessible through openings in the protective cover.

g. The Actuator Test Rig In order to operate the actuator and check the performance of the device some form of testing rig was needed. The rig was required to provide a vertical tube mounting so that the friction locking feature and the dynamic torsion capabilities of the device could be operated at, or near to, the full design ratings.

For the dynamic torsion aspect of the testing, the mounting tube was required to be torsionally stiff. The actuator can develop torque at the higher frequences only if the stiffness of the load permits only small angular movements to occur. This is explained if the actuator system is visualized as a driving torque in parallel with a lumped rotational inertia attached to the torsion tube. If the torsion tube is stiff, angular twist will be small and little torque will be lost in accelerating the inertial component. Below the resonant frequency of the mass/spring system the load will appear predominantly as a torsional spring and most of the torque developed will be coupled into the tube. Above this frequency, however, the inertial component dominates and the torque coupled into the tube falls off with the inverse square of the excitation frequency.

The actuator test rig (Figure 53) was a rigid frame constructed from 100-mm steel channel and box section and mounted on four 250-mm^2 friction pads. The pads comprised a steel plate, bonded to the underside of which was a special high friction rubberized cork material. The plates were coupled to the box section spans by angle section brackets which could be independently adjusted to rest squarely on the ground. This feature was felt to be necessary since uniform contact with the surface of the ground or floor was an important

Figure 53. The actuator test rig.

Figure 54. Testing in progress.

consideration for the friction coupling. The torque tube was machined from seamless steel tubing and was attached to the rig by a 12.5-mm-thick steel base plate. The test rig (Fig. 54) was designed to operate in the laboratory on a smooth finished (e.g. bonded linoleum tiled) floor. The dimensions of the rig and of the friction pads were such that only a small shear stress was transmitted through the friction pads due to the applied dynamic torsion. This minimized the effect of the friction pads and of the supporting surface in tending to reduce the effective stiffness of the rig.

The hydraulically actuated Ringfeder friction couplings were the first system to be tested using the experimental rig. The objective was to determine if the rig could satisfactorily suspend the actuator on the torque tube, and if it could transmit the dynamic torsional loading without slipping. The rig was found to work most satisfactorily, the only problems were difficulties in releasing the m· .anism. This was resolved by adding opposing jacking springs in the Ringfeder couplings. The springs were placed between the thrust rings of the coupling in the unused positions formerly occupied by the drawbolts. The locking mechanism was released by dropping the supply pressure to zero and hoisting the actuator off the torque tube. After only a slight initial movement, or working, the friction coupling released and the actuator was free. It was found that applying a certain amount of dynamic torsion greatly assisted in freeing the system. During the trials the linking mechanism began operating at pressures of about 2,000 lb/in^2. Positive locking for both static and dynamic functions occurred at 3,000 lb/in^2. Since the maximum rated pressure for the sealing system of the hydraulic actuators is 3,500 lb/in^2, it is recommended the locking pressure be maintained between the limits 3,000 to 3,5000 lb/in^2 for all future operation of the actuator.

During the evaluation of the dynamic performance of the actuator, the system was locked onto the test rig torque tube and dynamic drive was applied over a 10 to 50 Hz frequency range. The actuator was found to operate satisfactorily over this frequency range. Most of the rotational movement that was observed was that of the actuator structure about the central torsion tube, indicating that virtually all the developed torque was being delivered to the rig. Contrary to some concern felt during the development work about amplifier protection trip out, it was found that the system was particularly tolerant to a wide range of dynamic drive amplitudes. Using the maximum rated drive current for the power amplifier (25 A) caused the unit to trip out, but no other cause was observed during the trials. At practically all operating frequencies (except where the drive level caused excessive movement of the crank pins), the system was capable of accepting drive current at, or near, the maximum rated level.

Following the trials discussed above, the various subassemblies of the dynamic actuator were dismantled and inspected to determine any signs of wear or fatigue and to try to detect loosening of any parts that might have occured under the vibratory loading. The results of the inspection were most encouraging. No evidence of loosening or wear was found in the ball bearing assemblies. Some of the structural platform components (in particular the lock nuts on the outer sleeve of the torque tube coupling assembly, and the taper locking nut) were found to be loosening, but, this was readily cured by the sparing use of a low holding force adhesive locking compound (Loctite 290 penetrating adhesive).

The start-up and operating procedure developed during the testing and recommended for future use of the system is as follows (see Fig. 48):

(1) Ensure that the following interconnections are made:

   a. The multiway interconnecting cable between the summing system box and the actuator.

   b. The multiway interlinking cable between the driver units of the actuator.

   c. The coaxial cable between the summing system box and the input of the power amplifier.

   d. A connection to the input of the control system box from a function generator (sinewave source, variable output from zero to $\pm$10 V peak into 10 KΩ , 1 to 100 Hz variable).

   e. A connection to the power input socket of the amplifier from a 100 to 125 V, 50 to 60 Hz single phase main supply (minimum supply current capability, 9.5 A).

(2) Before switching on, set the front panel controls of the amplifier as follows: (The frequency control setting is arbitrary.)

   a. The output control fully counterclockwise.

   b. The range control to ext.

   c. The gain switch to X1.

(3) Set the output of the external function generator to 0.

(4) Press the mains on button and observe light-up of the mains on and reset button legends. Depress the reset button and hold until the light goes out.

(5) Check that the engraved lines on the driver unit chassis of the actuator and the rotating cover plates are aligned. Both plates must be set to their approximate operating positions.

(6) <u>Slowly</u> advance the output control until the static positioning control system begins to cause movement of the driver units. Wait

until the system has settled and continue to advance the output control until fully counterclockwise.

(7) Apply an input from the external function generator noting the voltage and current drive levels applied to the actuator. Exceeding the 25-A maximum drive current will cause the amplifier to trip out.

(8) If the amplifier trips, return the output control to the fully counterclockwise position and repeat this procedure from (3) onwards.

3. THE STATIC ACTUATOR

a. General Discussion The prime function of the static, or more correctly, the quasistatic actuator is the operation of the BSD probe for testing at high shear strain levels in the surrounding soil. The device is required to develop the full rated design torque for the system (10^4 N·m) and to be able to apply in a controlled manner a variety of loading histories (ie., cyclic or monotic functions).

The BSD test is a new concept. In the absence of direct field experience of the test, it is difficult to set down a detailed specification for a system which would be ideally suited to the static actuator application. From largely practical consideration, however, a number of features are readily suggested:

(1) The unit should be of modular construction and the subassemblies should be of manageable bulk.

(2) The unit should be capable of a controlled torque or controlled rotation characteristic.

(3) Operation should be provided over as wide a range of speed as possible. The maximum speed of operation should be governed only by considerations of power requirements.

(4) There should be free access to the inside of the torque tube string.

(5) To avoid possible injury or damage, the system should be inherently safe. This suggests that the unit should be characteristically stiff in order to minimize stored strain energy.

The design evolved to meet the above requirements is discussed in the next section. The unit, which is built into a compact and stiff steel plate structure, uses a pair of hydraulic motors as the mechanical power source. The static actuator is attached to the torque tube, as with the dynamic actuator, by an expanding friction coupling, and the design features a general reaction frame assembly which can be used with ground anchors or with a deadweight/friction system to provide restraint for the actuator.

b. Description of the Static Actuator A schematic of the static actuator is given in Figure 55 where it can be seen that the unit is basically a dual-drive, worm reduction gear system powered by a pair of hydraulic motors. These units, JA in the Figure 56, are mounted on opposite faces (J4, Fig. 56) of the rigid plate chassis of the assembly, and are coupled to a common pressure supply line. The motors drive the worm pinions directly via rigid shaft couplings (J7, Fig. 56). With this arrangement the torques applied to the worm pinions and, hence, the tangential forces about the wormwheel, are balanced. This results in a relatively pure torque output couple from the unit. Axial forces in the worms are absorbed by a sturdy system of taper roller bearings in the hydraulic motors, and the drive shafts are laterally restrained at the opposite end of the drive motor by a plate mounted bearing (JD, Fig. 56).

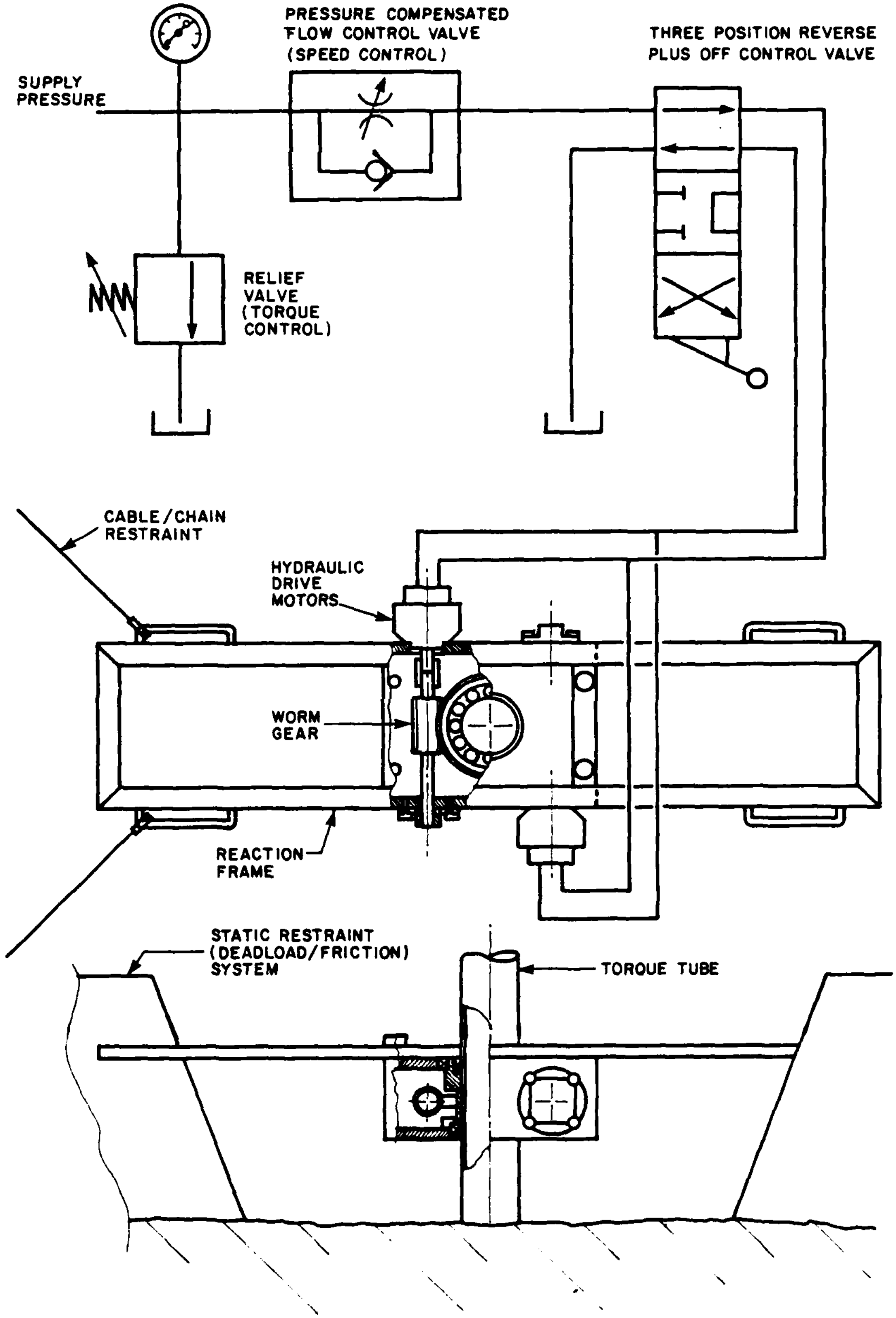

Figure 55. Schematic of the static actuator system.

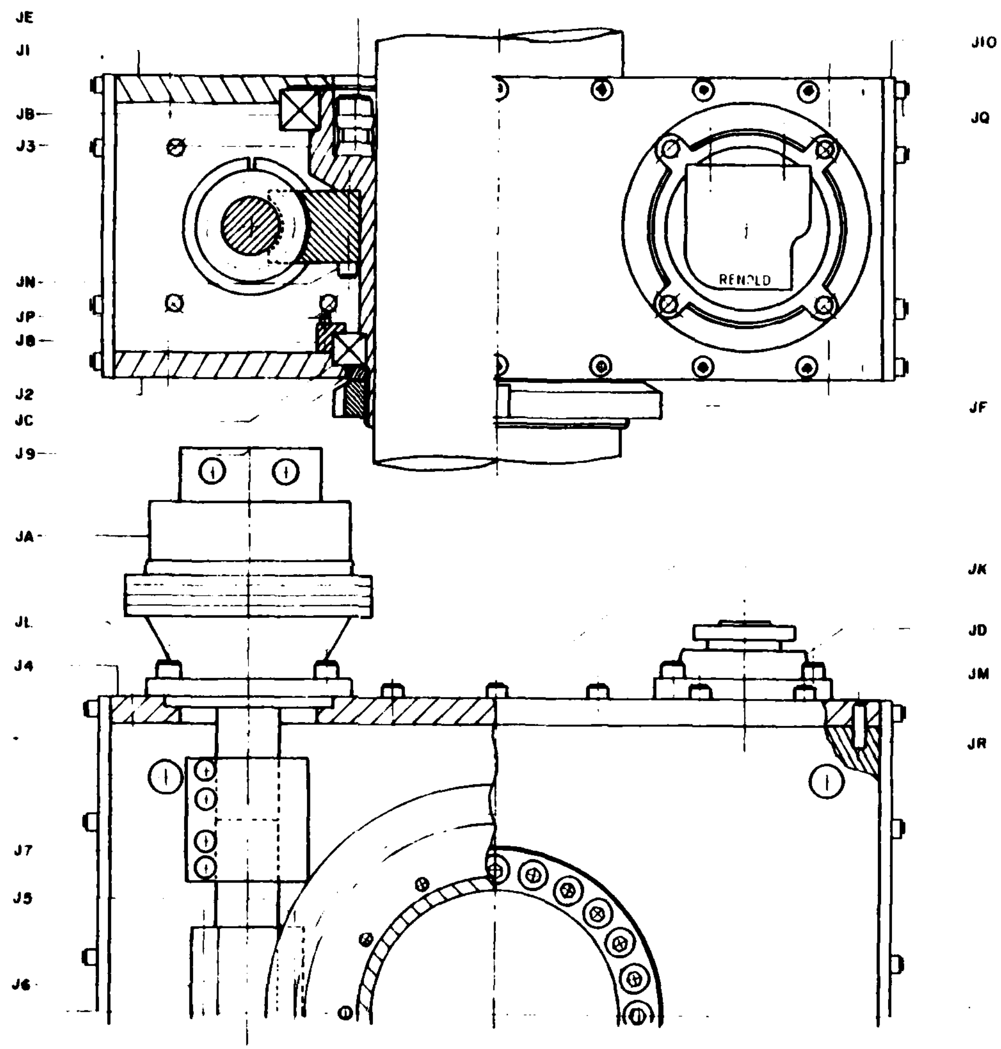

Figure 56. Assembly of the static actuator.

KEY TO FIGURE 56

ITEM	QUANTITY	DESCRIPTION
J1	1	Top Plate.
J2	1	Bottom Plate.
J3	1	Bearing Sleeve.
J4	2	Side Plate.
J5	1	Worm Wheel.
J6	2	Worm Piston.
J7	2	Shaft Coupling.
J8	1	Lower Bearing Retaining Plate.
J9	1	Bearing Nut Spacer.
J10	2	Cover Plate.
JA	2	Hydraulic Motor (worm drive motor), 20 KW radial piston motor type HM05R. Supplier - Reynold Limited, Carter Hydraulic Works, Thornbury, Bradford BD3 8HG, UK.
JB	1	Deep Groove Ball Bearing (upper support bearing), 260 x 320 x 30. Supplier - SKF ref 61852.
JC	1	Deep Groove Ball Bearing (lower support bearing), 190 x 240 x 25. Supplier - SKF ref 61838.
JD	2	Flange Mounted Self Aligning Bearing (worm pinion Shaft end), 40 shaft dia, Y flange, SKF ref FY40FJ.
JE	1	Expanding Friction Coupling (torque tube locking coupling) 180 x 235 type RFN 7012. Supplier - Ringfeder Limited, Forum Drive, Midland Industrial Estate, Rugby, Warwickshire CV21 1NT, UK.
JF	1	Bearing Retaining Nut (lower support bearing - retaining), M190 x 3, SKF ref KM38.
JK	28	Socket Cap Screw (side plate retaining), M10 x 40 HTS.
JL	8	Socket Cap Screw (motor retaining), M12 x 30 HTS.

KEY TO FIGURE 56 (Continued)

ITEM	QUANTITY	DESCRIPTION
JM	8	Socket Cap Screw (flange bearing retaining), M12 x 30 HTS.
JN	6 + 6	Socket Cap Screw + 10 dia Dowels (alternate positions - worm wheel retaining) M8 x 70 HTS, 10 x 40 dowel.
JP	6	Socket Cap Screw (lower bearing plate - retaining), M6 x 35 HTS.
JQ	32	Socket Countersunk Head Screw (cover plate retaining), M6 x 35 HTS.
JR	8	Dowels (side plate locating), 10 dia x 40.

This bearing, being a spherical shell double row ball type, is tolerant of slight angular shaft misalignment, and the bolted attachment method allows for setting up during assembly.

There are a number of advantages to using a worm and whee. drive system in the static actuator:

(1) Because of a relatively high torque inefficiency the unit will not run back under the reaction torque applied by the load. This simplifies the design of the control system for the unit, and is a desirable safety feature.

(2) The dual-drive worm reduction gear system has the lateral forces acting in one plane. This allows the system to develop a pure torque couple and eliminates the need to restrain the actuator against reaction torques about axes other than that of the main drive torque.

(3) The system readily provides a large central through bore for fitting over the torque tube. It provides an inherently high gear reduction ratio and high torque output capability. The assembly is compact, relatively lightweight, and extremely sturdy.

(4) The drive system is inherently bidirectional with unlimited output rotation.

As shown in Figure 56, torque from the worm wheel is transmitted directly to a central hub or sleeve (J3, Fig. 56) within the actuator frame. The worm wheel is suspended by upper and lower mounting bearings (JB and JC, Fig. 56) and makes a sliding fit over the torque tube. The Ringfeder friction coupling (JE, Fig. 56) is used to lock the sleeve onto the torque tube. It is located in a readily accessible recess in the top of the sleeve. The coupling is activated by an array of 24 socket-headed M14 draw bolts. A ratchet spanner with a hex key attachment would be a convenient tool for this purpose. The Ringfeder friction coupling draw bolt system used with the static-actuator transmits a higher torque than the system used with the dynamic actuator. It is for this reason that the draw-bolt method of activating the coupling is recommended and that a hydraulically actuated system such as that designed for the dynamic system is not proposed.

The fact that the Ringfeder coupling is acting at higher loads leads to another disadvantage for the static actuator in that it cannot be clamped onto the thin-walled secton of the torque tubes. As mentioned in Section II-2a, the torque tubes are not designed to cope with loads considerably higher than the design loads required by the in situ BSD probe. Calculations indicate that the hoop stresses induced by the expanding coupling transmitting torques of about 10^4 N.m are well above the loads that would cause the tube to fail. This is in addition to the effects of the stresses that result when the torsional loads are applied. Consequently, the static actuator can only be attached to the ends of the torque tube units where the extension couplings are fitted. The extension couplings are considerably stronger than the plain section of the torque tubes and will easily withstand the stresses induced by the friction locking coupling. If the fact that the actuator can only be attached at 2-m intervals along the torque tubes gives rise to problems, it is suggested that this may be overcome by using more, short tubes. If, in addition, activating and subsequently releasing the locking coupling is found to be inconvenient in the field, it is worth remembering that a simple adaptor could be made to mate the static actuator with the torque tube extension couplings. This adaptor could be permanently secured to the actuator by the friction locking coupling, thereby providing a simpler means for attaching the actuator in the field and a simpler and more convenient extension coupling system.

Out of balance forces in the worm drive system will give rise to lateral forces on the central sleeve component. To avoid high stresses in the coupling of these forces between the sleeve and the torque tube, the sleeve is designed to provide adequate bearing length in relation to the diameter of the tube and to be a snug sliding fit over it. Forces are coupled into the structure by upper and lower bearings which also position the worm wheel in the drive system. Since there is no requirement for very precise axial positioning of the sleeve in the chassis, the bearings are not pre-loaded. The lower bearing (JC, Fig. 56) provides location within the assembly. It is secured in a machined seating recess in the lower plate of the chassis (J2, Fig. 56) retained by plate (J8, Fig. 56) and secured against a shoulder on the central sleeve by the J9 spacer and JF bearing nut shown in Figure 56. The deadload of the actuator is thus carried by the lower bearing and, as can be seen from Figure 56, the upper bearing serves only to provide lateral restraint in the system. This unit, located by the machined recess in the upper plate (J1, Fig. 56) and a stepped diameter on the central sleeve, provides a locating fit in the assembly.

The central structure or chassis of the actuator is a compact module that is fully detachable from the reaction frame assembly. Consisting of a box configured in 20-mm-thick steel plate, the assembly is located by dowels and bolted together. To provide high shear stiffness under the loads imposed by the reaction frame and the drive system, the end faces of the top and bottom plates are bolted to the vertical faces of the side plates J4 and retained by a generous system of through bolts (JK, Fig. 56). The reaction frame can be attached to either the upper or the lower plate of the structure and provision is made for four 25-mm-diameter through bolts. The plane box exterior of the structure and the positioning of the through bolts allows an uninterrupted span over the structure for the reaction frame members. This has the advantage of eliminating the need to joint the frame in the center.

The reaction frame designed for the actuator essentially acts as a double cantilever system. The cantilevers are configured from two continuous lengths of hollow rectangular steel, with short cross spans positoned at the ends and at the points of attachment to the central drive unit. Torque reaction for the system can be provided by one or both of two means:

(1) By a ground anchor system.

(2) By a deadload friction system (i.e., heavy vehicle or ballast load arrangement).

Either arrangement can be used with the actuator which is designed to tolerate some movement in the reaction system. If the reaction frame cannot be directly attached to be made to bear against the anchoring system, then a system of steel chain or cable should be used. As shown in Figure 55, convenient points for attaching the lines are the lifting handles on the frame. These are solid bar handles and are firmly secured to the structure.

c. The Control and Drive System The use of positive displacement hydraulic motors as the drive units in the static actuator affords a number of advantages:

(1) They provide direct drive in high torque, relatively low speed applications.

(2) They are highly efficient and make good use of the available hydraulic power over a wide operating speed range.

(3) Operating in parallel on a common pressure supply, they can produce a balanced drive system.

(4) They are intrinsically safe, extremely rugged, and compact.

(5) Relatively simple hydraulic circuitry provides convenient speed and torque control.

(6) The power source most likely to be available on-site is hydraulic power. Due to their inherent efficiency hydraulic motors provide useful outputs over a relatively wide range of supply pressure.

As mentioned previously, the high proportionality of torque to pressure provided by the hydraulic motors is used to provide a balanced drive in the actuator system. Control of the torque or operating speed produced by the system is achieved by an external hydraulic circuit, shown in the upper portion of Figure 55. Direction of rotational drive is controlled by a three-position change over valve. In the extreme operating positions, the valve reverses the flow through the motors, and in the center-off position, isolates the motor circuit. Speed control is controlled by a flow control valve in the feed line to the reversing valve. It is recommended that the valve be a pressure compensated type to minimize variations in speed caused by load changes. Maximum torque is controlled by limiting the system pressure with a relief valve connected across the supply. For driving at constant speed the relief valve would be set to a safe limiting torque (as indicated on the system's pressure gauge), and the desired speed would be set using the flow control valve. Direction of rotation of the system in Figure 55 is controlled manually. If the system were to be automated, say, to respond to a predetermined program of load cycles under the control of signals from the torque transducer in the BSD, then an alternative arrangement to the three position valve would be required. Automatic control could be achieved by eliminating the center-off position of the reversing valve making it a two position device, and using a solenoid operated unit. Here the direction of flow could be reversed by electrical signals derived from a control and sequencing circuit. A master-off or shutdown control would still be required in the circuit but this could be achieved simply with another two position valve feeding the solenoid operated reversing unit.

The motors recommended for use in the static actuator system have a design output of 10^4 N·m (assuming a worm drive torque efficiency of 40 percent) at a supply pressure of 20 MPa (200 bar). Maximum rated speed at this pressure is 400 r/min which, for the gear worm reduction ratio of 30:1 gives a maximum rotational speed at full load of 13.3 r/min (1.4 rad/s). The no load maximum rated speed of the motors is 750 r/min giving a corresponding maximum output drive speed of 25 r/min (2.62 rad/s). The volumetric displacement of each motor is 0.15 l/rev, giving a total maximum flow of 7.5 l/min at zero load and 4.0 l/min at the maximum speed for full torque. The maximum hydraulic power consumption (at full torque and at the maximum full load speed) is approximately 40 kW.

APPENDIX A

THE TORQUE AND NORMAL LOAD TRANSDUCER CONSTANTS

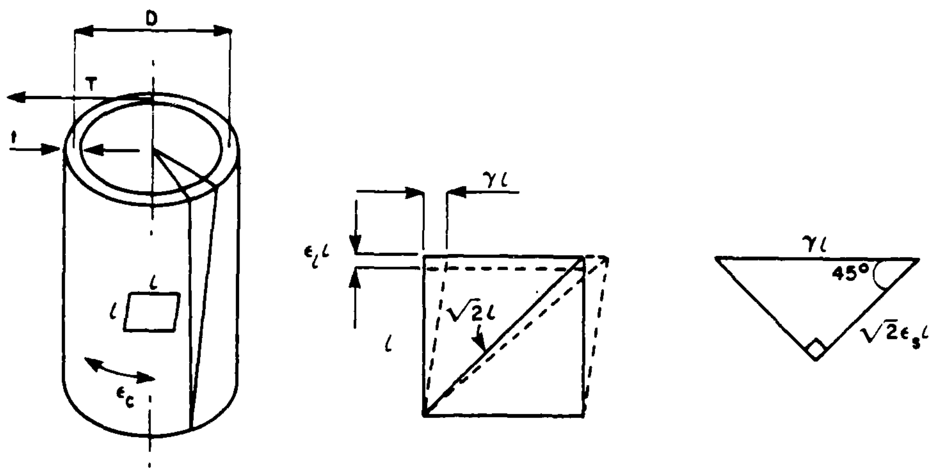

Figure A1. Representation of the gauge tube.

In the figure above let:

F = uniformly distributed normal load

T = applied torque

τ = stress in simple shear due to T

γ = strain corresponding to τ

ε_ℓ = longitudal strain due to F

ε_c = circumferential strain due to the effect of Poissons ratio

ε_s = strain due to T apparent at 45 degrees to direction of ε_ℓ

D = mean diameter of tube

t = wall thickness of tube

G = shear modulus of tube material

E = Youngs modulus of tube material

ν = Poissons ratio of tube material

For the configuration of Figure A1 assume $t \ll D$. The following equation can then be written:

$$\tau = \frac{2T}{\pi D^2 t} \tag{A1}$$

Substituting for τ in this equation from the basic relationship $\tau = G\gamma$ gives:

$$\gamma = \frac{2T}{\pi D^2 tG} \tag{A2}$$

As can be seen from the square element of side length ℓ in Figure A1, the strain in elongation or contraction due to T at 45 degrees to the axis of the tube is:

$$\varepsilon_s = \frac{\gamma}{\sqrt{2}} \cos \frac{\pi}{4} = \frac{\gamma}{2} \tag{A3}$$

substituting in E2 gives:

$$\varepsilon_s = \frac{T}{\pi D^2 tG} \tag{A4}$$

The longitudinal strain ε_c is given by:

$$\varepsilon_\ell = \frac{F}{\pi DtE} \tag{A5}$$

and the corresponding circumferential strain is:

$$\varepsilon_c = -\nu\varepsilon_\omega \tag{A6}$$

Let a system of strain gauges of gauge factor G_f be attached to the tube and connected into two separate wheatstone bridge circuits corresponding to the torque and normal load transducers. The arrangement is shown in Figure A2:

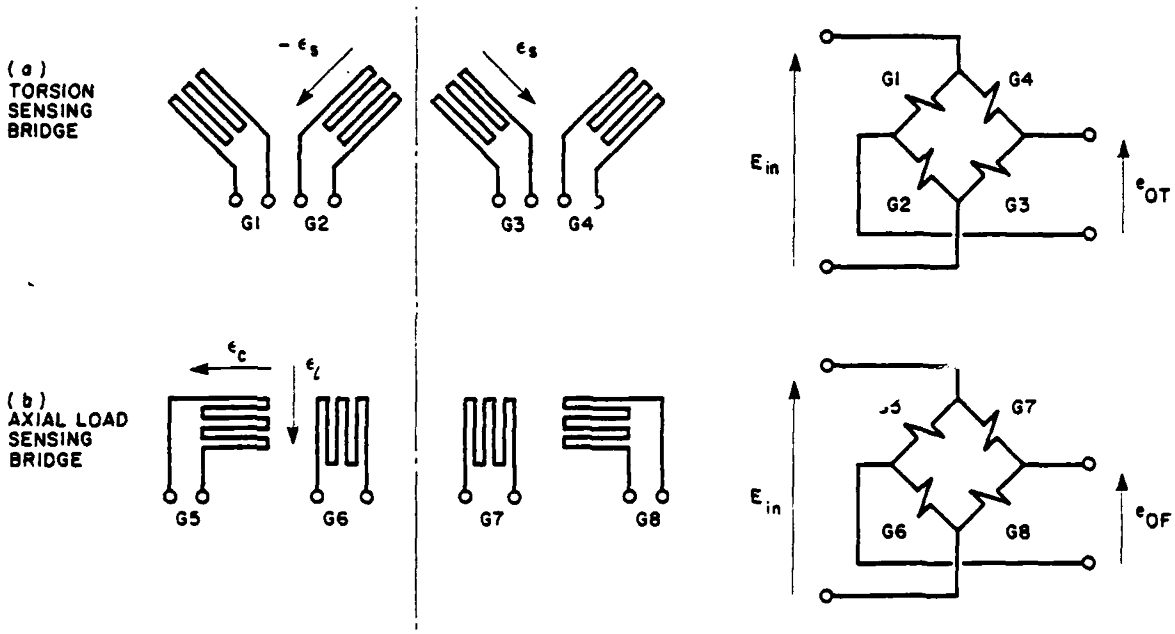

Figure A2. Configuration of the strain gauge bridges in the torque/normal load transducer.

Since gauges G_1 to G_4 in the figure above are subject to the same magnitude of strain, the output e_{ot} from the torque transducer bridge is:

$$e_{ot} = 2\varepsilon_s G_f E_{in}$$

which, from Equation A4 gives:

$$\frac{e_{ot}}{E_{in}T} = \frac{2G_f}{\pi D^2 tG} \qquad \text{(A7)}$$

For the normal load transducer, the circumferentially sensing gauges are subject to a smaller strain than the axial gauges. The output of this bridge is:

$$e_{of} = \varepsilon_{\ell} G_f (1+\nu) \qquad \text{(A8)}$$

which from equation A5 gives:

$$\frac{e_{of}}{E_{in} F} = \frac{G_f (1+\nu)}{\pi D t E} \qquad \text{(A9)}$$

For the gauge tube section of the BSD probe

$$D = 177.5 \text{ mm}, \; t = 2.5 \text{ mm} \qquad \text{(A10)}$$

Assuming $G = 7.96 \times 10^{10}$ Pa, $E = 2.07 \times 10^{11}$ Pa, $\nu = 0.3$ and $G_f = 2.0$, the sensitivities for the torque and normal load transducer systems is 2.03×10^{-7} V/V/Nm and 9.01×10^{-9} V/V/N. At 5 V energization the outputs are 10.15 mV for 10^4 N.m and 9.01 mV for 200 kN.

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