COLLEGE OF AERONAUTICS

(Proposed Cranfield Institute of Technology)

DEPARTMENT OF PRODUCTION ENGINEERING

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REPORT OF CONTROL SYSTEMS COMMITTEE

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SUMMARY

Starting from the initial concept of a small Numerically Controlled lathe, a measurement and actuation system has been developed.

An "absolute" fluidic control system with rotary and linear encoders has been chosen and designed.

Hydraulic actuation for spindle and axis drives has been chosen, the spindle having a hydrostatic drive system involving no mechanical gearing.

An estimation has been made of the cost of the system.

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C.S.1 Introduction

Terms of Reference of the Committee

To investigate suitable control and actuation systems. To decide on the best control and actuation system for the machine under consideration. To design the final adopted control and actuation system.

To achieve the objective given above, an investigation into the type of control system to be used was first carried out. The committee was then divided into control logic and control actuation sections, and two more members taken on, to investigate in detail the chosen system and produce a final design.

It was felt that, at present, a linear path system was the most complicated numerical control system to which a feasible fluidic control could be applied. Costing of the system at the decision stage was difficult, but it was thought that a fluidic system could be produced more cheaply than its electrical or electronic counterpart, especially when taking a long term view. The advantages of predictable delivery dates and costs when supplying one's own system were not overlooked, and the marketing of the control system separately was a possibility.

The use of fluidics as the control medium posed certain difficulties. The number of fluidic interface devices on the market is limited and many aspects of the technology are unproven. However, wherever possible, currently available commercial units are specified and all components are viable propositions.

C.S.2 Determination of the Type of Control System

System Specification

The control system is to be of the Linear Path type i.e. to be able to move to a position without overshoot and at a controlled feedrate.

Specification Derived from the Technical Survey

Accuracy	± .001"	(± .025 mm.)
Maximum diameter of workpiece	4"	(10 cm.)
Maximum length of workpiece	8"	(20 cm.)

C.S.2.1 Lathe Control Systems - General

Types of system [1]

Analogue	Chain dimension
	Reference Datum (Co-ordinate system)
Incremental - digital	Chain dimension
	Reference Datum.

Absolute Digital

On lathes, a reference datum system recommends itself because turned parts already have a natural reference axis in the form of the centre line. Diameters are invariably represented as absolute dimensions, and length dimensions are conveniently referred to a face, although lengths could be chain dimensioned.

Further advantages of reference dimension input are:-

1. Tool offset is possible over the complete range.
2. Position readout is readily available with an absolute digital control (but not with an analogue reference system).
3. The machine datum is in the measurement system.
4. There is no difficulty with programme interruption and mains supply failure.

It is suggested that an absolute reference system is best suited to a turning machine.

C.S.2.2 The Linear Path System

The original specification of a linear path control was considered a suitable one for application to a lathe. The majority of movements on a lathe are either longitudinal or transverse. This is supported by an investigation at

1 See Section C.S.2.7.

Messrs. Gebr. Boehringer [1] in Germany, which showed that 80% of their turned parts could be produced with a linear path, or straight-cut machine.

The original specification suggested by the main Board for the machine included the ability to produce tapers, threads, helical milled grooves and other configurations involving a precision control between either the movement of the two axes, or between the headstock rotation and longitudinal traverse.

Feedrate control in most linear path control systems is achieved by introducing velocity feedback into the control loop. Indeed many Linear Path or "Straight Cut" systems are produced by adapting "Point to Point" systems in this way. Certain refinements are also added to prevent overshoot.

The accuracy of the velocity control system, being normally an analogue feedback loop, is of the order of ± 5%. More refined systems are of course available but in order to produce a taper of accurate form the refinements are such that the system approaches a continuous path control in complexity and hence cost.

Thread-cutting poses similar problems if the relevant axis moments are to be interlocked using the control system. The principle is to control the traverse movement with respect to the headstock rotation instead of with respect to time, this requiring a system refinement not normal with linear path control. It is significant that the G.E.C. Mark Century threadcutting system can only be used on their contouring controls, since function generators are required. The problem would be removed if leadscrew actuation were employed, but this necessitates the inclusion of a gearbox; an expensive and space-consuming item with the complication of tape commanded gear changing and, ideally, a separate precision leadscrew. Leadscrew actuation, however, is a strong possibility if a bought out system is used as these tend to be the cheapest systems.

Conclusions

Taper turning and threadcutting using the control system are not feasible for a linear path system. Threadcutting can be performed using a leadscrew and gearbox, or by using die-boxes for threadcutting or threadrolling.

Enquiries are being made about Baldwin Threadcutting Encoders, although it is not envisaged that these will be used.

[1] See Reference 1.

C.S.2.3 Investigation of Control Systems

It was decided that the original investigation of the systems should be divided into two sections:-

Sub-committee A Electrical, Electronic, and Hydraulic Systems.

H. Bera
A. Melsom

Sub-committee B Fluidic Systems.

M. Norton
V.A.M. White

Electrical, Electronic, and Hydraulic Systems

Enquiries were made to the five major control system suppliers in Britain.

I.G.E.
Ferranti
Airmec-A.E.I.
Plessey
E.M.I.

It was the view of the committee that in addition to buying out a standard or custom built complete system, the feasibility of building a system using bought out measuring elements and other suitable components should be considered.

The machine tool manufacturer is well catered for in terms of control systems. All firms expressed the importance of liaison between the manufacturers of the machine tool and the control system so that full advantage could be taken of more economical standard modules, and so that machine and control system were suitably designed.

A custom built control could thus be readily obtained with a large number of features and optional extras available. In view of what was decided previously in terms of taking full advantage of the relative simplicity and low cost of a linear path system, the committee recommended that a two axis Airmec system be used. A brief specification and quotation for this system is given below.

The manufacture of a control system from bought out parts was a less attractive idea. The present electronic and electrical systems are highly developed, and the committee found no economically attractive schemes. Plessey were willing to provide "Inductosyn" scales and synchro-resolvers, and I.G.E. said that their measuring system was available separately. However, it is possible that the inclusion of certain devices in a control system would infringe patent rights. This also applies to certain proprietary thread-cutting and encoder systems.

The indexing of the headstock would be effected using a shaft encoder, and the system interlocks could be readily synthesized using static switching elements. Electrical stepping motors would be suitable for any indexing operations of the tool post or tool magazine.

Recommendations of Committee A

1. That a two axis straight cut system be purchased from Airmec-A.E.I. Ltd.

2. That a shaft encoder be included in the headstock arrangement. A suitable model would be a Moore Reed "Hybrid" encoder which could be suitably geared. This encoder can run at high speed and gives a pure binary output.

3. That with this system electrical actuation be used to take advantage of low cost, and all subsidiary functions be electrical e.g. indexing and interlocks, in order to keep the services to the machine to a minimum.

Control System Specification

Airmec Autoset type N410

Accuracy	± .0002"	(± .005 mm.)
Repeatability	± .0001"	(± .0025 mm.)

Estimated Machine figures for medium grade slideways

Accuracy	± .001"	(± .025 mm.)
Repeatability	± .0005"	(± .0075 mm.)

Fast Traverse rate 120 - 180"/min. (3 - 4.5 m/min.) for a 5 t.p.i. leadscrew.

Positioner Unit type N334

Principle

The system utilises switching circuits between carbon brushes in rotating brush carriers and a fixed disc commutator. Four brush carriers are used with a reduction ratio of 10 : 1 between each. Basically an End Position Indication system with lead/lag information, and additional refinement when fitted with the "Auto Mill Unit".

The positioner units measure absolute angle and consequently sequence interruption or mains failure do not necessitate re-zeroing.

Control System Quotation

1.	Basic Autoset (5 digit)	N410	£2,200
1.	Automill Unit	N411	100
1.	D.C. motor control unit including motors	N419	450
			£2,750

Optional Extras

1.	Inching unit	N414	£ 25
1.	Number display unit	N412	100
1.	Tape spooler	N413	35
1.	Autoset tester	N366B	35

Further requirements are the rotary encoder for the headstock, the tool and work loading actuation, and the machine sequence and interlock system. However since these are available in either electronic, electrical, pneumatic, or fluidic form, it was decided to base the decision of which system to use mainly on the attributes of the two axis positioning system.

Fluidic Systems

The sub-committee reported that a fluidic logic system was a feasible proposition for a paraxial control with the specification of section 1.2.

Recommendations of the sub-committee

1. The system should be of the "absolute" type.
2. The two linear axes should be supplied with similar linear fluidic encoders.
3. A rotary fluidic encoder should be incorporated in the headstock.
4. The measurement system should be a comparator type circuit with either a Gray code read out from the encoder and lead/lag compensation, or a "U" or "V" scan system, again with some form of lead/lag signal to prevent overshoot.

Estimate of Cost of a Fluidic System

It was pointed out that the complete system including the linear encoders would have to be developed and that these development costs should be borne in mind.

However, assuming that the system were to be designed using an estimated 500 fluidic elements per linear axis, a quotation for the elements had been obtained on the basis of purchasing 20,000 elements per year.

Quoted cost of element		12s.	6d each
Cost per axis approx.	£312	-	-
Estimated cost of rotary axis	£100	-	-
Estimated cost of system complete with linear encoders and hydraulic actuation	£2,500	-	-

The sub-committee pointed out that if a manufacturing concern were set up, they would almost certainly produce their own elements with a resulting saving of up to 90% on the cost of the elements.

C.S.2.4 Comparison of the Systems

It was decided that there was little to choose between the two in terms of cost at present. The price of the electrical system was fixed whereas the development costs of the fluidic system could escalate.

Decision of the Control Committee

The committee decided that it would recommend to the board that a fluidic control system be developed for incorporation in the numerically controlled lathe.

Reasons for the Decision

1. Although the fluidic system presented immediate development costs, it promised to provide a cheaper system in the future when the manufacturing concern could produce its own elements and encoders.

2. The manufacturing concern would have an original and patentable control system. The control system appeared to be a viable proposition in its own right.

C.S.2.5 Further Proposals for the Fluidic System

1. That encoding be envisaged to .010" and thereafter a mechanical offset or proportional subdivision be used for the final digits.

2. That velocity feedback be obtained from the fine digit readout or using a.c. fluidics. A fluidic tacho generator should also be investigated.

3. That the tape input be I.S.O. standard binary.

4. That machine interlocks and sequence controls be effected using pneumatic gauging and fluidic elements.

C.S.2.6 References

1. V.D.F. News, No. 30, December 1966.

2. Manufacturers publications on the systems mentioned above.

C.S.2.7 Definition of Terms

Chain dimensioning

In chain dimensioning every target point is dimensioned and programmed relative to the preceding point.

Reference datum dimensioning

Reference datum dimensioning, also known as co-ordinate dimensioning relates target points to a machine-dependent co-ordinate system.

CS.3 Actuation Sub-Committee Report

Chairman A. Melsom

R. Crooks

C.S.3 Actuation Sub-Committee Report

C.S.3.1 The Spindle Drive

A hydrostatic drive motor was chosen for the headstock. Factors influencing this decision were:-

1. The moving headstock of the design made a compact spindle drive desirable. It would be very difficult to incorporate a conventional electric motor and gearbox in the headstock.

2. The requirement of a wide speed range (100 r.p.m. to 3000 r.p.m.), preferably without the use of a gearbox for compactness and ease of control.

3. The presence of hydraulics on the machine.

Advantages of a Hydrostatic Drive

1. High Power/Weight and Power/Volume ratios.

2. Continuously variable speed.

3. High drive stiffness.

4. Ease of control on open or closed loop.

The report on spindle drives (see Appendix C.S.3.1) indicated that the simplest hydrostatic drive system with the required characteristics was the fixed pump/variable motor arrangement. This gave a suitable horsepower/speed envelope and sufficiently good speed holding characteristics to permit open loop control. However, the speed range was limited to four or five to one and the incorporation of a three speed gearbox would be necessary.

The variable pump/variable motor configuration gave a sufficiently wide speed range and suitable characteristics for a machine tool drive. This arrangement is an unknown quantity in terms of commercial applications, although its characteristics have been investigated (see references 1, 2, and 4).

The important factors at this point were considered to be:-

1. The use of fluidic control limited the commercially available components (e.g. interface devices) and possible techniques which could be applied to a control circuit.

2. Efficiency, in terms of power consumption, was not an important aspect. Its effect upon component size and heat generation were more important.

3. Spindle speeds must be selected from the punched paper tape.

The following systems were considered:-

1. A variable displacement pump/variable displacement motor (V.P./V.M.) system utilising a hydraulic servo arrangement to operate the pump and motor swashplates in correct sequence with a pressure feedback from the spindle proportional to spindle speed. The arrangement is shown in figure C.S./3/1/1.

2. A V.P./V.M. system in which the swashplates are indexed through discrete positions to obtain a series of speeds.

3. A spool valve operating in conjunction with a variable displacement motor. This would be a compromise to obtain the required speed range with a reasonable torque/speed characteristic.

4. Using a series of on/off fluidic/hydraulic valves with a V.M. This was to reduce the cost of 3 above by replacing the expensive fluidic/hydraulic interface valve.

The systems 3 and 4 were discarded owing to the inherent limitation of valves in variable load conditions, i.e. the severe speed droop characteristic. This effect could be minimised by limiting the range of valve operation, in terms of command signal and maximum valve pressure drop.

A suitable control system for scheme 1 above is shown in fig. C.S./3/1/2. This complete system would be expensive (estimated at approximately £750) and would complicate the headstock design by necessitating the inclusion of a hydraulic pump. A hydraulic restrictor valve which could be altered from a fluidic signal would also be necessary.

Scheme 2 had two obvious limitations:-

1. It was an open loop system. Since each speed effectively represented a fixed pump/fixed motor configuration, however, it would give speed holding better than 5% which was all that was required.

2. The feature of a continuously variable speed range would be lost. This feature would not really be necessary unless some form of workpiece diameter compensation system were used. The use of an infinite

number of spindle speeds would present programming difficulties.

However, it was thought that using this system a V.P./V.M. system could be included in the machine at a reasonable cost to give the required speed range. The power envelope of the system would be suitable and the speeds would be reasonably easy to programme.

The method chosen for altering the swashplate angle utilised digital actuators made up of pneumatic rams, see fig. C.S./3/1/3 for details. It was thought at first that the use of two rams per unit, giving four positions of the swashplate would be sufficient, as sixteen speeds would result. However, the speed ranges in this case would overlap to a large extent and the available torque would have a saw-tooth characteristic.

The best performance is obtained from the system by altering the pump displacement at the lower end of the speed range, and then the motor displacement. It was therefore decided to have four positions on the pump swashplate and eight on the motor swashplate to give eleven available speeds.

Ideally the spindle speeds should follow a geometric progression as laid down in international standards (see reference 10), but this was not possible with the system used. The changeover point from pump to motor control should occur at such a speed so as to give the two phases approximately equal speed ranges.

$$\text{i.e. Change Speed} = \sqrt{\text{Maximum speed} \times \text{Minimum speed}}$$

$$= \sqrt{3000 \times 100}$$

$$= \underline{550 \text{ r.p.m.}}$$

A proposed speed range is given in fig. C.S./3/1/4.

The motor or pump displacement is assumed to vary linearly with the piston stroke. This is a reasonable assumption as demonstrated in fig. C.S./3/1/4.

An estimate of the cost of this actuation is included in fig. C.S./3/6/1Aand it can be seen that this is an economical way of effecting the swashplate deflection.

The arrangement of the jacks with their valving is shown in fig. C.S./3/1/5.

Noise

Hydraulic units tend to be more noisy than the equivalent electrical unit. If the motor or pump can be isolated from sound emanating surfaces in terms of vibration transmission, this noise will be greatly reduced. The mounting of the pumps and motors in the machine frame should also reduce noise.

Temperature

Owing to their high power/weight ratio hydraulic units run at high temperatures unless precautions are taken.

Oil coolers can be introduced to reduce the temperature and a large enough cooler would reduce the temperature to near ambient. This is not practical however in terms of cooler size. Two solutions are possible

a) Cool the oil to give an operating temperature of approximately 40°C and insulate the power pack and spindle motor from the machine frame structure so that very little heat is transmitted to the frame.

b) Have all units in close contact with the machine frame so that a constant working temperature is soon reached. On start-up, the oil from the pumps can be dumped straight through a valve to decrease the warming up period for the oil.

Inching of the Spindle

To position the spindle using the encoder it is necessary to rotate the spindle at 1.5 r.p.m.

To achieve this, when the tape calls for a creep mode the pump displacement is set at minimum, the motor displacement at maximum and a low pressure relief valve is connected from the pressure line by opening an on/off fluidic/hydraulic valve. The relief valve can be adjusted to give the required creep speed, and when the spindle is locked there is no build up of pressure.

This arrangement is shown on the diagram of the hydraulic circuit, fig. C.S./3/6/1.

Swashplate actuation calculations

See figs. C.S./3/1/3; C.S./3/1/4

Linear movement required at 3.75" radius = 1"

Torque required to index swashplate = 450 lb.ins.

Torque available from pneumatic cylinders = 300 x 3.75

= 1125 lb.ins.

Tape input to pneumatic cylinders

On an eight bit tape the system requires part of one row.

The pump requires two bits.

The motor requires three bits.

The indexing or creep signal requires one bit.

C.S.3.1 APPENDIX

Variable Speed Drives

Factors

1. Horsepower speed envelope.
2. Speed range.
3. Torque-speed characteristic.
4. Efficiency.
5. Cost
6. Ease and flexibility of installation.

A. A.C. Induction Motor

1. Variable Slip

Involve speed variation by energy dissipation. A small speed range is necessary for reasonable efficiency (2 : 1)

2. Variable Supply Voltage

Limited range (2 : 1).

3. Variable Pole Pitch

Speed range 3 :1.

4. Commutator Motor with its own Frequency Changer

Speed range 4 : 1.

5. Single Phase a/c Motor with Control Gear

Speed range 6 : 1. (Single phase machines tend to be heavier and less efficient than poly-phase machines).

B. D.C. Electric Motor

Variation of either the flux density or the armature voltage, the latter being preferred.

Speed range of 20 : 1 plus.

The torque-speed characteristics can be varied widely.

The D.C. supply can be obtained from a rotating converter (Ward Leonard System) or from static rectifiers.

Require auxiliary equipment and control gear.

The maximum armature current of a D.C. motor is limited - the available torque is almost constant over the speed range.

The torque/unit weight ratio in an electric motor has a fundamental limitation in the saturation flux density of the armature material.

C. Mechanical Drives

Belt, friction, impulse drives.
Speed range limited - 12 : 1 maximum.

D. Hydraulic Drives

Hydrokinetic Drives

Unsuitable owing to torque/speed characteristics.

Hydrostatic Drives

High torque/weight ratio - can be an order of magnitude greater than that of an electric motor.

1. Valve controlled drive

(a) Four way valve

Maximum output pressure = $\frac{2}{3}$ Ps

Under normal conditions efficiency $\ll$ maximum, and consequently
(i) Size of unit is increased.
(ii) Large heat production.

Torque Speed Envelope

See fig. C.S./3/1/1A

(b) By-pass type variable flow control valve

See fig. C.S./3/1/2A.

The system pressure rises to meet load demand.

The torque is more or less constant over the speed range.

2. Pump displacement controlled drive

The pump is normally driven at a constant speed.

Ideally

(a) The whole of the power output from the pump is utilised.

(b) If the maximum pressure is limited, the ideal torque is constant over the speed range, and the motor speed is independent of load torque.

Departure from the ideal in practice is considerable owing to friction and leakage.

3. Motor displacement control

For a limited system pressure the Horse Power is ideally constant over the speed range. The ideal efficiency is 100% and the motor speed is independent of load torque. Losses again cause deviation.

The available speed range is limited by efficiency considerations to 6 : 1.

4. Variable pump and motor control

The first part of the speed range is achieved by varying the pump displacement at maximum motor displacement and thereafter by varying motor displacement.

Speed range 30 : 1.

Expensive, but the size of unit required for a given speed range with a specified maximum torque is less with this arrangement than with any other type of hydraulic drive.

5. Hydro-mechanical transmission

Have a higher efficiency than an all hydraulic drive. Generally used where the drive has a ratio of 1 : 1 for most of the time.

A Machine Tool Drive

The power-speed envelope requirement of a machine tool varies with the different uses to which it is put.

With steel and cast iron, if depth of cut, feedrate, and cutting speed remain constant, the power required for machining one material is virtually independent of spindle speed.

A constant horsepower envelope thus recommends itself. See fig. C.S./3/1/3A.

The transmission losses tend to increase with speed. At low spindle speed, the maximum torque is limited by the strength and rigidity of the transmission components, and thus the torque could well be limited over this range which is mainly used for taking heavy cuts on large diameter workpieces.

Hydraulic Drives - advantages

1. High Power/Weight Ratio

A hydraulic motor can be mounted directly in the headstock of a lathe, which is particularly advantageous if a power source is required in a moving member.

2. Stepless Speed Variation

Not often required.

3. Control

Hydraulic drives lend themselves to automatic control systems, which is important in an N.C. machine.

4. Horsepower - Speed Envelope

Range limited by efficiency, but a variable displacement motor can give constant horsepower over a range of 5 : 1.

5. Speed Holding

Open loop speed holding of a displacement - controlled motor drive can be 5%. Closed loop systems are easy to apply.

Disadvantages

1. High Temperatures

High power/weight ratio necessitates an oil cooler.

2. Efficiency

Low efficiency attributable generally to the large number of energy conversions limits the speed range and produces excessive heat.

3. Noise

Hydraulic units are noisier than electrical/mechanical systems, but development will probably reduce this.

4. Cost

Hydraulic systems are generally more expensive than electrical systems.

A simple FP-VM drive of 10 H.P. would cost £500.

A VP-VM drive with closed loop control would cost in the order of £1,000.

5. Dynamic Performance

Low speed interrupted cut performance can be poor - not so important on a lathe as on a milling machine, caused mainly by the high torque/weight ratio.

Types of Hydraulic Drive

1. Variable Pump - Fixed Motor

The torque is approximately constant, and thus power increases with speed.

As has been stated earlier, this is not ideal for a machine tool, but is more suitable in the lower speed range.

Speed holding with pump displacement control deteriorates as initial speed is reduced. A closed loop control is necessary.

At low speed, dynamic behaviour problems are accentuated by the low inertia of the motor.

2. Fixed Pump - Variable Motor

The horsepower is approximately constant over the speed range, which is desirable for a machine tool.

Open loop torque-speed characteristic can be sufficiently good to make a closed loop system unnecessary, an improvement in cost, complexity, and reliability.

A speed range of 5 : 1 with a three speed gearbox would give a range of 125 : 1.

3. Variable Pump - Variable Motor

Has a suitable power/speed curve. See fig. C.S./3/1/4A.

Expensive, and the speed range would probably be insufficient, however, a compact epicyclic gearbox could be used.

This method is more attractive for a numerically controlled machine tool where speeds are demanded from tape. The pump displacement controlled speed range could be used for inching.

General

The forces required to alter the displacement setting of a variable displacement unit are large and a position servo is often used - this is suitable for automatic speed control.

A hydraulic speed control servo could be employed since accuracy is not critical. These tend to be cheaper.

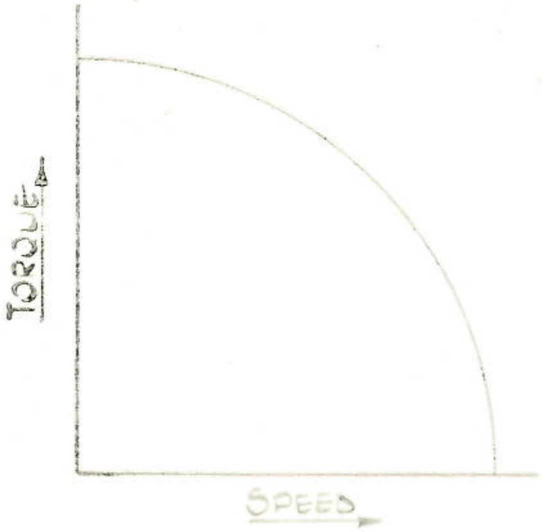


FIG CS/3/1/1A

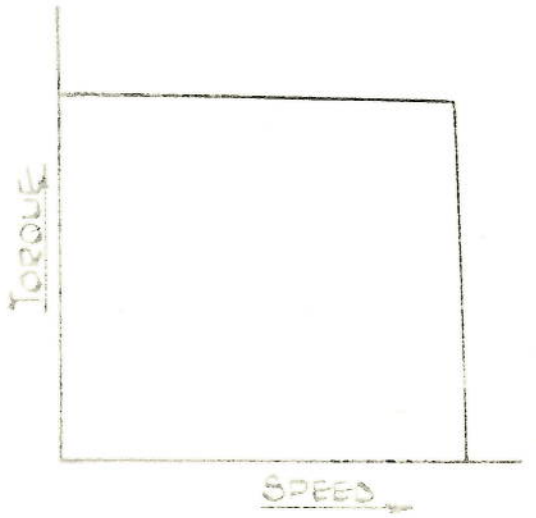


FIG CS/3/1/2A

TYPICAL CHARACTERISTICS OF A VALVE CONTROLLE
DRIVE

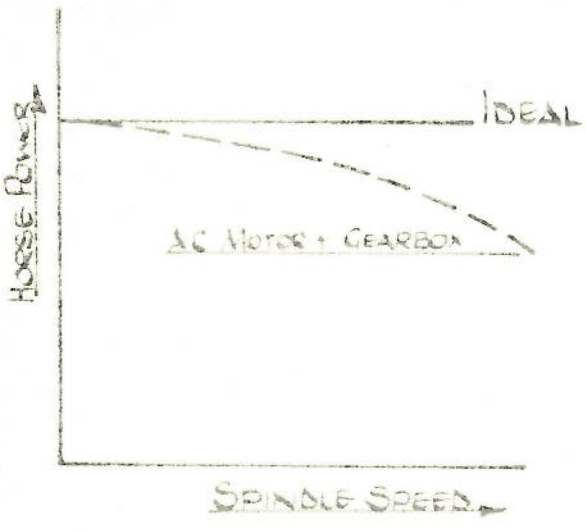


FIG CS/3/1/3A

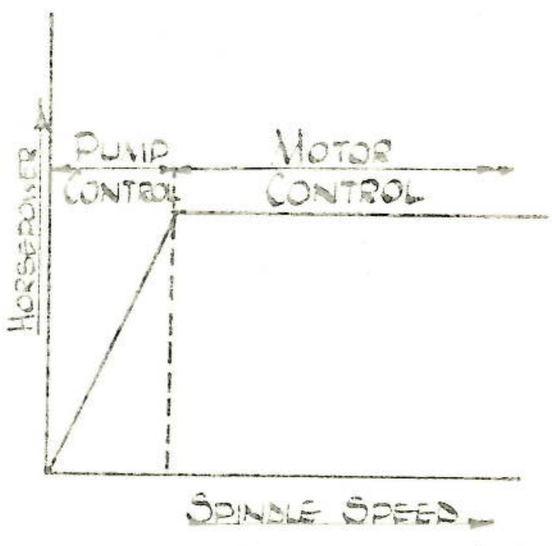
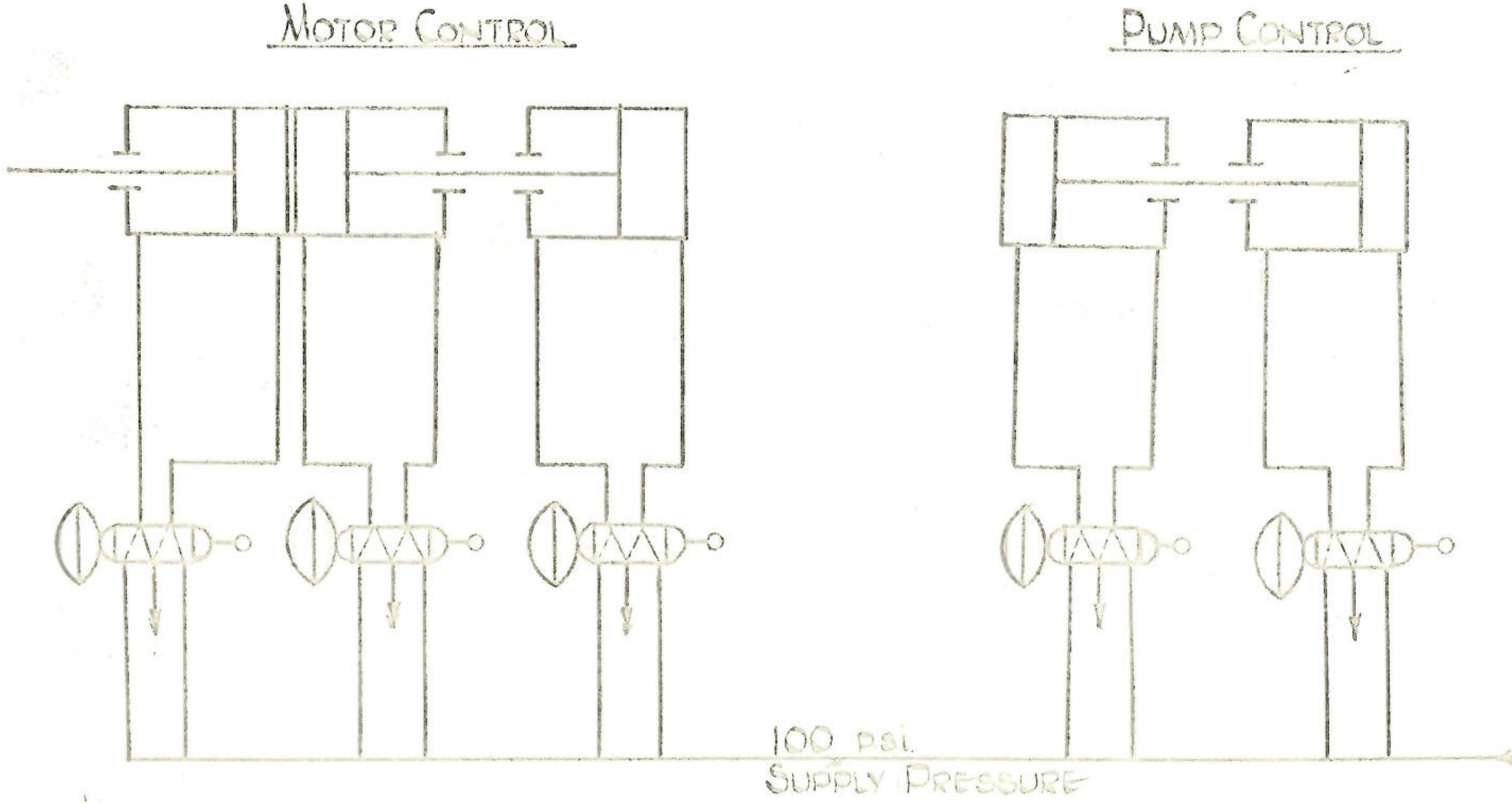


FIG CS/3/1/4A

VARIABLE PUMP/VARIABLE MOTOR

HORSEPOWER/SPEED CHARACTERISTICS



SCHEMATIC DIAGRAM OF SWASHPLATE ACTUATION SYSTEM FIG CS/3/1/5

C.S.3.2 Slideway Actuation

Choice of Actuation.

The two suitable forms of axis actuation were considered to be:-

a) A leadscrew driver by an electric or hydraulic motor.
b) Hydraulic rams.

Hydraulic rams were chosen for the following reasons:-

a) The strokes on the two axes were of the order of 10" (25 cm.) on the controlled axes.

b) Linear encoders had already been chosen to give an absolute system with the minimum number of error sources outside the control loop.

c) The Design Committee wished to use hydrostatic bearings on the machine slideways making a high pressure oil supply necessary.

d) The existence of a fluidic/hydraulic precision interface in the form of the Moog two stage valve made the use of hydraulic rams a feasible proposition.

Hydraulic actuators are most successfully applied to short strokes up to approximately two feet in length. With greater lengths the compliance of the hydraulic column presents difficulties when precision control is required.

Design of the Actuators

The linear encoders have a minimum bit size of .001" (.025 mm.) although it will position more accurately i.e. ± .0003" (± .0075 mm.). Thus there can be no error signal in the loop until the slide has moved .001" (.025 mm.). The system therefore depends on the ram stiffness for accuracies greater than this.

In roughing operations where very high cutting forces occur, i.e. 500 lb.f. (225 kg.f.) this is not of great importance as accurate work is not undertaken with such conditions. A finishing cut might provide a force of the order of 20 lb.f. (9 kg.f.).

The rams are designed from a stiffness figure of 5×10^5 lb./inch (9×10^4 kg./cm) which is very high for a system of this kind. The resulting large ram area also means that actuating forces are very high compared with cutting and inertia forces.

The high stiffness and low inertia of the sliding assembly means that the system has a high natural frequency.

The damping factor required for stability under these conditions is negligible and will be inevitably exceeded by the inherent damping of the system.

The control system is of the digital type and calculations of the loop gain for stability are not applicable. The signal to the Moog valve can be adjusted by altering the supply pressure to the Digital to Analogue convertors.

Velocity Control of the Axes

To satisfy feedrate requirements the feedrate of the axes had to be controlled to within 5%. Velocity control systems were investigated (see section C.S.4.5).

It was decided that a sufficiently accurate feedrate could be obtained by an open loop system depending upon the flow of oil through the valve associated with each axis. The valve would be temperature and load sensitive, but load forces were very small compared with actuation forces (a maximum value of load forces/actuation forces is $\frac{1}{30}$). By dumping oil through a by-pass valve on starting up the machine, the running temperature could be reached quickly.

See fig. C.S./3/2/1 for the ram specification. The rams are identical for the two axes.

A block diagram of the linear path control is shown in fig. C.S./3/2/2. As explained earlier, the velocity control is of the open loop type, there being no velocity information fed back from the encoders.

C.S.3.3 The Indexing Ram

A ram is required to move the headstock and encoder assembly through a distance of approximately 14" (35 cm.).

Two systems were considered:-

a) Using a locking ram.

b) Using a conventional ram to hold the headstock against set stops.

The first system would prove expensive as the rams would have to be specially made. It was decided to adopt systems (b).

Low compliance of the ram would minimise any movement owing to cutting forces, and the short dwell period which must be programmed at any shoulder on the workpiece would effectively remove this effect. The positioning force would be high compared with cutting forces as in the case of the axis rams.

A five port fluidic/hydraulic valve is necessary so that indexing can be programmed from the tape. Such a valve is available commercially at a reasonable price. (See fig. C.S./3/6/1A).

See fig. C.S./3/2/1 for the ram specification.

C.S.3.4 The Tailstock Actuation

While the tailstock is in operation, it is maintained in contact with the workpiece by the oil pressure in the hydraulic ram. This pressure is limited by a relief valve, the setting of which can be varied for different workpieces. Thus the tailstock will follow the workpiece, and be pushed backwards by the workpiece according as the headstock is moving away from or towards the tailstock.

The tailstock is retracted by admitting flow to the opposite side of the ram.

The forces exerted by the tailstock are small compared with those exerted by the headstock and cutting and positioning will not be impaired. To prevent flow wastage the tailstock ram will be of a minimum diameter and the supply pressure will be reduced by a restrictor, consistent with maximum flow requirements.

The necessary valves are detailed in fig. C.S./3/6/1.

C.S.3.5 The Toolpost Motor

The most important factor in this application is size.

Ideally a constant power drive should be used, and this could be approximated to by a variable displacement motor. However this would be bulky and would require a compact fluidic/hydraulic actuation system to alter the displacement.

It is therefore decided to use a small high speed fixed displacement motor and to alter its speed by regulating the flow with a valve. This gives a constant torque output and so the motor has to be rated in terms of torque at the low speed requirement. This leads to excess power when running at high speed, and hence inefficiency.

However, in a low power application of this type, this is relatively unimportant. The valving could be effected using on/off valves and rated restrictors, or by using a proportional valve commanded from a Digital/Analogue fluidic converter. The only proportional fluidic/hydraulic valve at present available is a high performance servovalve and hence is costly. Such a high performance valve is not necessary, and a less refined valve is specified.

C.S.3.6 The Hydraulic Power Pack

This comprises two main power sources.

(i) The headstock drive system.

(ii) A constant pressure supply for the rams and auxiliary equipment.

(i) The headstock drive is described in section C.S.3.1 and the circuit layout shown in fig. C.S.3.6.1. This is in effect a separate system, but the output from the motor is fed to the common reservoir so as to simplify cooling arrangements.

The boost pump and restrictor in this circuit are to maintain positive pressures of approximately 50 p.s.i. at pump inlet and motor outlet and hence prevent cavitation.

A brief specification of the pump and motor follows:-

Downel model 125 Variable displacement pump.

Displacement	1.25 cu.in./rev.
Input speed	1500 r.p.m.

Headstock Motor

Displacement	4 cu.in./rev.

System Pressure 3,000 p.s.i.

The headstock motor is a modified axial piston unit (see Design Report) and no part number can be quoted. Figures needed in the design of the circuit were obtained by inspection of data for the Downel motor model 325.

(ii) The pump for the axes and auxiliaries is identical to that specified in section (i) above with the addition of a constant pressure servo.

The supply pressure is 3,000 p.s.i.

C.S.3.7 Cost of the Actuation

See fig. C.S.3/6/1 for details.

The total cost of the actuation system is estimated at £1,300.

C.S.3.8 Recommendations for Further Work

1. An investigation into the most suitable system for the toolpost motor might provide a superior system.

2. Developments in hydrostatic drives, e.g. ball motors, could provide an improved spindle drive system of an even more compact form.

3. In the future, the techniques used should be reviewed in the light of available techniques and equipment.

C.S.3.9 References

1. PERA report No. 116

2. PERA report No. 151

3. PERA report No. 155

4. "Hydrostatic Lathe Spindle Drive" - N.E.L.

5. Hydraulic Systems - D.F. Williams, Dowty Rotol Ltd.

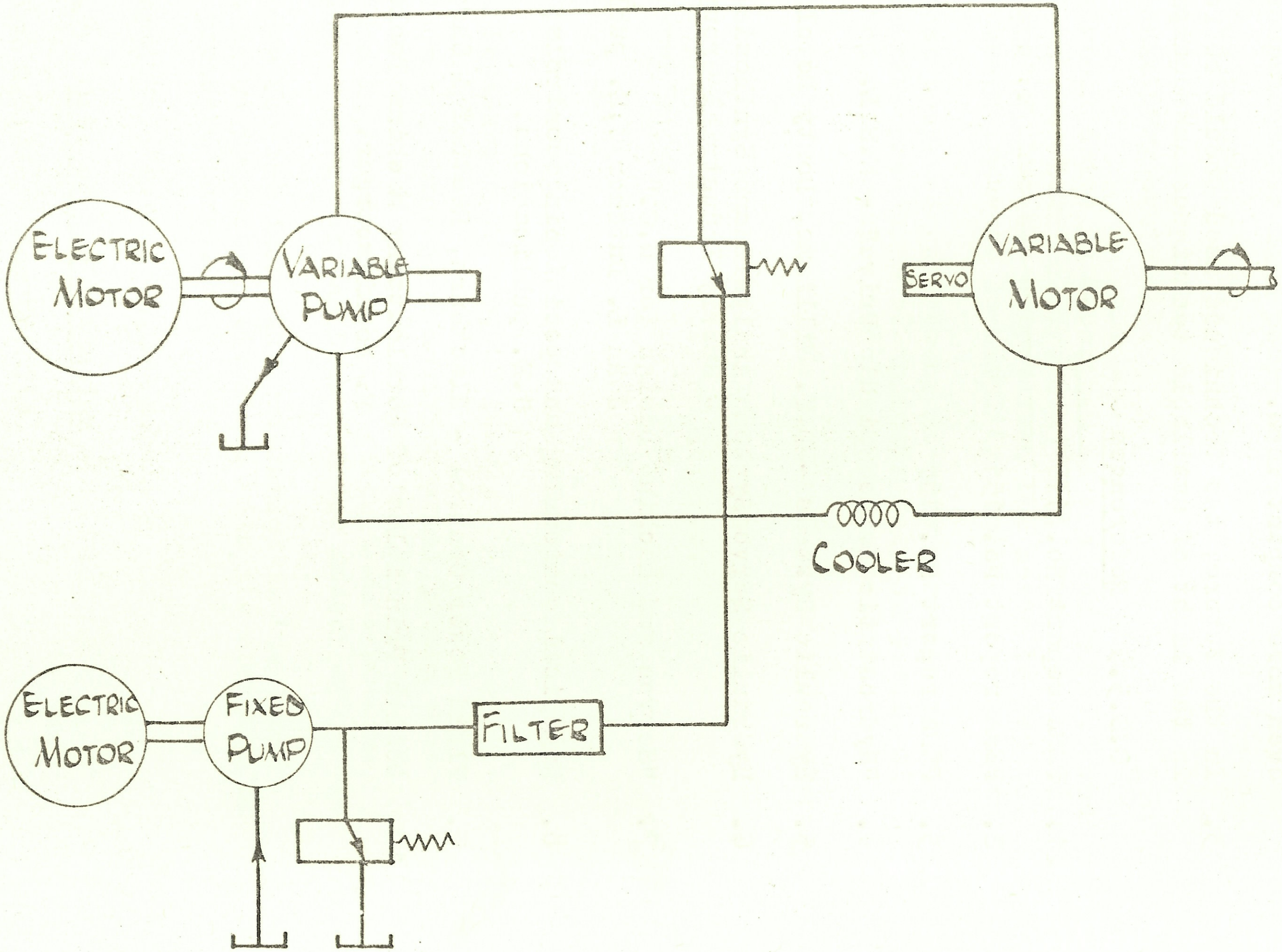
6. Hydraulic Servo Systems Analysis and Synthesis - Sperry Industrial Group.

7. "Servos and Machine Design for N.C." John L. Dutcher/G.E. publication.

8. Numerical contouring and positioning controls, G.E. publications.

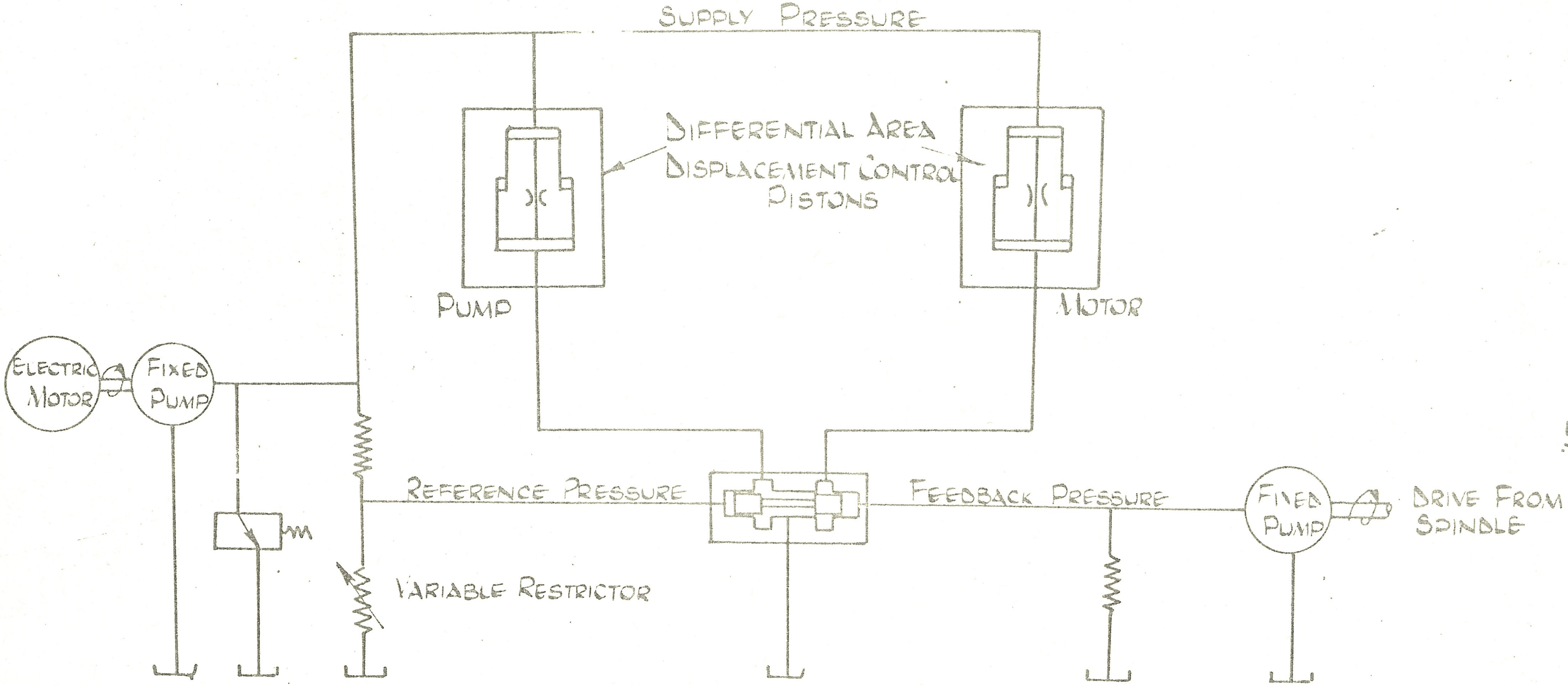
9. Fluid Power Control - Blackburn, Reethof & Shearer.

10. Design Principles of Metalcutting Machine Tools, F. Koenigsberger.



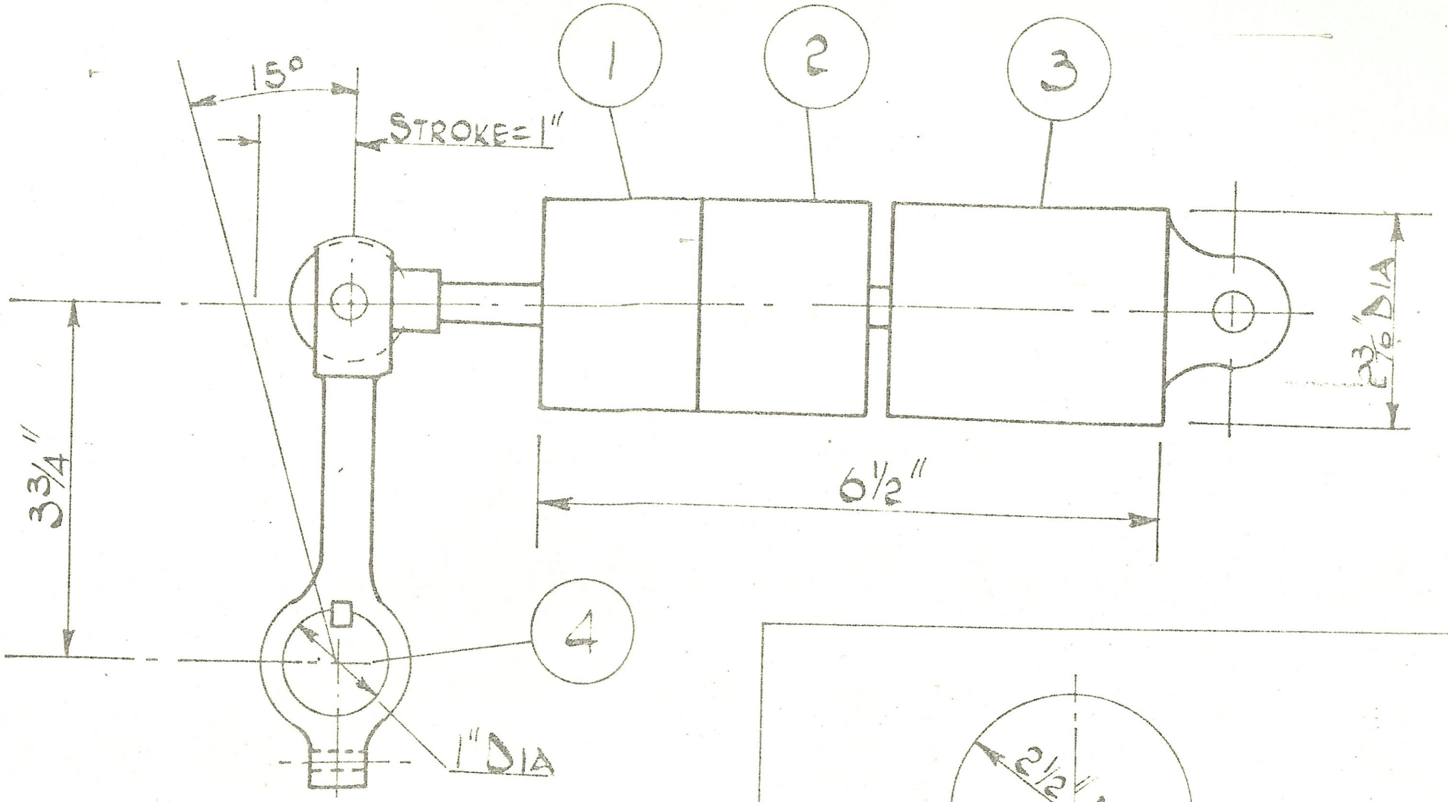
HYDRAULIC CIRCUIT FOR A VARIABLE PUMP/VARIABLE MOTOR SYSTEM

FIG CS/3/1/1

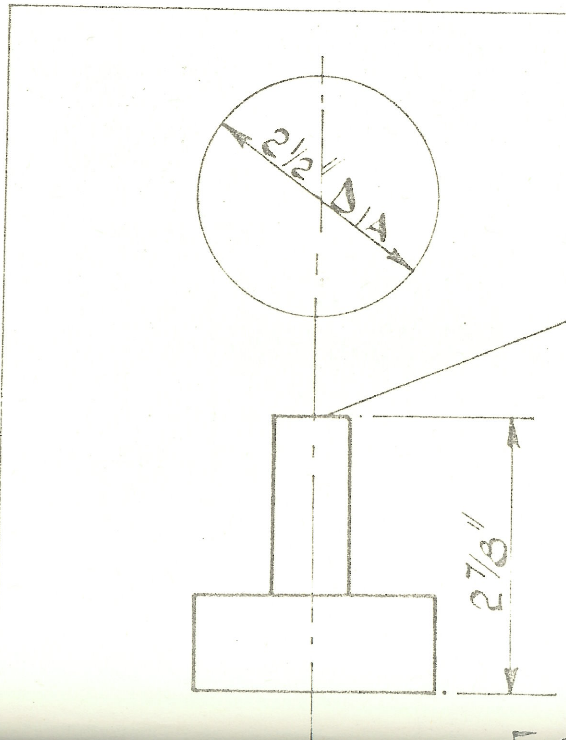


HYDRAULIC CLOSED LOOP SPEED CONTROL SYSTEM FOR VARIABLE PUMP/VARIABLE MOTOR HYDRAULIC DRIVE

FIG CS 3/1/2



PART No	NAME	COMMENTS	No-OFF
1	PNEUMATIC CYLINDER	SUPPLIERS:- KAY	1
2	"	PNEUMATICS	1
3	"	TYPE No:- KCN200	1
4	SWASHPLATE ACTURTION SPINDLE		1
5	FIVE PORT VALVE	SUPPLIERS:- KAY PNEUMATICS TYPE No:- KV28/D2S	3



DIMENSIONS OF SWASHPLATE ACTUATION SYSTEM FOR THE HYDRAULIC MOTOR

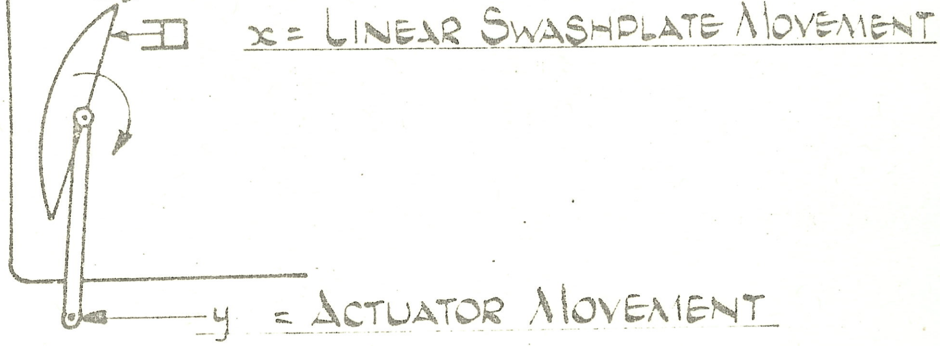
FIG CS/3/1/3

	Spindle Speeds R.P.M.										
	1	2	3	4	5	6	7	8	9	10	11
Pump Control	100	200	350	450							
Motor Control				450	800	1200	1550	1900	2250	2650	3000

(a)

Relative Lengths Of The Pneumatic Actuator Strokes

(b)



Speed $\propto$ Motor Stroke $\propto x \propto y$

(c)

Pneumatic Cylinder Strokes And Spindle Speed Range

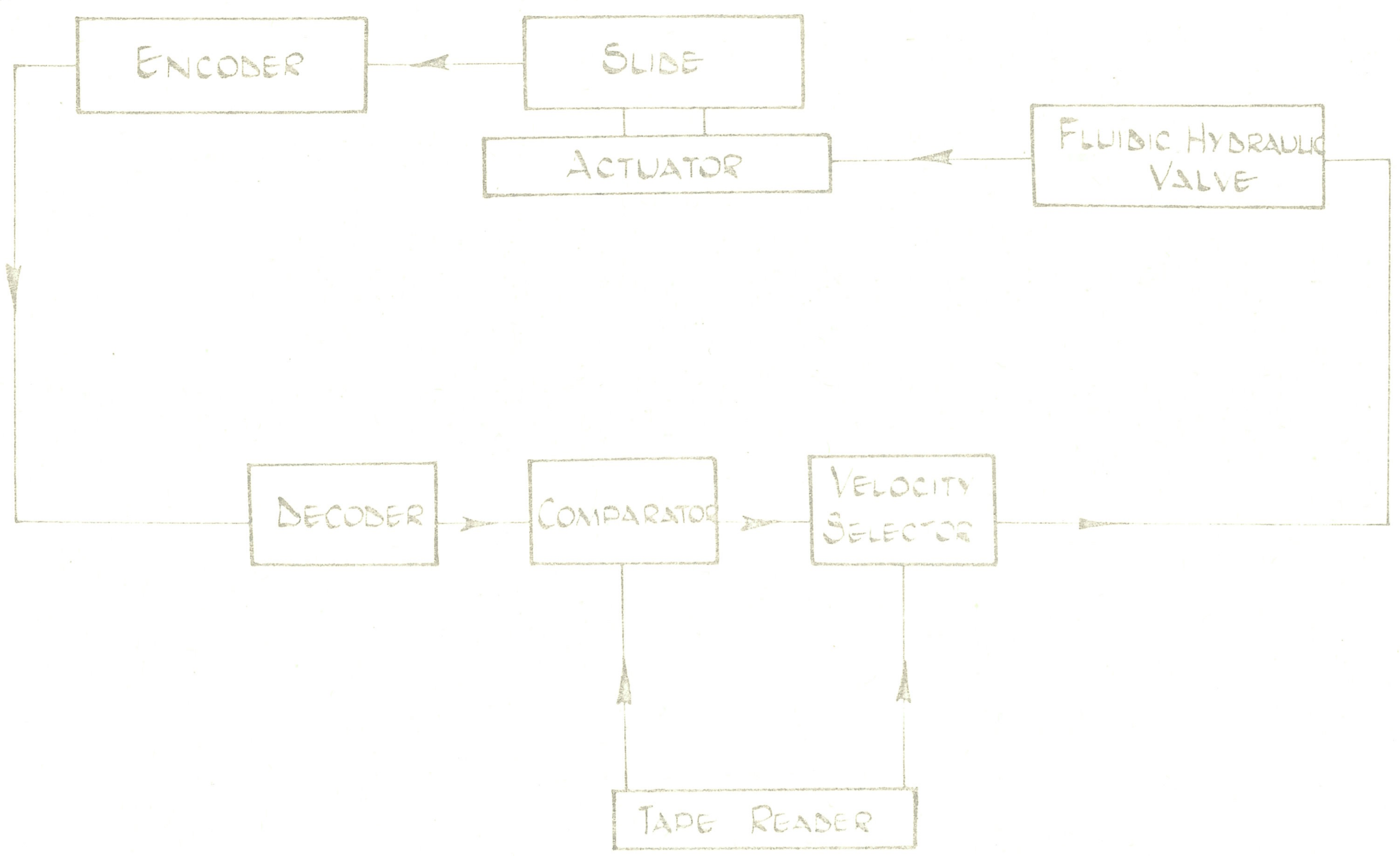
COMPLIANCE	2 μ'' / lb	10^{-4} mm/kg
LENGTH OF STROKE	10"	25 cm
PISTON DIAMETER	2.9"	7 cm
RECIPROCATING MASS	180 lb	80 kg
NATURAL FREQUENCY	175 cps	175 cps
MAXIMUM SPEED	100"/min	2.5 m/min
MAXIMUM OIL FLOW	10 cu in/sec	100 cc/sec
OIL BULK MODULUS	2×10^5 lb/in^2	1.4×10^4 kg/cm^2

CALCULATED SPECIFICATION OF LINEAR ACTUATORS

COMPLIANCE	7 μ''/lb	3.5×10^{-4} cm/kg
LENGTH OF STROKE	18"	4.5 cm
PISTON DIAMETER	2.1"	5 cm
MAXIMUM SPEED	100 in/min	2.5 m/sec
MAXIMUM OIL FLOW	5 cu in/sec	80 cc/sec

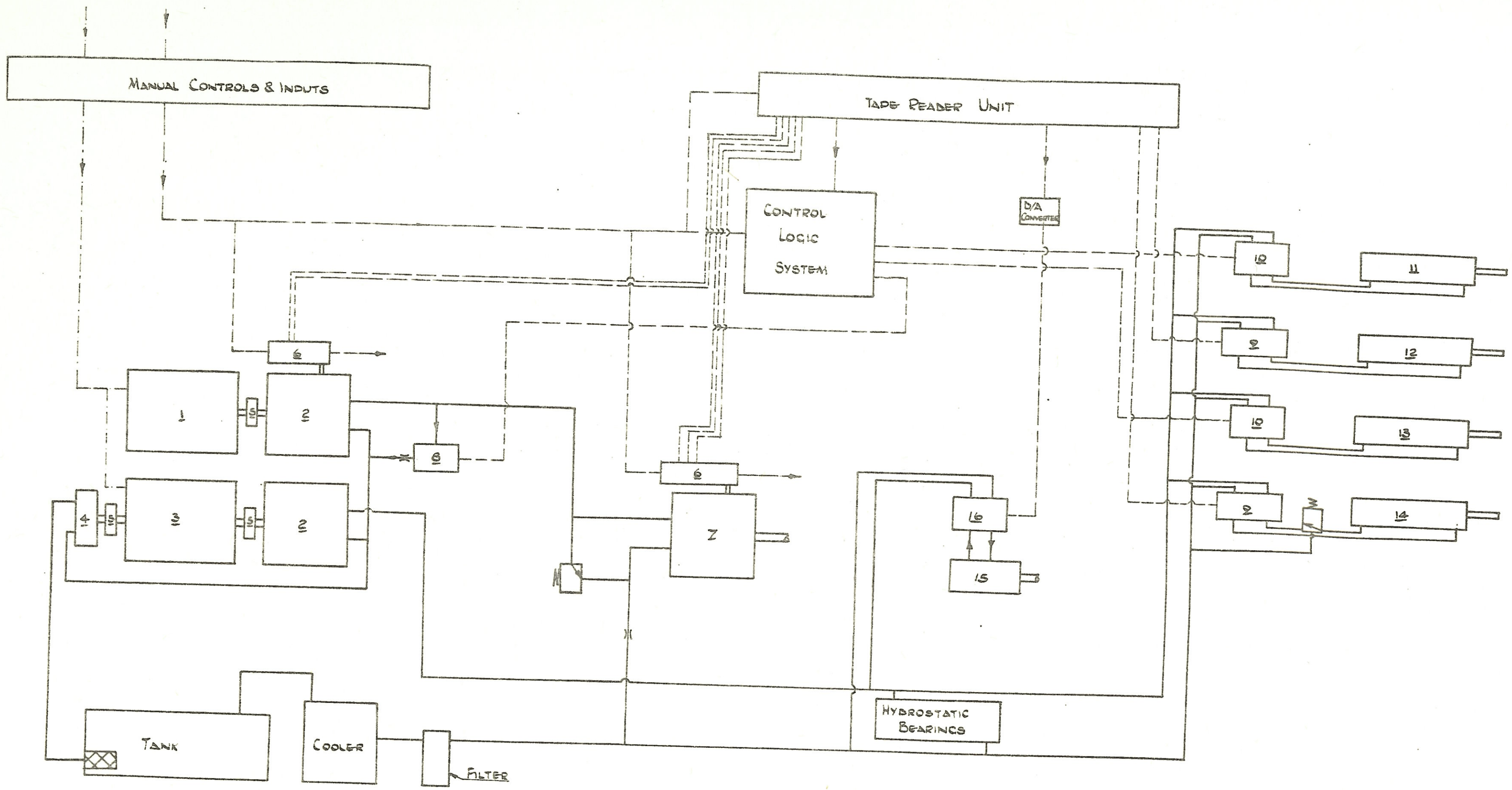
CALCULATED SPECIFICATION OF THE INDEXING ACTUATOR

FIG. CS/3/2/1.



BLOCK DIAGRAM OF THE LINEAR PATH CONTROL

FIG CS/3/2/2



THIRD ANGLE PROJECTION

TITLE - CONTROL ACTUATION CIRCUIT

THE COLLEGE OF AERONAUTICS

CS/3/0/1

ITEM No	DESCRIPTION	SUPPLIER	No OFF	COST (£)
1	ELECTRIC MOTOR		1	47
2	VARIABLE HYD PUMP	BOULTON PAUL	2	240
3	ELECTRIC MOTOR		1	30
4	BOOST PUMP	DOWTY HYDRAULICS	1	20
5	FLEXIBLE COUPLINGS	BOULTON PAUL	3	20
6	SWASH PLATE ACTUATORS	SEE FIG		55
7	HEADSTOCK MOTOR		1	200 (EST)
8	3 WAY VALVE	NORRIS BROTHERS	1	20
9	5-WAY VALVE	NORRIS BROTHERS	2	60
10	MOOG VALVES	DOWTY ROTOL	2	240
11	HEADSTOCK RAM	BALDWIN FLUID POWER	1	
12	HEADSTOCK INDEX RAM	" " "	1	100
13	TOOLPOST RAM	" " "	1	
14	TAILSTOCK RAM	" " "	1	
15	TOOLPOST MOTOR		1	40
16	PROPORTIONAL VALVE		1	80
	TANK			
	COOLER	BOULTON PAUL		60
	FILTERS			
	RELIEF VALVES			
	INTERCONNECTIONS		·	30

TOTAL £1300 (APPROX)

C.S.4 Control System Logic Sub-Committee Report

Chairman: V.A. White

H. Bera
M. Norton
M. Scarffe

SUMMARY

This report details the design of a fluidic control system giving rotational position control to 0.5° on chuck indexing, 0.001" positional control on linear motions and allows the selection of one of nine feed rates. The total cost of the control section is estimated at approximately £600.

CONTENTS

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CONTENTS

C.S.4.1 The Linear Encoder

General

An encoder is a device which presents information; position in this case, in a form which can be understood or utilized as an input to a control system. An encoder may give an analogue or digital output; the slide rule or micrometer are examples of analogue devices in that their reading is continuously variable and the final significance of the reading depends on the judgement of the operator, a desk calculator is a digital device because the information presented varies by discrete digits and not by proportions of digits.

As previously stated, a fluidic control system was chosen for this machine. Current fluidic analogue devices are not sufficiently accurate to enable an analogue encoder of the required accuracy to be designed with confidence. It was therefore decided to use a digital system.

The encoder could take one of two distinct forms; rotary or linear. Each has its advantages and disadvantages. Rotary encoders are mounted on the machine leadscrew or driven via a rack and pinion from the slide. In either case backlash in the mechanical mechanism can give encoding inaccuracies. However by the use of suitable gear ratios and encoder sizes finer discrete digits can be sensed. Linear encoders are incorporated in the slide to be controlled or are rigidly attached to it so that there are no backlash errors. However, the smallest digit used is limited by the sensitivity of the encoder sensing system. It must be noted that the smallest digit is not necessarily the smallest discrete position change that can be sensed. Ref. 1 gives details of a rotary encoder.

The committee considered that a linear encoder, in conjunction with ram actuation, was more suitable to this application. It was also decided that the encoder should take the form of a separate assembly which could be bolted to the slide because:-

1. A malfunctioning unit could be removed and a replacement fitted with the minimum of machine down time.

2. The same type of unit could be used on both the headstock and tool slides.

3. There would be a possibility of marketing the encoder and a readout for use in a similar manner to the Digiturn System.

4. The slide way design is not complicated by an integral encoder.

The encoder can be considered to comprise two parts; the coded strip and the sensing head. The sensing head translates the data contained on the coded strip into a form which can be used in the control logic circuitry. Either the strip or the sensor is fixed to the machine frame and the other to the slide. Since the sensor requires input and output connections it is more convenient to fix the sensor to the frame datum so that problems of paying out and taking up slack tubing are not encountered.

Choice of Code

There are many different codes which could be used to obtain the required information. A basic requirement of the code is that it should give an unambiguous signal at any position. In the binary code, more than one digit changes at a time, which can result in an ambiguous reading. This can be overcome by using a dual nozzle sensing system such as U-scan or V-scan, or by using a code in which only one digit changes at a time such as Grey code (a reflected binary code) or Petherick code.

The use of U-scan, V-scan on an anticipatory system increases the number of elements required in the sensor, hence increasing its bulk and power consumption. The code pattern is more complex and hence manufacture is likely to be more expensive than with the LIBAU and CRAIG code. This can be seen in Table 1, which shows the code pattern for "Binary", "Grey", "Petherick" and "Libau and Craig" codes; the latter can be more conveniently referred to as creeping code although it must be remembered that there are other creeping codes.

From Table 1 it can be seen that if each decimal change is 0.001" the smallest digit sizes required are; Binary 0.001", Grey 0.001" (dec. no. 9), Petherick 0.002" and creeping code 0.005" and that the creeping code can be represented by an inclined bar. The creeping code was therefore chosen because it is a code which should be cheap to manufacture and also because it gives the largest digit size for 0.001" increments.

The Sensing Head

For each decade the sensor contains five readout units. The signal coming from the sensing jet is a very low pressure one requiring shaping and amplification. This can be done by a mono-stable or bi-stable element. Generally mono-stable elements have a lower maximum operating frequency than bi-stable elements. Also the use of two control jets 180° out of phase (one jet coming on as the other goes off) can with correctly designed jets give a "push-pull" control signal to the element which enables a digit on the code of smaller physical size to be sensed.

The correct phasing of the control jets can be obtained by spacing the jet centres at an odd multiple of the encoder-track digit size or by using a track for each jet, the two tracks being 180° out of phase. In the latter, the jet which runs on the "true" track can be considered to be the set jet and the other the reset jet. See Figs. C.S.4.1a and 1b.

Dec. No.	BINARY				GRAY			
0	0	0	0	0	0	0	0	0
1	0	0	0	1	0	0	0	1
2	0	0	1	0	0	0	1	1
3	0	0	1	1	0	0	1	0
4	0	1	0	0	0	1	1	0
5	0	1	0	1	0	1	1	1
6	0	1	1	0	0	1	0	1
7	0	1	1	1	0	1	0	0
8	1	0	0	0	1	1	0	0
9	1	0	0	1	1	1	0	1

Dec. No.	PETHERICK				LIBAU and CRAIG (Creeping Code)				
0	0	1	0	1	0	0	0	0	0
1	0	0	0	1	0	0	0	0	1
2	0	0	1	1	0	0	0	1	1
3	0	0	1	0	0	0	1	1	1
4	0	1	1	0	0	1	1	1	1
5	1	1	1	0	1	1	1	1	1
6	1	0	1	0	1	1	1	1	0
7	1	0	1	1	1	1	1	0	0
8	1	0	0	1	1	1	0	0	0
9	1	1	0	1	1	0	0	0	0

TABLE 1

The Coded Strip

An encoded movement of 8" is required in 0.001" divisions, this requires four decades:-

Decade	Measuring	Digit Size
1	0-0.009"	0.005"
2	0.01-0.10"	0.050"
3	0.100-1.0"	0.500"
4	1.00-10.0"	5.000"

Five sets of nozzles are required per decade.

The cheapest satisfactory method of producing the strip is etching in shim from a master produced on a N.C. photographic machine.

The code pattern must be arranged so that exhausted air from a track does not affect the adjacent tracks. At the same time the shim must be adequately supported. Both objects are achieved; at the expense of increased width, by mounting the strip on a base plate which has ridges under the junction of the tracks.

Control Jets

Ref. 2 gives some information on the performance of different size control jets. The control jets used in the design here are of the back pressure type, integral with an amplifying element (see PT/CS/C/114). Very little detailed work on the use of such jets has been published but from the previous experience of the committee it was concluded that for satisfactory jet pressure rise time and definition the jet should be rectangular in shape, at least 10 - 20% narrower than the digit to be sensed, and have an aspect ratio of 3 to 6.

The sensing gap (distance from the coded strip to the jet orifice) is very critical particularly as the jet size decreases. Consequently the encoder must be designed to maintain this gap at 0.0005" to 0.001". Shielded or reflected jets could be used at wider gaps but would increase the overall size of the assembly.

Since the smallest digit is 0.005" a jet of 0.004" wide must be used.

Encoder Scheme 1

The creeping code pattern can be arrived at by using a horizontal code pattern and displacing the reading heads relative to the code or by inclining the code and having the readout head-nozzle datum in a straight line normal to the axis of the encoder. The first possibility is considered here. It is assumed that the required elements are 0.1" thick and made of etched shims or injection moulded. Dycril elements could not be used because of the variation in section depth of the element i.e. 0.004" nozzle and, say, 0.020" - 0.040" supply jet.

This system was subsequently rejected because of the need to maintain accurately the total element thickness and jet positions and the effects of cumulative tolerances on etched shim elements. Table 2 gives the required relative jet and code positions.

With regard to Table 2, L_n is the distance of the control jet centre line denoted by the suffix from the centre line of control jets for element 1 : 1. This dimension is made up by adding suitable spacing shims between element units.

Decade	Sensor Element	Effective Reading Position (in)	"True" Track start	L_n
1	: 1	0.001	0.000	0.000
	: 2	0.002		0.101
	: 3	0.003		0.202
	: 4	0.004		0.303
	: 5	0.005		0.404
2	: 1	0.010	0.010	0.510
	: 2	0.020		0.620
	: 3	0.030		0.730
	: 4	0.040		0.840
	: 5	0.050		0.950
3	: 1	0.100	0.150	1.050
	: 2	0.200		1.150
	: 3	0.300		1.250
	: 4	0.400		1.350
	: 5	0.500		1.450
4	: 1	1.000	2.550	1.550
	: 2	2.000	3.650	1.650
	: 3	3.000	4.750	1.750
	: 4	4.000	5.850	1.850
	: 5	5.000	6.950	1.950

TABLE 2 Relative Arrangement of Codes and Sensors

Encoder Scheme 2

If the sensing heads are placed parallel to the line of motion (axis of encoder) (Fig. C.S.2.1.1A) a system which is easier to understand is obtained. However, the design of elements for fabrication from etched shim is more complicated than for Scheme 1. The elements could still be injection moulded but as the maximum practical aspect ratio for this type of manufacture is approximately three, and since the width of the control jet is 0.004", the depth would be 0.012". Elements of this size are not at the moment in general use but are perfectly feasible. Alternatively one of the low aspect ratio elements currently being developed might be suitable.

Elements of a standard type could be used for decades 1, 2,3, that is, fifteen bistables, although the distance apart of the control jets would vary. For decade 4, five monostable units are used. The operating frequency of decade 4 is considerably less than that of the remainder and consequently the use of a monostable is no disadvantage.

One set of the bistable elements' control jets and the control jets of the monostables are set on the effective encoder zero datum line. The reset jets of the bistable elements are positioned at odd multiples of the digit size i.e. so that one jet goes off as the other comes on. It is desirable that the jets are as close together as possible so that they can all be supplied from the same manifold. The control jet centre line distances then become, decade 1 0.475", decade 2 0.450", decade 3 0.5".

No displacement of the codes is required as the set reading heads are all positioned on the encoder zero datum, but an inclined code is therefore used.

Either scheme could be used but scheme 2 offers a more suitable design for laboratory study and is therefore shown in the accompanying drawings. Scheme 1 would require a different trolley assembly and a different coded strip.

The coded strip is etched from both sides in 0.005" thick INVAR. This material has been selected because of the possible thermal expansion effects, for example if the entire encoder were made of mild steel an error of approximately 0.003" could occur due to thermal expansion. This is at maximum extension and assuming a 10° C temperature rise, however the system will be thermally shielded and temperature rise is not expected to exceed 3° C. The strip is mounted on a mild steel base plate which contains exhaust passages. Longitudinal location is provided by one of three pairs of dowels positioned half-way along the length of the strip. The strip is held on to the base plate by the runners. These runners do not clamp the strip. The surfaces of the runners and base plate which contact the strip are sprayed with P.T.F.E. to give a solid lubricant film so that their expansion does not affect the strip.

The sensors are mounted on an input manifold block, the sensing gap is controlled by four roller bearings so that the manifold sensor assembly forms a trolley. The trolley is spring loaded against the strip and connected to the machine slide by an INVAR rod.

An encoded length of 8" is required but because the sensing elements are parallel to the direction of motion and the effective trolley length is 0.5", that is the control jet centre line distance for decade three, an actual encoded strip length of 10" is used so that the total encoded distance is 9.5".

The central manifold is 0.5" thick, the elements are 0.040" thick, mounted on either side of the manifold and clamped by two end plates. Thin Dycril, Type 30, can be used for evaluation purposes giving an element depth of 0.016". Each track is 0.025" and the tracks are spaced at 0.020" intervals so that 0.005" shims are placed between the elements to give the correct spacing. Shim thickness is varied to correct for element thickness variations. As the tracks are 0.025" wide and the sensing jets 0.016" x 0.004" a cumulative tolerance of 0.009" is permissible on the element spacing. The jet spacing and output positions (PT/CS/C/114) are given in Table 3 and the drawing number and part list in Table 4.

Supply input is via hole D(PT/CS/C/101) one end of which is blocked. The outputs from the sensing elements can be taken direct from the elements after drilling and fixing of suitable terminal tubes or via a manifold screwed on to the ends of the central manifold.

Supply, and output tubes are bound to the encoder connecting rod. A thin rubber sheet is fixed over the open end of the encoder to prevent ingress of contaminants.

It may be desirable to have the code zero datum at either end of the encoder travel. To allow for this situation the sensor trolley is provided with two attachment points so that the trolley and strip can be turned through 180°.

Three pairs of dowel mount holes for the strip are provided. If the encoder is mounted by its ends, then one of the outer pair is used; if the encoder is mounted by the base then the central pair is used.

Zero position adjustment is obtained by varying the effective length of the eyelet bearings at either end of the connecting rod. These bearings are precision aircraft servo linkage bearings and should be sufficiently accurate for our needs. If it proves that they are not sufficiently accurate, spring loaded spherical bearings can be used.

If it is required to decrease the digit size to 0.0001", this can be achieved by the addition of a coded drum to the trolley, driven via a rack and pinion from the strip mounting.

The estimated cost is £80 - £140 per encoder.

Element	Type	Z inches	Output	Decade	Decoder Input
1	Bistable	0.4750	1	1	E
2	"	"	2	0.001"	D
3	"	"	1		C
4	"	"	2		B
5	"	"	1		A
6	"	0.4500	2	2	E
7	"	"	1	0.010"	D
8	"	"	2		C
9	"	"	1		B
10	"	"	2		A
11	"	0.5000	1	3	E
12	"	"	2	0.100"	D
13	"	"	1		C
14	"	"	2		B
15	"	"	1		A
16	Mono-stable	-	2	4	E
17	"	-	1	1.0 "	D
18	"	-	2		C
19	"	-	1		B
20	"	-	2		A

TABLE 3 Sensing jets centre line spacing "Z" and output position 1 or 2 see PT/CS/C/114

Element numbers refer to relevant track numbers, see PT/CS/C/100. The decoder inputs given are those at the rear of the block i.e. closest to code datum. The other outputs give the negated signal.

C.S.4.2 Code Converter for Linear Encoders

The code used was invented by Libau and Craig. The code has a decimal, creeping form, i.e. the basic unit is subdivided into ten sections and starting from the least significant digit the code advances progressively and uniformly with increase in significance, until digit 5 and thereafter decreases progressively. This is clearly seen in the table below.

Decimal No.	Libau and Craig (Creeping code)					Binary code			
	A	B	C	D	E	W	X	Y	Z
0	0	0	0	0	0	0	0	0	0
1	0	0	0	0	1	0	0	0	1
2	0	0	0	1	1	0	0	1	0
3	0	0	1	1	1	0	0	1	1
4	0	1	1	1	1	0	1	0	0
5	1	1	1	1	1	0	1	0	1
6	1	1	1	1	0	0	1	1	0
7	1	1	1	0	0	0	1	1	1
8	1	1	0	0	0	1	0	0	0
9	1	0	0	0	0	1	0	0	1

The binary code is normally used for the logical representation and processing of data. However, direct encoding in binary presents serious problems. This means that either the data from the encoder must be processed in creeping code which means converting the static binary (ISO standard) signal from the reader into creeping code or converting the encoder signal into binary and processing in binary.

Libau and Craig to Binary converter

By inspection of the table above it can be seen that:-

$$W = A.\bar{C}$$

that is, the only time 1 appears in the W column of the binary code is when 1 appears in the A column and 0 appears in the C column of the creeping code. Data relating to B.D.E. is not required to define W,

$$\therefore W = A.\bar{C}$$

similarly $X = B.C$

$$Y = \bar{B}.D + C.\bar{E}$$

$$Z = \bar{D}.E + \bar{B}.C + A.E + C.\bar{D} + A.\bar{B}$$

These relationships can also be arrived at by Vietch mapping techniques.

The creeping code is represented on a five variable map Fig. C.S.4/2/1. Of the thirty-two possible combinations only ten are required, the remainder are termed "Don't care functions" and can be combined with the actual combination required to minimise the relationship.

Consider binary W which exists for decimal 8 and 9 only. This is shown in Fig. C.S.4/2/2 and 3. Both figures represent the same conditions. Fig. C.S.4/2/3 is included to show that the combinations of 8 and 9 with "don't care functions" is valid.

Combination gives:

$$W = A(B.\bar{B}).\bar{C}.(D.\bar{D}).(E.\bar{E})$$

Now $B.\bar{B} = D.\bar{D}$

$$= E.\bar{E} = 1$$

$$\therefore \underline{\underline{W = A.\bar{C}}}$$

which was previously shown by inspection. This process is repeated for X, Y, Z and verifies the previous conclusions.

Logic circuits are then designed to satisfy these equations. The circuit may be made up of active only or active and passive elements. In this case active elements only were used.

It should be remembered that the encoder includes amplifying elements so that for each digit a true and negated signal (i.e. A and $\bar{A}$) can both be used as inputs to the decoding circuit simultaneously. This simplifies the logic and reduces the fan-out required. Fig. C.S.4/2/4 shows the creep to binary converter. Twelve two input, and one three input, OR/NOR-s are used. The longest circuit path is three elements. Four decoders are required per axis, estimated cost, £56 for both axes.

Binary to creep converter

By the same procedure the circuit shown in Fig. C.S.4/2/5 is derived. This could be used after the tape reader to convert the static signal into creep which could then be compared in this form. This is preferable to converting the dynamic signal. However, it was found that the comparator design was then unsatisfactory inthat the direction sense was more difficult to obtain.

Here the relationships are:-

$$A = W + XZ + XY$$

$$B = X + W\bar{Z}$$

$$C = X + YZ$$

$$D = X\bar{Y} + \bar{X}Y + XZ$$

$$E = X\bar{Y} + \bar{X}Y + \bar{W}\bar{Y}Z.$$

47

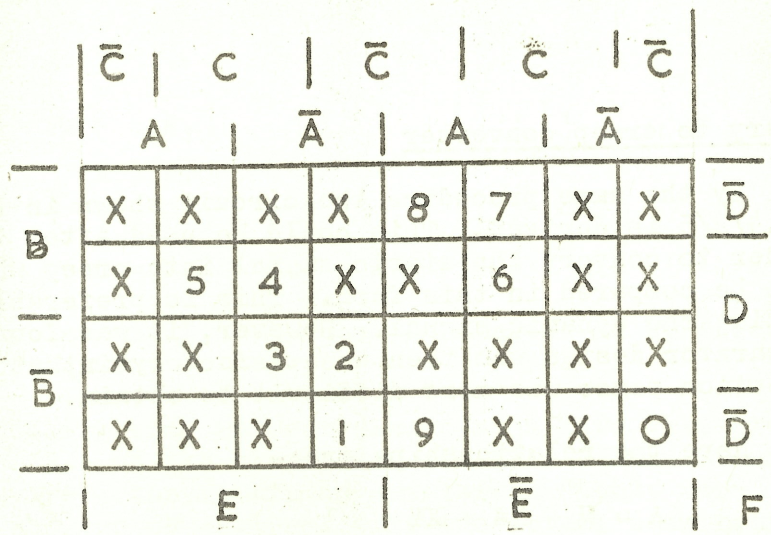


FIG. CS4:2:1

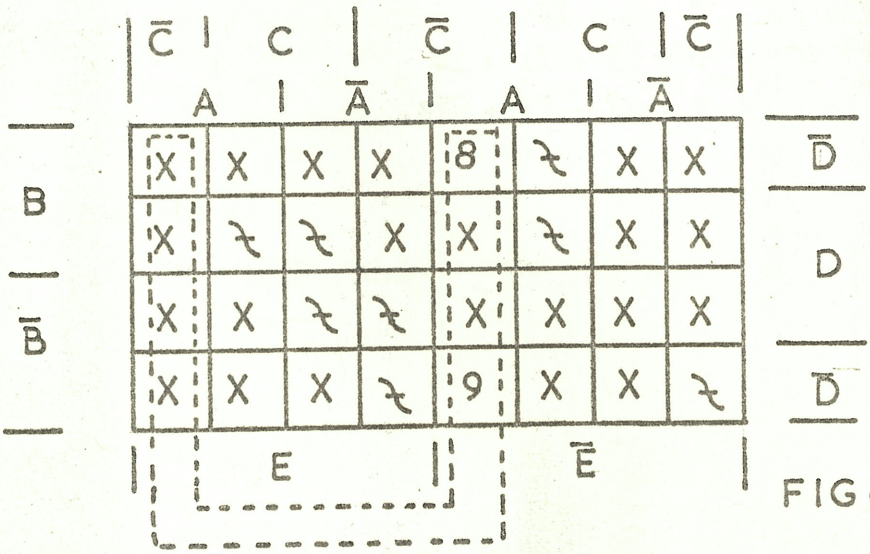
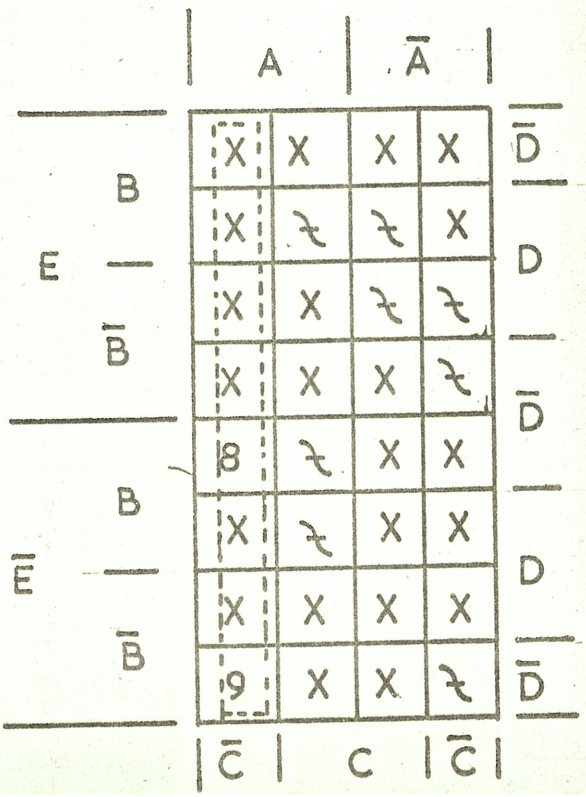


FIG CS4:2:2

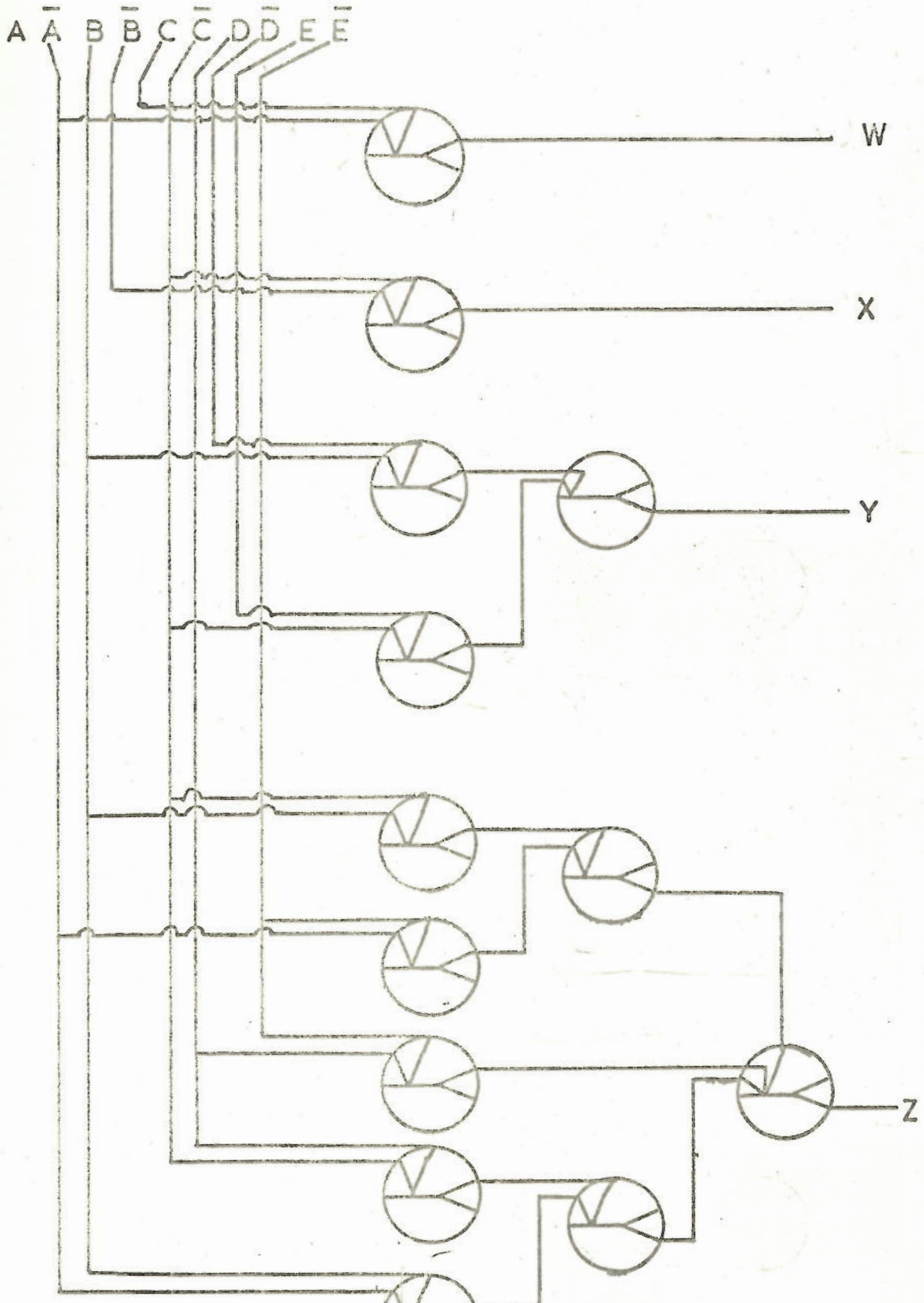


X "DON'T CARE FUNCTIONS"

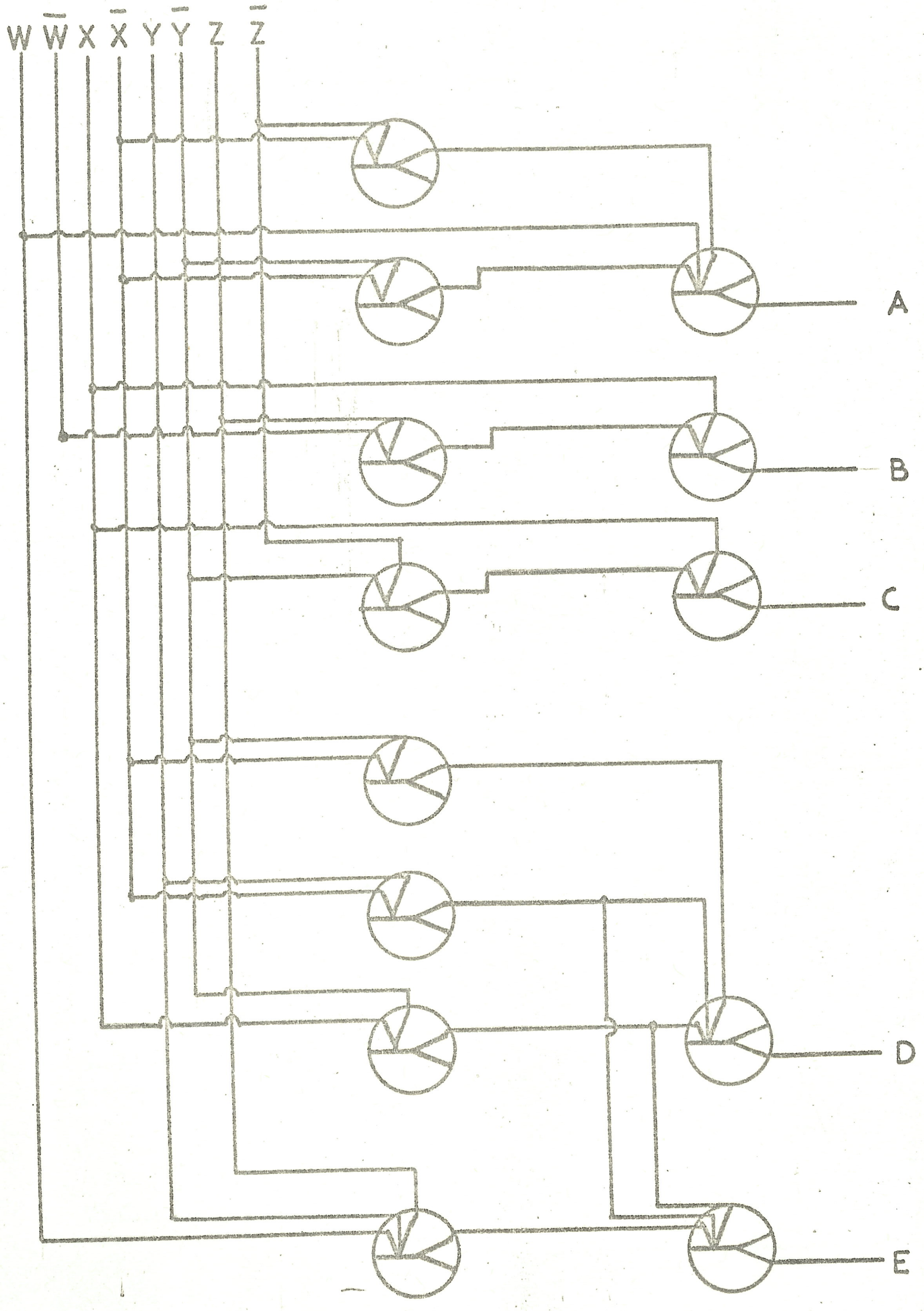
OTHER DECIMAL NUMBERS

PATH OF COMBINATION

FIG. CS4:2:3



LIBAU & CRAIG TO BINARY CONVERTER
FOR ONE DECADE



BINARY TO LIBAU & CRAIG CONVERTER FOR ONE DECADE

FIG CS4:2:5

C.S.4.3 Comparator

Introduction

Many circuits have been developed for processing command signals and actual position signals so as to feed intelligible information into a control system. The complexity of such circuits is in general related to the requirements in achieving the final position. This statement can be explained by briefly describing some existing linear positioning systems.

If the requirement was that the system must move a mass from point A to point B, and overshoot can be tolerated, then the "comparator" need be nothing more than a sense-of-error device coupled with an equivalence signal generator. This type of system has been built (Ref. 1) and has met the design requirements.

If the requirement was, that the movement above must not overshoot, then the design can take one of several forms. One is to build into the first system a method of anticipating the final position, so that when equivalence occurs the mass is moving so slowly that it stops on this position. This is done by the table sending to the "comparator" a signal that is physically leading the true position signal. When this is equivalent to the command signal the machine switches to a creep rate. The "comparator" now compares the true position with the command signal, and, when they are equivalent stops the movement, thus overshoot is reduced to an acceptable value. This method has the disadvantage that only the highest approach speed gives the fastest possible response, since the reading head is positioned at a distance equal to the stopping distance at the highest speed. Also, there is the added complication of building two additional sets of reading heads; one for forward travel "lead", and one for reverse travel "lag".

Another form is a subtractor circuit which continuously gives the difference between command and actual position signals. The disadvantage of this type of "comparator", apart from the large number of elements it requires, is that additional logic has to be added to prevent overshoot (Ref. 2)

Project Requirement

The requirement of the project was to move a mass from point A to point B without overshoot, and at a controlled feedrate. The system had to accept binary tape inputs in four decades (B.C.D.) and also position signals in the same form. The signals required by the velocity control circuit from the comparator are outlined in the following truth table.

Forward	Reverse	Creep	Meaning
1	0	0	The difference between tape and machine is greater than creep initiation distance so read velocity and go forward.
0	1	0	The difference between tape and machine is greater than creep initiation distance so read velocity and go reverse.
1	0	1	Initiate creep speed and go forward.
0	1	1	Initiate creep speed and go reverse.
0	0	0	Stop all motion.
1	1	1)	Any of these combinations indicate a malfunction of the comparator and should be used to switch the machine control off.
1	1	0)	
0	0	1)	

Space limitations in the linear encoders made the fitting of lead and lag nozzles difficult and it was therefore decided that the comparator should detect the point at which creep rate is started. To prevent overshoot under certain conditions such as approaching exact numbers, i.e. when all decades change at the final position, it was decided that the comparator should anticipate these. The method chosen is to take the absolute difference in a given decade. When this is reduced to unity, a carry 'one' signal is generated, and sent into the next decade down. This causes the decade to give a large error which again when reduced to unity sends a 'carry one' signal, thus it is impossible for all decades to change simultaneously.

The method used to detect the creep rate change point, was to look at the 1in. decade together with the 0.1in. decade, when the forward or reverse signals from these became zero it meant that 0.1in. had been carried forward. In machine distances this means that dependent on the remaining 0.01in. and 0.001in. decades the mass could be 0.1in. from final position or at worst 0.199in. Because of the simplicity of the circuit it was decided that this variable creep selection point was acceptable. Also, overshoot of the change point is bound to occur because of the time delays in the fluid and hydraulic circuits.

Circuit Operation

With reference to Fig. C.S.4/3/1, the function of the D/A converters is to produce a pressure or flow output that is an analogue of the binary input, Fig. C.S.4/3/2A shows an idealized output from such a device. If two elements are connected as shown in Fig. C.S.4/3/1, then the output of amplifier PA_1 will be proportional to the difference between the binary input signals.

If PA_1 is centre dump, then the output from a side leg will be normally zero, and increase for increasing differentials. Figs. C.S.4/3/2B, C, D show idealized output pressures from such an amplifier. If the dump output of PA_1 is fed to the input of PA_2, and this input is balanced by a bias pressure B_3, then the output, O_3 will be as Fig. C.S.4/3/2E for various input errors. The output pressures shown on graphs C.S.4/3/2B, D and E can now be utilised to trigger low hysteresis OR-NOR elements at the required points.

Inspection of the overall circuit reveals a symmetry in each outer circuit, and therefore for clarity only the forward loop is described. B_1 biases ON_1 so that for tape > machine by 2in. flow is out of the OR leg, at the transition between 2in. and 1in. flow switches to the NOR leg. B_2 biases ON_2 such that for tape > machine by 1in. the flow is switched to the OR leg, at the transition between one and zero, flow switches to the NOR leg. Thus for tape machine by 2in. ON_1 holds ON_4 in the OR output. ON_2 is on the OR output so therefore ON_6 switches to the NOR leg (forward). At the transition from two to one ON_1 switches to the NOR leg, this causing ON_6 to switch to the OR leg. ON_4 now switches to the NOR leg (carry 10 forward). At the transition from one to zero, ON_2 switches to the NOR output which switches ON_4 to the OR output. C_R and C_F are bias inputs applied to the D/A converters which have a value of 10 units. This input is not used on the 1in. decade but is used on all finer ones. The total output graph of the D/A converter is shown in Fig. C.S.4/3/2F. It should be realised that the proportional amplifiers should be designed so as not to saturate when an input analogue pressure of nineteen is present.

The circuit shown in Fig. C.S.4/3/1 can be represented by a function block, these blocks can be coupled together as shown in Fig. C.S.4/3/3 so as to produce the required signals for the velocity control circuit.

Number of Elements

The number of elements required per axis for the comparator are as follows:

Active	Passive	Total	Cost
41	8	49	£30.10.0.

Total Time Delay

The time delay for the circuit has been estimated to be 2.8mS.

Circuit Feasibility

As yet the type of D/A converters specified have not been developed, a crude element whose output pressure is as graph F, Fig. C.S.4/3/2, has been developed by introducing a bias port into a D/A converter.

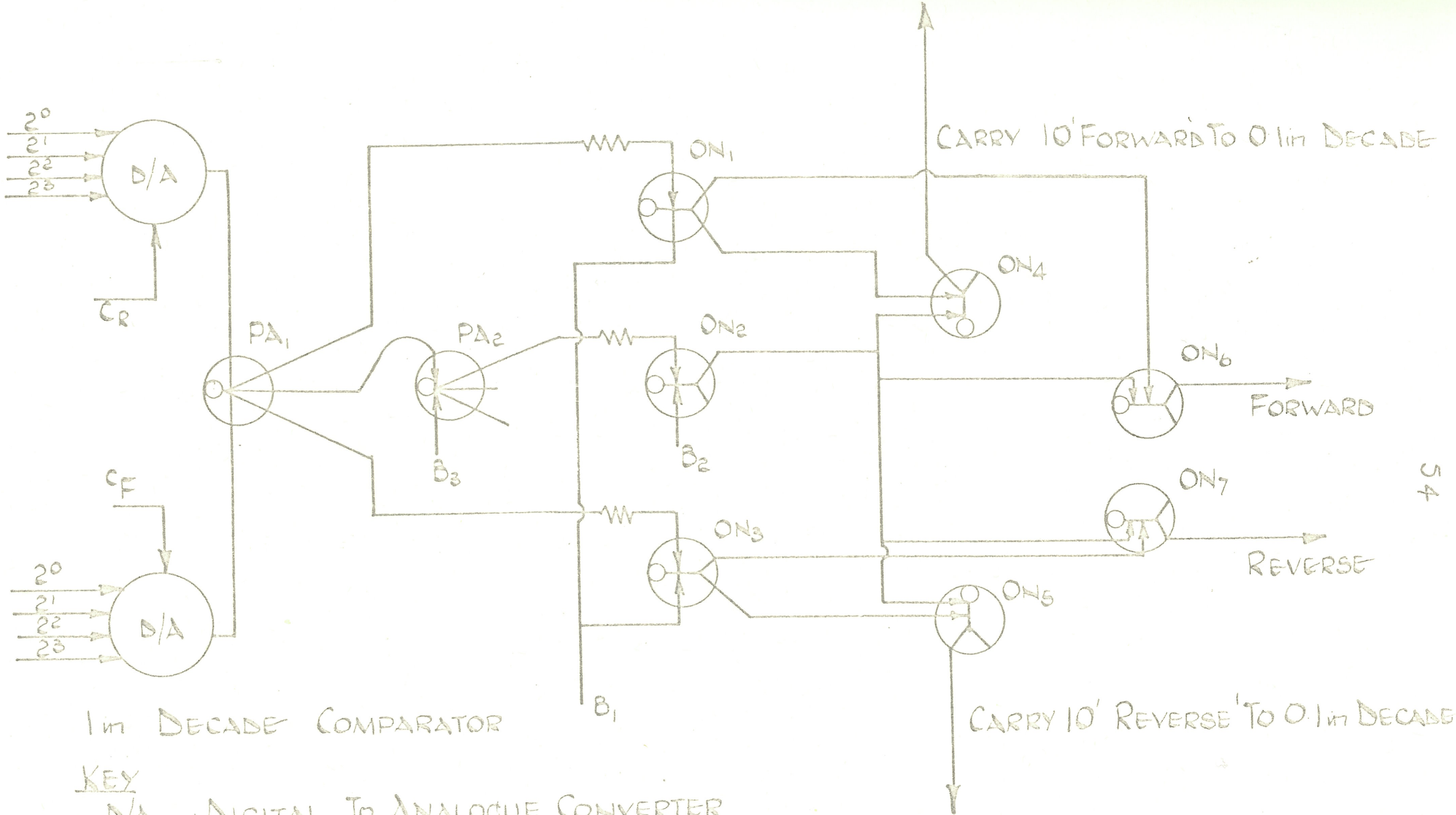
As ON_{1-3} only in the worst case have to detect a step of one part in nineteen this should not be too difficult to achieve.

Only one element ON_2 is required to have a fan-out of 4.

REFERENCES

1. AVIV, I., BIDGOOD, R.E. and RAMANATHAN, S.R. — The design and construction of a pure fluid four-bit binary comparator. 2nd Cranfield Fluidics Conference, 1967, paper H6.

2. VEASEY, R. and POWELL, E.A. — Drilling system design project. Final Report of Control Systems Committee, 1967.



1 in DECADE COMPARATOR

KEY
D/A DIGITAL TO ANALOGUE CONVERTER
PA_{1-2} CENTRE DUMP PROPORTIONAL AMPLIFIERS
ON_{1-3} LOW HYSTERESIS OR - NOR ELEMENT
ON_{4-7} OR - NOR ELEMENTS

FIG CS4/3/1

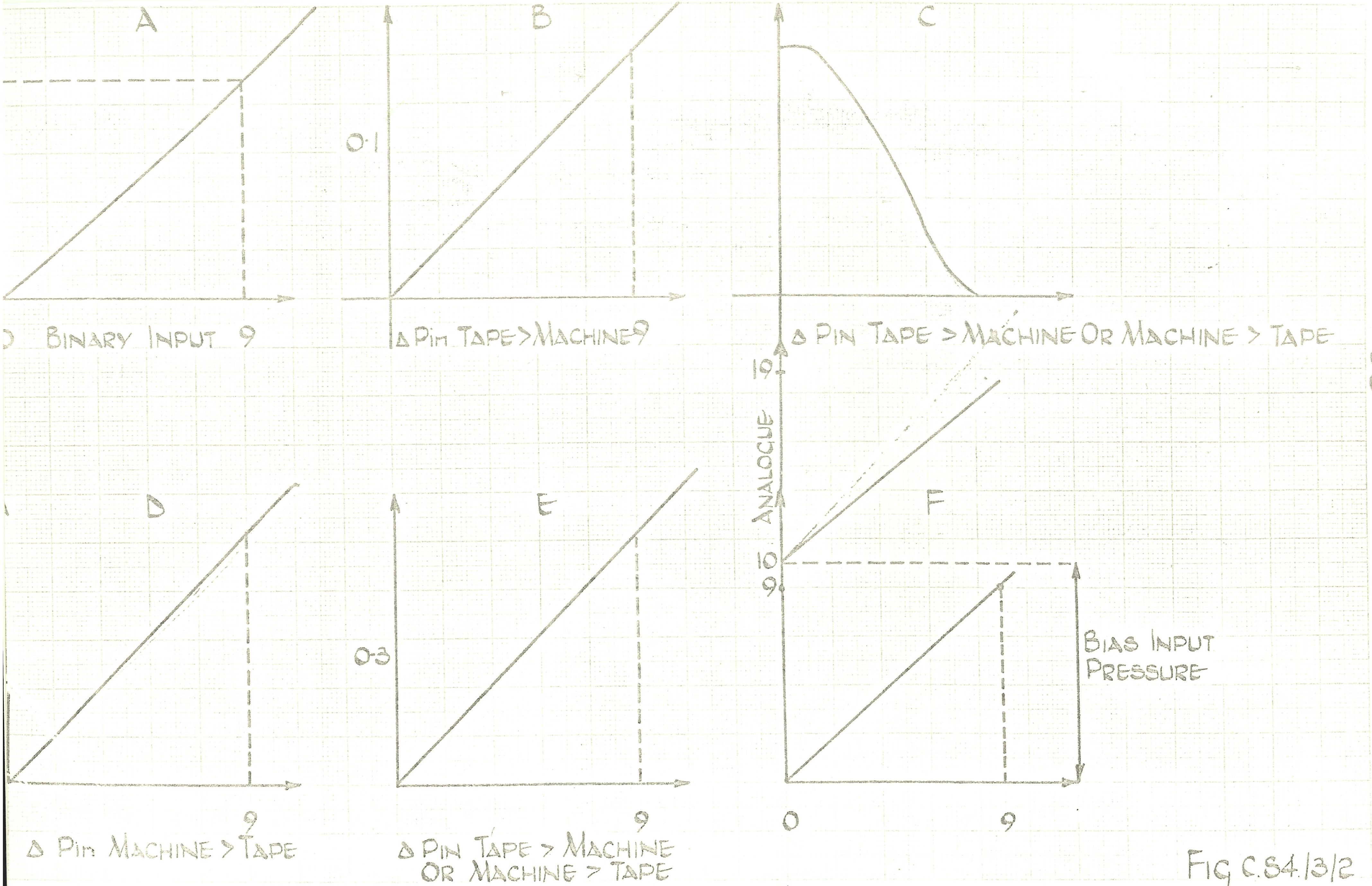

A
B
C
0·1
0
Binary Input 9
Δ Pin Tape > Machine 9
Δ Pin Tape > Machine Or Machine > Tape
19
Analogue
10
9
F
D
E
0·3
Bias Input
Pressure
9
Δ Pin Machine > Tape
9
Δ Pin Tape > Machine
Or Machine > Tape
0
9

Fig C.S4/3/2

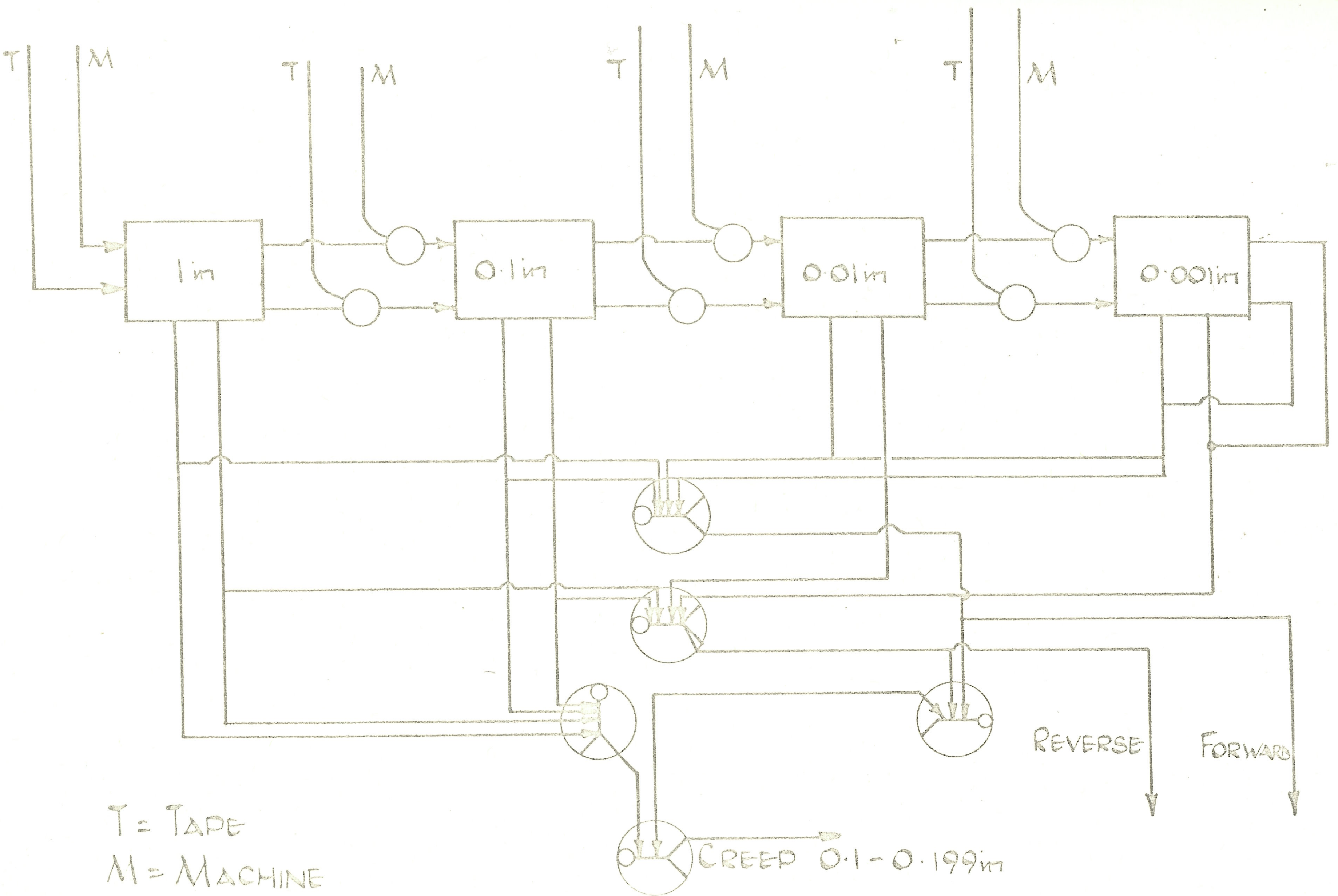

T
M
1 in
0·1 in
0·01 in
0·001 in
REVERSE
FORWARD
CREEP 0·1 - 0·199 in
T = TAPE
M = MACHINE

FIG CS4/3/3

C.S.4.4 Creep Rate Selection and Time Delay Effects

In this application positional overshoot must be limited. This means that the creep speed, circuit delay and machine deceleration rate must be considered in conjunction with the desired positional tolerance.

Component	Estimated Delay m.sec.	Effective Element Path Length
Encoder	0.8	1 + 2 jets
Decoder	1.05	3
Comparator	2.8	8
Velocity control	1.05	3
Interconnections	5.00	
Moog valve	5.00	
Sub total	15.7	
say,	16	
Actuation cylinder	14	
Total	30	

The time of 14 m.sec. is greater than that which is likely in practice (as is the figure for interconnections). The actual figure is dependent on the machine parts sliding inertia and the actuator's hydraulic stiffness.

With reference to Fig. C.S.4/4/1 the distance travelled d must not exceed 0.0003" in the worst condition.

$$\therefore \quad 0.0003 = \frac{u}{60} \left\{ \frac{16}{1000} + \frac{14}{2 \times 1000} \right\} \quad \text{where } u = \text{"/min.}$$

$$\therefore \quad u = 0.0003 \times 1000 \times \frac{60}{23} \doteq 0.9 \text{ "/min.}$$

$$\therefore \quad \text{deceleration} = \frac{0.9}{60} \times \frac{1000}{14} \times \frac{1}{12} = 0.086 \text{ ft/sec.}$$

$$= \frac{0.086}{32.2} \doteq \underline{\underline{0.003}}\ g$$

A deceleration rate of 0.003g is extremely low and the deceleration would be much higher than this, therefore a creep speed of 1"/min. can be safely used.

The highest feed rate is 30"/min. Considering the change from 30"/min to creep feed we have the situation in Fig. C.S.4/4/2. The rate of deceleration required to change from 30"/min. to 1"/min. in 14 m.secs. is approximately 0.1g. During the deceleration period the tool is cutting but it is unlikely that surface finish would suffer especially as the deceleration is not instantaneous.

The feed change takes place at 0.1" to 0.199" prior to the desired position.

From Fig. C.S.4/4/2

$$d = \frac{30}{60} \times \frac{16}{1000} + \frac{29}{60} \times \frac{7}{1000} + \frac{1}{60}\left(\frac{14 + 16 + 7}{1000}\right) + \frac{1}{60} \times \frac{x}{1000}$$

$$= \frac{8 + 3.4 + 0.62}{1000} + \frac{x}{60{,}000} \quad ; (x = \text{m.sec.})$$

$$= .012 + \frac{x}{6} \times 10^{-4}$$

taking $d = 0.199"$

then $x \triangleq 11$ sec. (1)

taking $d = 0.100"$

then $x \triangleq 5.4$ sec.

The remaining feed change situation which requires consideration is the change from fast traverse to feed rate prior to the tool meeting the work. Here the worst combinations are anticipatory distance 0.1", and a desired feed of 1"/min. i.e. creep feed.

This is represented by Fig. C.S.4/4/3

$$\text{distance travelled } d = \frac{100}{60} \times \frac{16}{1000} + \frac{1}{60} \times \frac{14}{1000} + \frac{99 \times 7}{60 \times 1000}$$

$$\triangleq 0.040"$$

which is well within the permissible 0.1".

Hence the deceleration would be 0.3g.

From the above it is seen that a fast traverse speed of 100"/min., fast feed rate of 30"/min., creep feed rate of 1"/min. and an anticipatory distance of 0.1" to 0.199" can be used.

The deceleration rates used are feasible and a higher fast traverse speed could be used on the tool slide depending on the total effective mass of the system. Throughout the calculations the decelerations used were pessimistically low, 0.5g would be more likely.

To reduce the time of 11 secs, (1) above, the comparator could be modified to give an anticipatory range of 0.075" to 0.100".

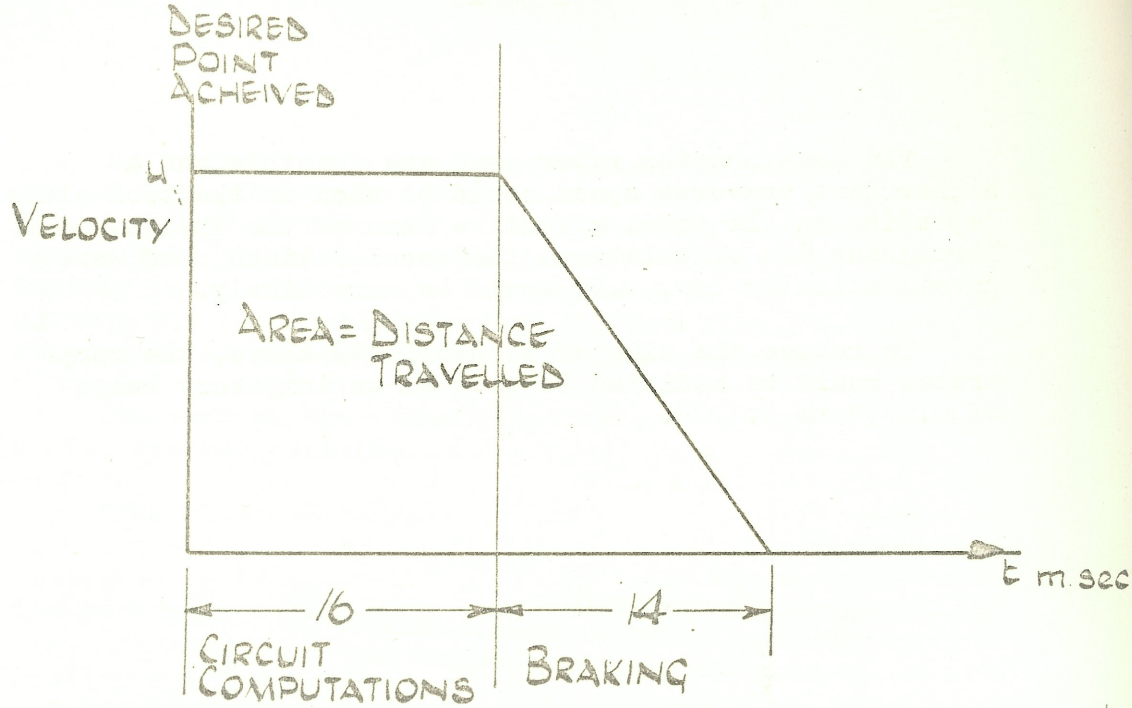


FIG CS4/4/1

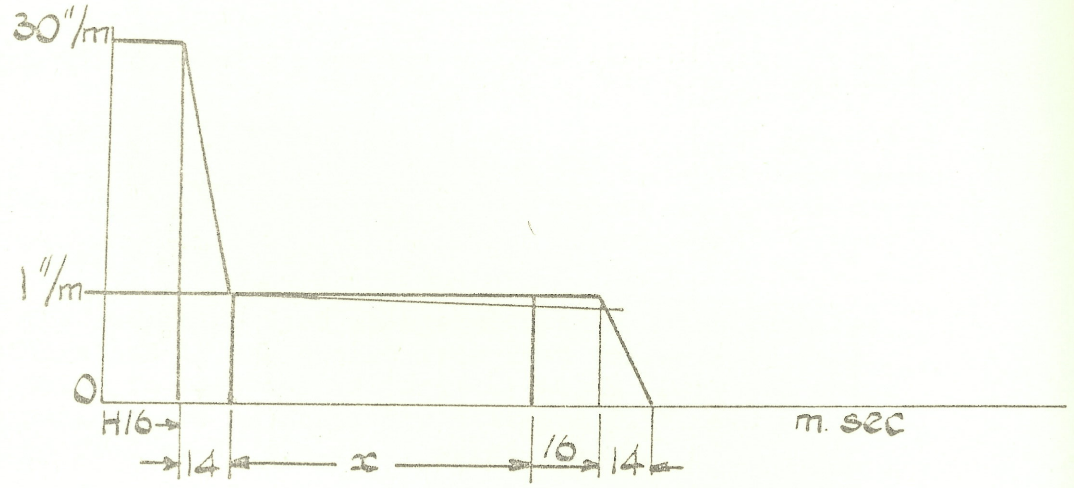


FIG. CS4/4/2

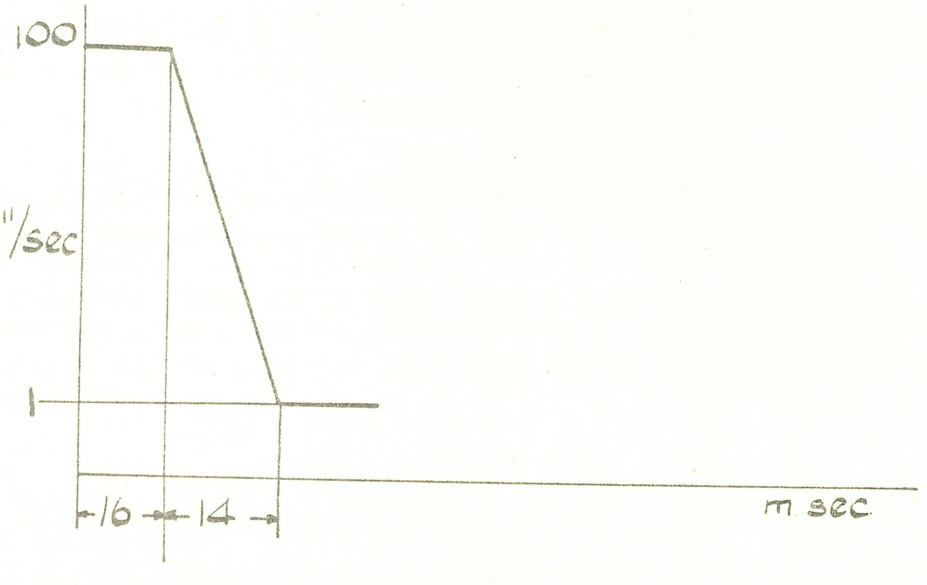


FIG CS4/4/3

C.S.4.5 Velocity Control

The investigation into velocity control has concluded that for this application an A.C. system as initially envisaged is not necessary. It is succeeded by a system for selecting particular velocities digitally. The system was Binary to Analogue converters to provide an analogue control output.

The initial conception was to control velocity by comparing two frequencies. One being representative of the required speed and the other of the actual speed at that instant. Alternatively D.C. levels could be taken as representative in the same way. Frequencies may be produced by oscillators or by resonators. The General Electric Co. of America (Boothe, Ref. 1) use Helmholtz resonators, presumably because the chamber volume may be adjusted to give frequency variation.

The international standard (ISO) for velocity control on a machine such as that being considered is +5% - 3%. It was decided that the maximum linear velocity required would be 30 inches per minute i.e. 3000 rev.per.min. at 0.010 inches per rev.

At an early stage in the investigation a system using basically one fluid proportional amplifier was rejected because it was considered to be insufficiently accurate. Consequently the investigation continued with systems of the type comparing frequencies. It was felt that a frequency representative of actual speed could be obtained from a pick off system from the linear encoder strips.

Basically two resonators are available, being tuned reeds and Helmholtz resonators. G.E. state that reeds are preferable for the control of one particular frequency and that to facilitate for a range would prove too complicated to be worthwhile. It is felt that a simple system could be used employing only one reed, the frequency of which could be variable by means of an adjustable sleeve. See Fig. C.S.4/5/1. The sliding sleeve could prove difficult to position sufficiently accurately.

Situated at the end of the reed is a vane which interrupts flow to the sensing nozzle from the supply nozzle. For a reed frequency of zero, flow is zero, and for increase in frequency flow increases proportionally. A twin reed system is described in detail in Ref. 1 where the system has been successful using air, hydraulic oil and water as the fluid.

The control system illustrated in Fig. C.S.4/5/1 compares the differences between consecutive sums of pressures and frequencies. The resulting output from the comparator making the necessary speed adjustment as required.

Systems using amplitude and phase discrimination are also described in Ref. 1. At one time it was considered that a modified version of the phase discrimination system could be used. However the necessary modifications were too many and difficult so the system was disregarded. The modifications required were: a square to sine wave pulse shaper, a system whereby the resonator need not be adjusted and consequently rely on piston positioning repeatability for accuracy of control, and a compatibility circuit as shown in Fig. C.S.4/5/3. The first of these modifications being the most difficult to achieve. The actual system is described in Ref. 1 and is shown in Figs. C.S.4/5/2 and C.S.4/5/3.

Another alternative control system could be developed and employed. This was the principle of the edgetone, used in a variable mark space ratio mode. However, at this time edgetone devices are not sufficiently developed and the system is not recommended for that reason, although the system could be acceptable in the near future. The system is illustrated in Fig. C.S.4/5/4. It is appreciated that it could be undesirable to superimpose two control frequencies onto the basic edgotone frequency. In order to overcome this situation a D.C. level should be used in preference.

The required feeds were determined from Ref. 2. In fact the initial minimum of 0.001 inches per revolution at 100 revolutions per minute i.e. 0.1 inches per minute was considered to be too slow. The range was determined to be as follows:-

> 1 creep, 3, 8, 12, 16, 20, 24, 27, 30 and 100 inches per minute. These speeds will be programmed on the tape as 1 to 10 (Binary) respectively, 100 inches per minute being fast traverse.

After a discussion with Mr. Cunningham of N.E.L., as previously mentioned, it was decided that a velocity control system of comparable sophistication to those previously mentioned was unnecessary. Consequently a velocity selection circuit was designed and adopted. See Fig. C.S.4/5/5. The system would be duplicated for each linear axis.

The system consists of six sections, comparator, back pressure tape reader and logic, forward and reverse banks of diodes and Binary converters, additional stop circuit, and the final proportional amplifier moog valve drive system.

The comparator determines the direction of movement, creep, and may command stop by 0 0 for forward and reverse. If the comparator exhibits 1 1 a malfunction is indicated within the comparator. The tape reading logic system passes speed information to both banks of valves and Binary to Analogue converters. The signal from the comparator inhibits the banks accordingly. The output of the desired direction bank then drives the moog valve system as required. The additional stop circuit is a safety circuit. The box indicating "POWER OFF" in the circuit diagram represents the condition of compressor failure.

The step up relay is used for amplification purposes.

The reason this system has been chosen is primarily that it is relatively simple and of sufficient accuracy. It uses basically only nine elements, eight foil valves, two converters and a moog valve.

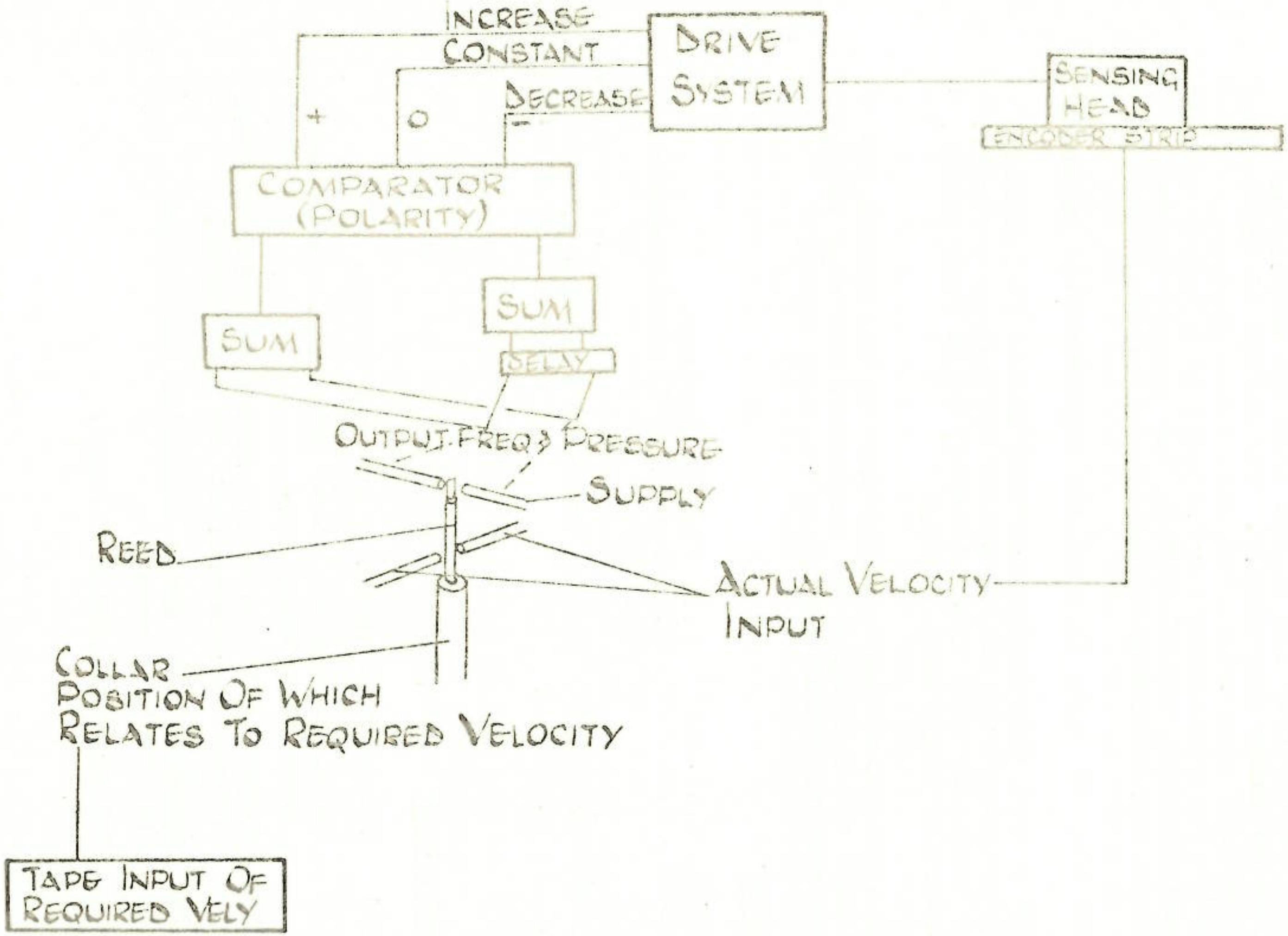
An approximate cost of this part of the system, per linear axis not including the comparator and tape reader would be just below £20 excluding the moog valve.

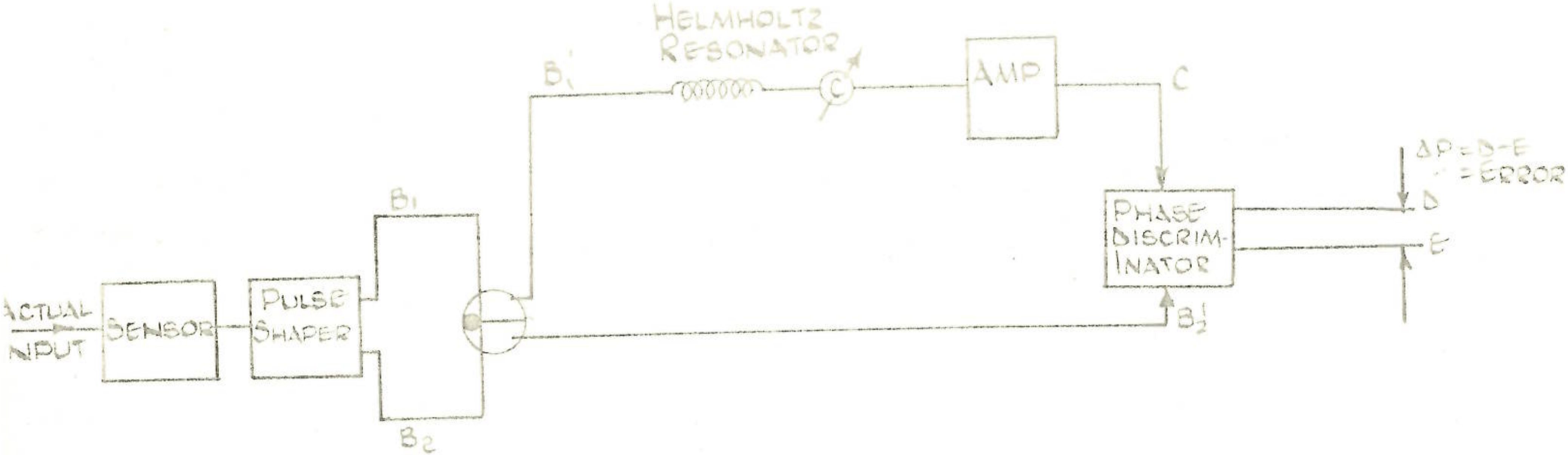
REFERENCES

1. BOOTHE, W.A. and WOODSON, C.W. — A.C. Fluidic Systems. G.E. Technical Information Series 66.C.434.

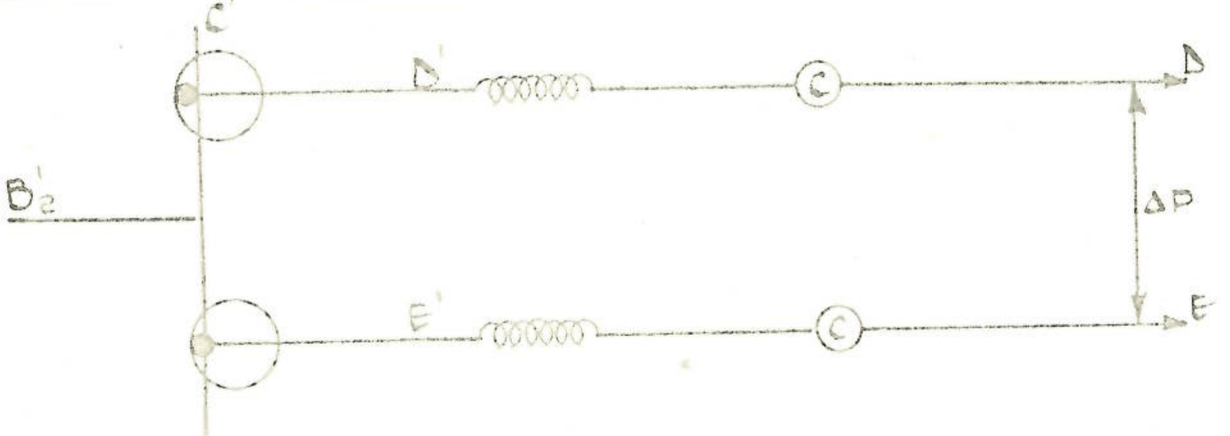
2. WRIGHTBAKER, H. Ed. — Modern Workshop Technology.

LIST OF FIGURES


INCREASE
CONSTANT
DECREASE
DRIVE SYSTEM
+
0
-
COMPARATOR (POLARITY)
SUM
SUM
DELAY
SENSING HEAD
ENCODER STRIP
OUTPUT FREQ & PRESSURE
SUPPLY
REED
ACTUAL VELOCITY INPUT
COLLAR POSITION OF WHICH RELATES TO REQUIRED VELOCITY
TAPE INPUT OF REQUIRED VELY

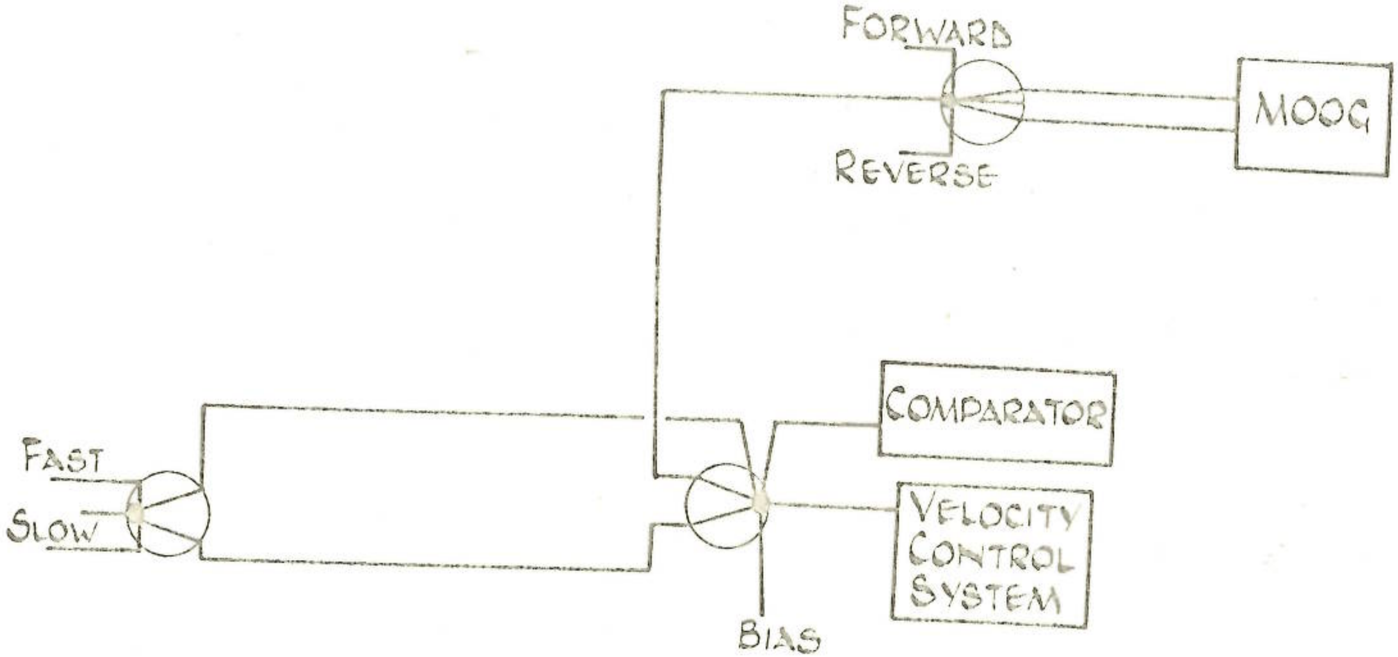


BLOCK DIAGRAM FOR PHASE DETECTION SYSTEM



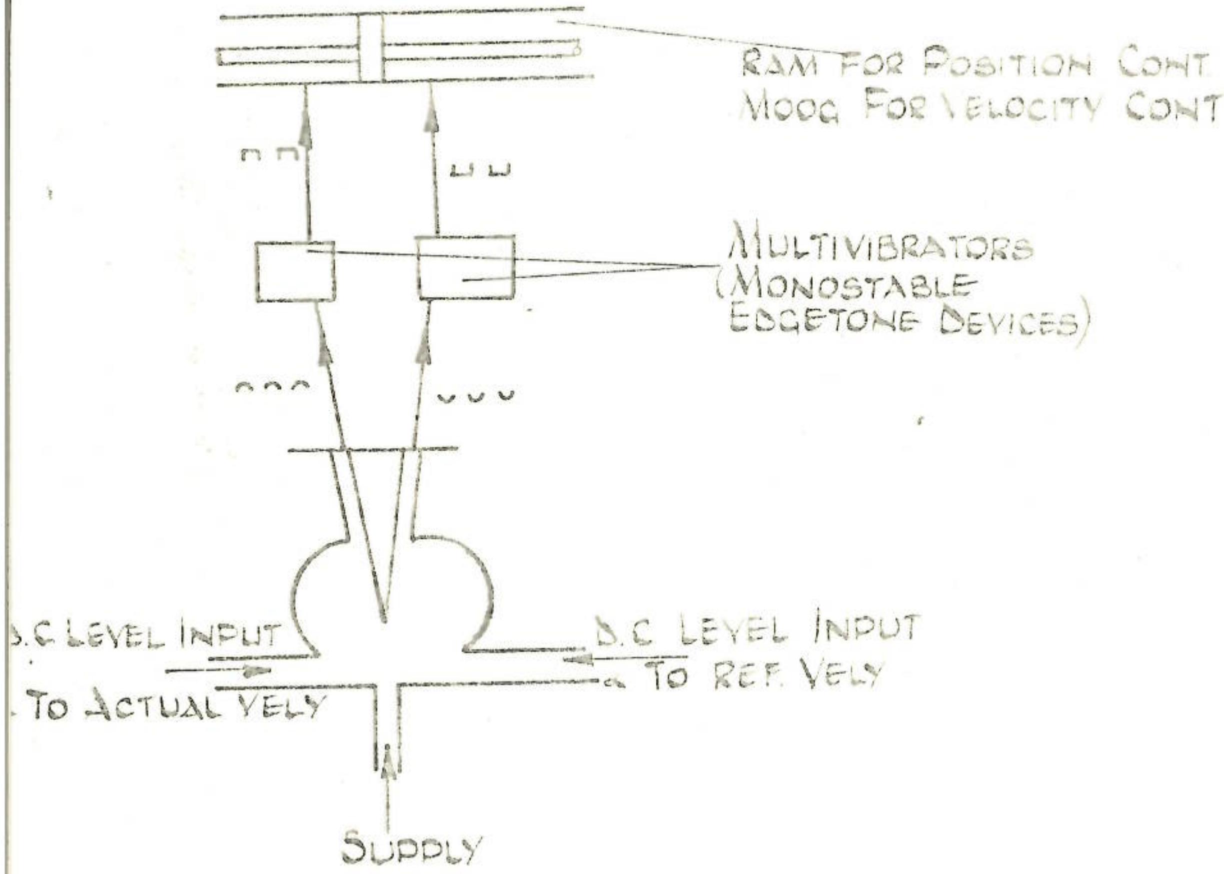
PHASE DISCRIMINATOR

FIG CS4/5/2



COMPATIBILITY CIRCUIT BETWEEN VELOCITY CONTROL (PHASE DISC)

AND MOOG VALVE

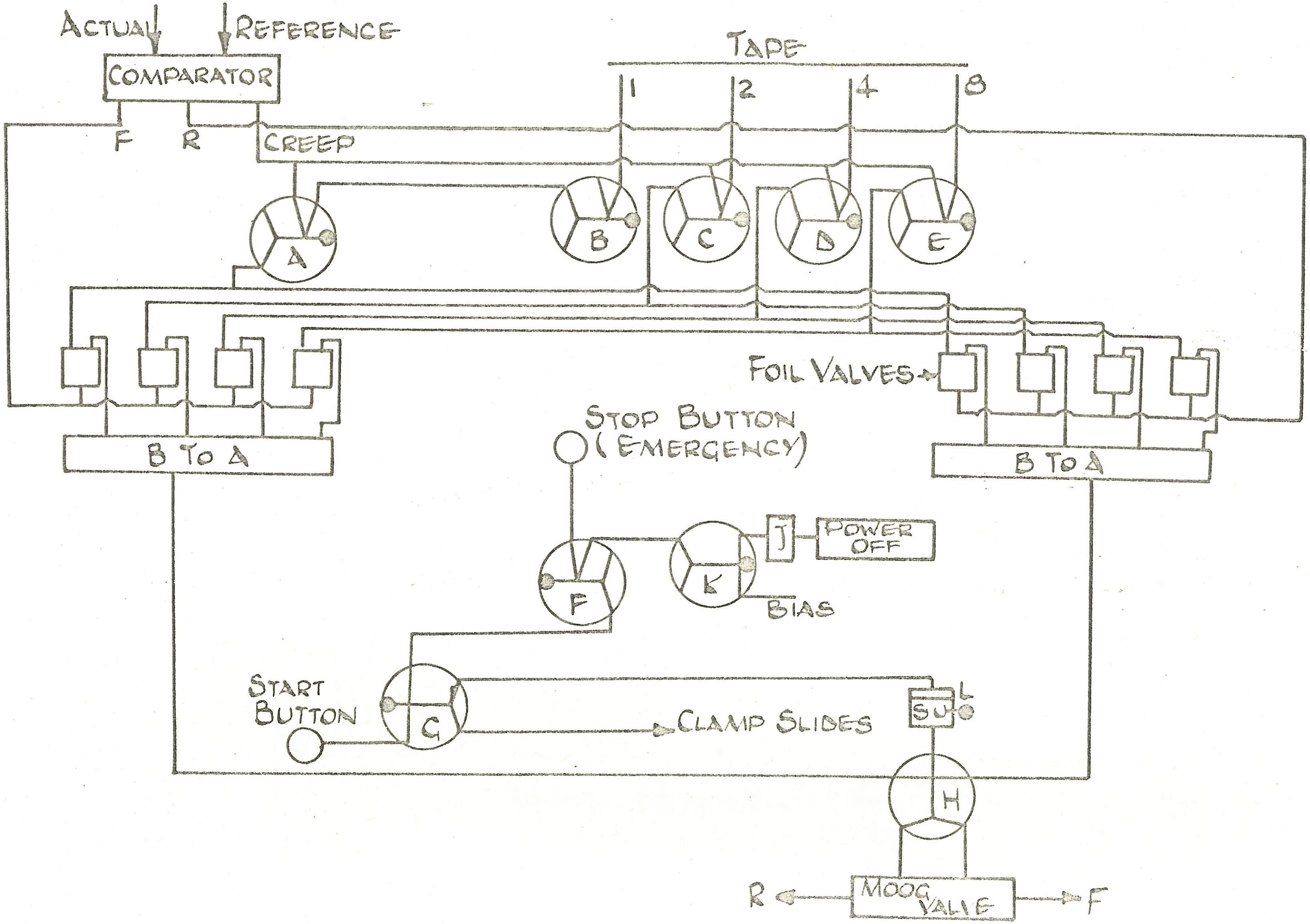


EDGETONE MARK SPACE

MODULATOR SYSTEM

HELMMOLTZ RESONATOR

FIG CS4/5/4



KEY
A-F 2 INPUT ORNOR
G, & K. BISTABLE
H, PROPORTIONAL AMPLIFIER
J STEPDOWN RELAY
L STEP UP RELAY

68

VELOCITY SELECTION CIRCUIT

FIG CS4/5/5

C.S.4.6 Rotary Encoder

A rotary encoder is a device to convert an angular position of a shaft into a digital form (Ref. 1). It consists essentially of a number of engraved marks or grooves around the periphery of the shaft (or disc attached to it) which divides the circle into convenient subdivisions. The marks or grooves themselves can then be made to generate pulsed signals as they are scanned by a pick-up device.

Requirements for the Encoder

1) To measure an angular displacement of shaft in steps of .5 degree.
2) That it be an absolute measuring device.
3) Diameter of encoder disc should be as small as possible. The size of the finest digit is chosen as .02" which gives the diameter of the disc as 4.585".

Design of Encoder

To measure an angular displacement of $.5^{\circ}$ with an absolute measuring system it is required to design an encoder with 4-decades as follows:

1) 100° - Decade: - To measure 360° in steps of 100°.
2) 10° - Decade: - To measure 100° in steps of 10°.
3) 1° - Decade: - To measure 10° in steps of 1°.
4) $.5^{\circ}$ - Decade: - To measure 1° in steps of $.5^{\circ}$.

Codes

Used in this encoder is a "creeping code", a code with a property that successive intervals differ in one bit only (Table C.S.4/6/2B); such a code does not require a special technique to avoid ambiguous read-out and at the same time possesses an inherent ease of manufacture.

100° - Decade

The hundred degree decade is designed to measure 360° in steps of 100°. It consists of a disc of 4.585" diameter and is 1/8" thick. A set of 3 nozzles is placed around the disc so that the axes of the nozzles are normal to the circle.

The shape of the disc and the relative position of the nozzles is shown in Fig. C.S.4/6/1B.

Two nozzles (4-coded) are enough to measure 360° in an interval of 100°. But as 360° is not an exact multiple of 100°, the width of the groove and the land on the periphery of the disc are not equal. From Fig. C.S.4/6/1B it will be noticed that the groove subtends an angle of 200° at the centre of the disc, while the land occupy only 160°. This inevitable unequal distribution of groove and land on the circumference of the disc results in an inaccurate reading after 200°. This is illustrated in Fig. C.S.4/6/1A which shows a disc and a set of two nozzles. It is assumed that the disc is fixed and the set of nozzles is moving around the disc in a clockwise direction.

Position-1, Position-2 etc. are the positions, where the nozzles sense the change of signals. The signals received by the nozzles at these four positions are given in Table C.S.4/6/1A. Position-4 senses the change of signal 40° earlier than the required position, i.e. 300° is read, when the correct position is 260°.

In order to eliminate this incorrect read-out of the sensor, it is suggested that 3 nozzles are used instead of 2. These nozzles are to be mounted so that the axes of the two adjacent nozzles make angles of 60° and 40° at the centre of the disc (Fig. C.S.4/6/1B) respectively.

The signals received by these nozzles and the corresponding angular displacements of the nozzles with respect to the disc are given in Table C.S.4/6/1B. This table contains 6-codes but only 4-codes are required to sense 360° in steps of 100°. The two redundant codes are recognised by studying Table C.S.4/6/1B. A new Table C.S.4/6/1C is constructed, which shows that both the codes 001 and 011 read 100° - 200°. Similarly the two codes 111 and 110 read 200° - 300°. A logic circuit for decoding these 6 creeping codes to 4 binary codes is shown in Fig. C.S.4/6/1C.

<u>10° - Decade</u>

Fig. C.S.4/6/2A shows a coded disc of 4.585" diameter and 1/8" thickness and a set of 5 nozzles mounted around the periphery of the disc. The adjacent nozzles make an angle of 10° at the centre of the disc. As the nozzles move around the disc, they receive signals according to the creeping code pattern until they have turned through 310°. At 310° the situation of the nozzles is shown at Position-1 and the subsequent positions are at Position-2, Position-3 etc.

At Position-1, the nozzles read 00001, but at the next position of changing of signals, the nozzles read 00010, which does not follow the creeping code pattern and neither does the following positions shown in Table C.S.4/6/2A. The nozzles situated anywhere between 0° and 310° will sense signals according to the code pattern shown in Table C.S.4/6/2A. A logic circuit can be designed to decode these two codes (mentioned in Table C.S.4/6/2A and C.S.4/6/2B) to binary. The same disc will produce 2 different codes, depending on the position of the nozzles with respect to the disc and can be used to measure the angular displacement of 10°. But Table C.S.4/6/2A indicates that the successive change of the signals differs in 2 bits and as a result ambiguous readings are expected anywhere inbetween 310° and 360° measured clockwise.

To eliminate these ambiguous read-outs, it is suggested to use 2 sets of 5 nozzles, instead of one set. The arrangement of nozzles and the disc is shown in Fig. C.S.4/6/2B. The two sets of nozzles are mounted so that when one set is inbetween 310° and 360°, the other is outside this region.

The angular distance between the two identical nozzles a and a', b and b' etc. must be 100° or a multiple of 100°, so that these identical nozzles will read the same signals, except when one set is in the region of 310° to 360°. When for a particular instance, one set is in the above mentioned region, this set will receive a signal according to the code shown in Table C.S.4/6/2A, while the other set sense the creeping code pattern, as shown in Table C.S.4/6/2B. If the signals from the two identical nozzles are fed as an input signal to an OR-gate, before being decoded, then a creeping code signal will be received at the input of the decoder for any position of the set of nozzles. This means an extra five elements for the complete elimination of ambiguity due to unequal width of the land after 350°. The connection of the nozzles to the OR-gate is shown in Fig. C.S.4/6/2C.

1° - Decade

Fig. C.S.4/6/3 shows a disc with 72 slots on the periphery of the disc. The slot subtends an angle of 5° at the centre of the disc. A set of 5 nozzles are placed around the periphery of the disc. The angular distance between adjacent nozzles is 11°. As the disc rotates, the nozzles receive signals according to the code pattern shown in Table C.S.4/6/2B. A logic circuit for decoding these 10 creeping codes to binary codes is shown in Fig. C.S.4/6/4.

$.5^{\circ}$ - Decade

Consist of a disc of 4.585" diameter and 1/8" thickness, and a nozzle placed around the periphery of the disc.

This disc has 360 slots of .02" wide milled equispaced on the periphery of the disc. The nozzle is to sense signalling codes "0" and "1" for an angular displacement of 0° and $.5^{\circ}$ respectively. The output of the nozzle is fed directly into the comparator, no decoder is required.

Decoder

In general decoders must perform the reverse function of an encoder, they must recognise a multiple input signal combination and convert into a code suitable for comparison. The problem involved here is to decode a creeping code into binary code for a comparator which can recognise a binary code only. The design of such a decoder can be explained by studying these two codes given in Table C.S.4/6/2B.

From the Table C.S.4/6/2B:

$$W = 1, \text{ when } W = AB\bar{C}\bar{D}\bar{E} + A\bar{B}\bar{C}\bar{D}E$$

$$= A\bar{C}\bar{D}E \text{ (for decimal 8 and 9)}$$

It can be seen from the Table C.S.4/6/2B that $W = 1$, occurs only when $A = 1$, $C = 0$

$$\text{i.e. } W = A\bar{C} \qquad (1)$$

Similarly

$x = 1$

$$\text{when } x = \bar{A}BCDE + ABCDE + ABCD\bar{E} + ABC\bar{D}\bar{E}$$

$$= BCDE + ABCE \text{ (for decimal 4 to 7)}$$

$$\text{or } x = BC \qquad (2)$$

$$y = 1, \text{ when } y = \bar{A}\bar{B}\bar{C}DE + \bar{A}\bar{B}CDE + ABCD\bar{E} + ABC\bar{D}\bar{E}$$

$$= \bar{A}\bar{B}DE + ABC\bar{E}$$

$$\text{or } y = D\bar{B} + C\bar{E} \qquad (3)$$

$$Z = 1, \text{ when } z = \bar{A}\bar{B}\bar{C}\bar{D}E + \bar{A}\bar{B}CDE + ABCDE + ABC\bar{D}\bar{E} + A\bar{B}\bar{C}\bar{D}\bar{E}$$

$$\text{or } z = E\bar{D} + C\bar{B} + AE + C\bar{D} + A\bar{B} \qquad (4)$$

SUMMARY:

$$W = A\bar{C} \qquad (1)$$

$$x = BC \qquad (2)$$

$$y = D\bar{B} + C\bar{E} \qquad (3)$$

$$z = E\bar{D} + C\bar{B} + AE + C\bar{D} + A\bar{B} \qquad (4)$$

A logic circuit for the above expression is given in Fig. C.S.4/6/4.

Decoding 6 creeping codes into 4 binary codes

From Table C.S.4/6/1C

$$x = 1, \text{ when } x = ABC + AB\bar{C} + A\bar{B}\bar{C}$$
$$\text{or } x = A \qquad (1)$$

$$y = 1, \text{ when } y = \bar{A}\bar{B}C + \bar{A}BC + A\bar{B}\bar{C}$$
$$\text{or } y = C\bar{A} + A\bar{B} \qquad (2)$$

SUMMARY:

$$x = A \qquad (1)$$
$$y = C\bar{A} + A\bar{B} \qquad (2)$$

A logic circuit for the above expression is given in Fig. C.S.4/6/1C.

Angular Velocity of the Encoder

Angular velocity of the encoder at which the sensing can take place depends on the following:-

1) Width of the finest digit.

2) Time elapsed between sensing of a signal and the stopping of the moving parts i.e. the logic time delay and the machine parts inertia.

Time Delay

N = RPM of the encoder.
V = Constant velocity of the moving parts of the encoder in inch/min.
W = Width of the finest digit = .02".
L = Total length of pipe in interconnection in ft. = 4ft.
t_D = Time delay in decoding in millisec.
t_p = Time delay in pipe in millisec.
t_c = Time delay in comparing in millisec.

Delay in pipe

The signal is propagated through the pipe at the speed of sound; at N.T.P. approximately 1 ft/millisec.

$$t_p = L \times 1 = 4 \times 1 = 4 \text{ millisec.}$$

Delay in Switching

Switching time for an element is .55 millisec.

$t_D = N_D \times .55$ where N_D = max. number of elements a signal passed through

$= 3 \times .55 = 1.65$ millisec.

Delay in Comparator

$Te = Nc \times .55$ Nc = maximum number of elements a signal passed through

$= 4 \times .55$

$= 2.2$ millisec.

Delay in brake actuation $= T_B = 20$ millisec.

T = Total time $= t_p + t_D + t_e + T_B = 4 + 1.65 + 2.2 + 20$
$= 27.58$

Distance travelled by the moving parts of the encoder during time T; W = V x T

$$\therefore W = VT$$

$$V = \frac{W}{T} = \frac{.02 \times 1000}{27.85}$$

$$= \frac{20}{27.85} \text{ inch/sec.}$$

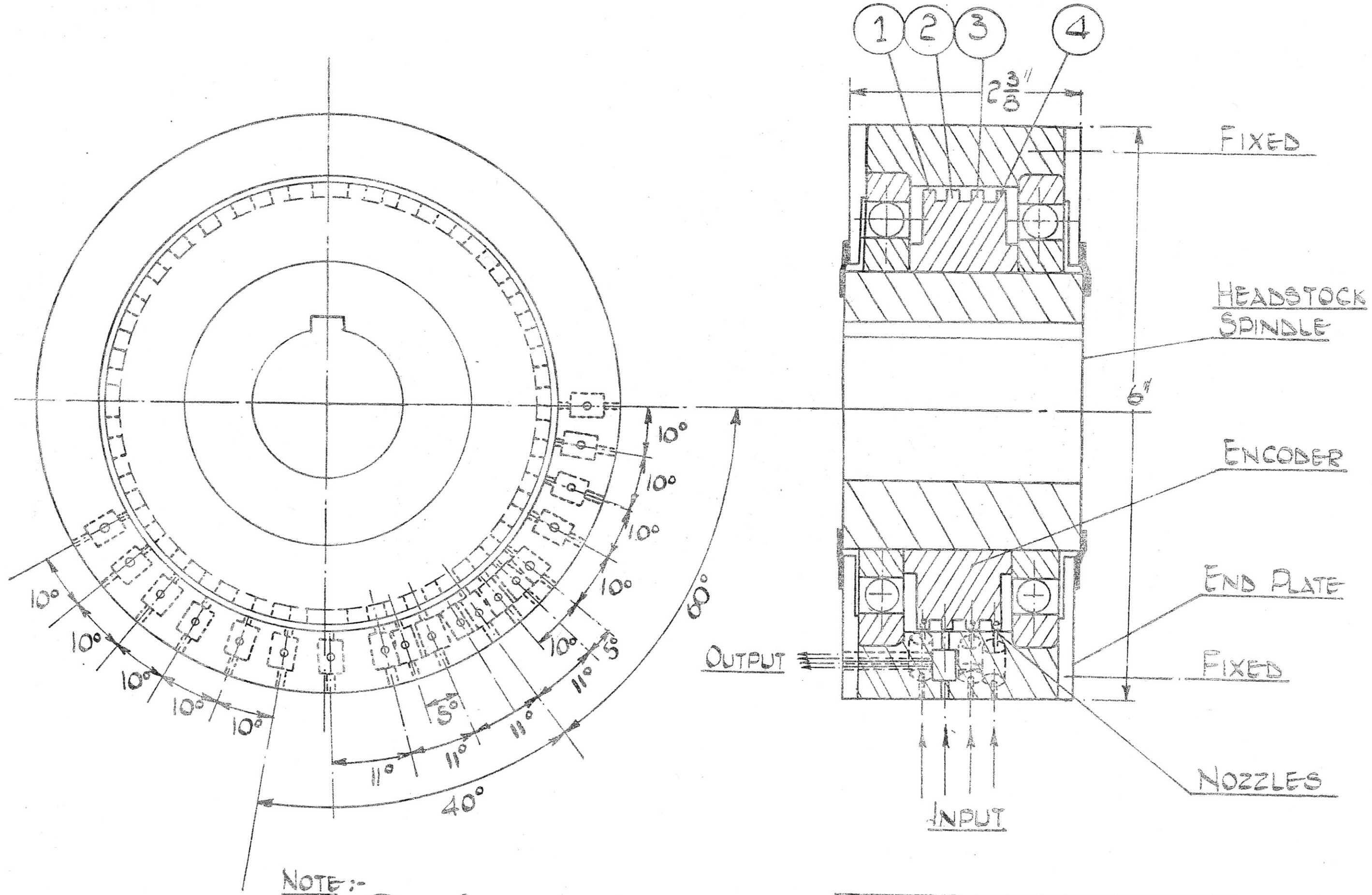
$$N = \frac{20 \times 60}{27.85 \times \pi \times 4.585} = 3\text{R.P.M.}$$

Allowing 50% margin, to sense the finest digit, the encoder is required to rotate at a speed not more than 1.5 R.P.M.

Application

The output of the decoding logic feeds into 4 binary comparators, Ref. 1. For sensing of all digits a maximum of rotational speed of 3 R.P.M. can be used. It is suggested that a maximum rotational speed of 1.5 R.P.M. is used. The output of the three comparators are fed into a 4 input OR/NOR, the NOR of which give total equivalence. This signal is then utilized to actuate the disc brake.

Ref. 1 Drilling System Design Project 1967. Final report of control systems committee. College of Aeronautics, Department of Production and Industrial Administration.



NOTE :-
1 = ½°- DECADE
2 = 1°- DECADE
3 = 10°- DECADE
4 = 100°- DECADE

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS
DRAWN	APP'VD.	TITLE :- ROTARY ENCODER			
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.			SCALE FULL	DRAWING No. PT/CS/A3/115	SHT. OF Sh.

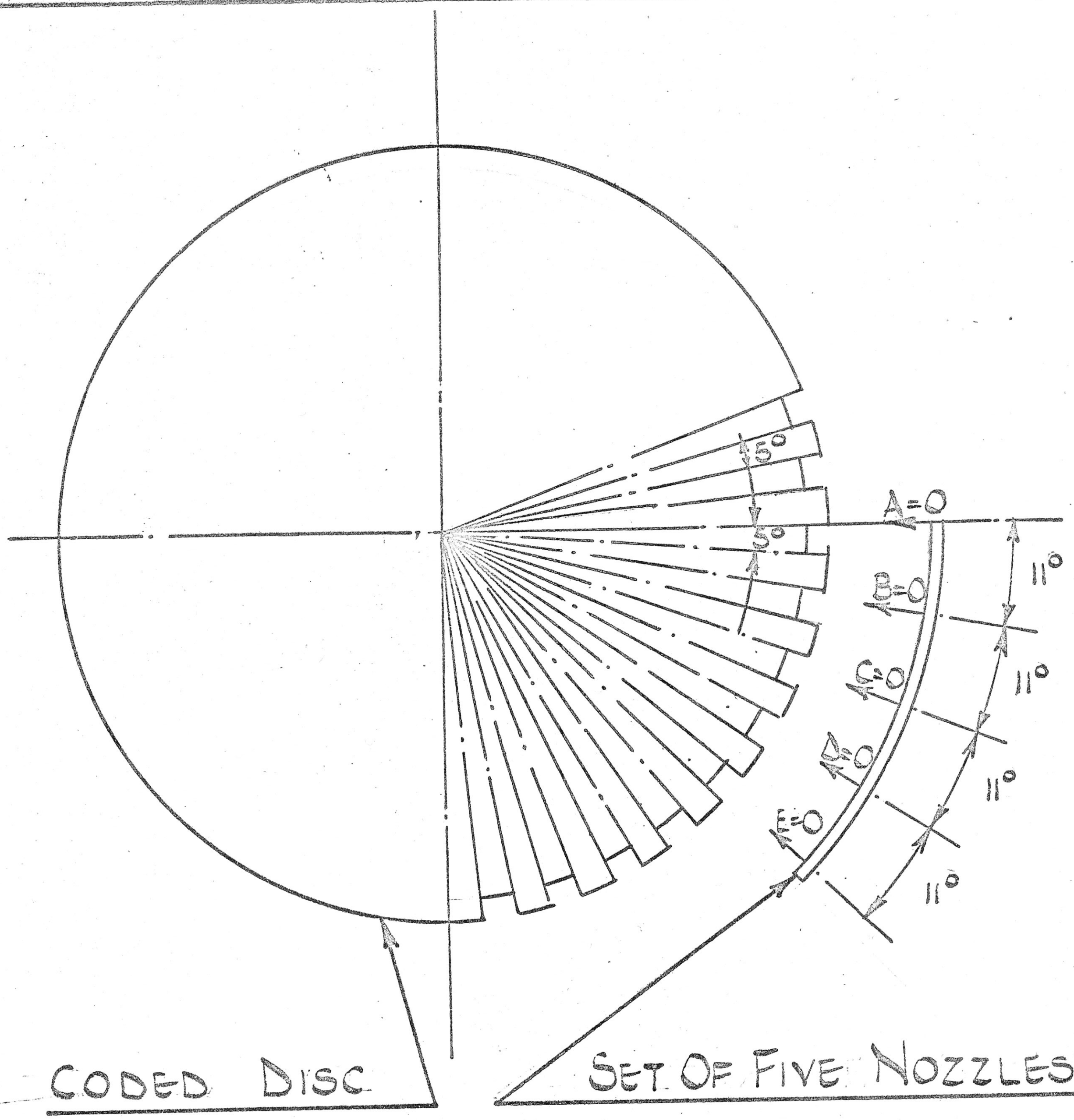


FIG - CS4/6/3

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN	APPVD.	TITLE:-
H. BERA		1° DECADE (ROTARY ENCODER)

	SCALE	DRAWING No.
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD. BEDFORD.		PT/CS/A4/120
		SHT. OF SHTS

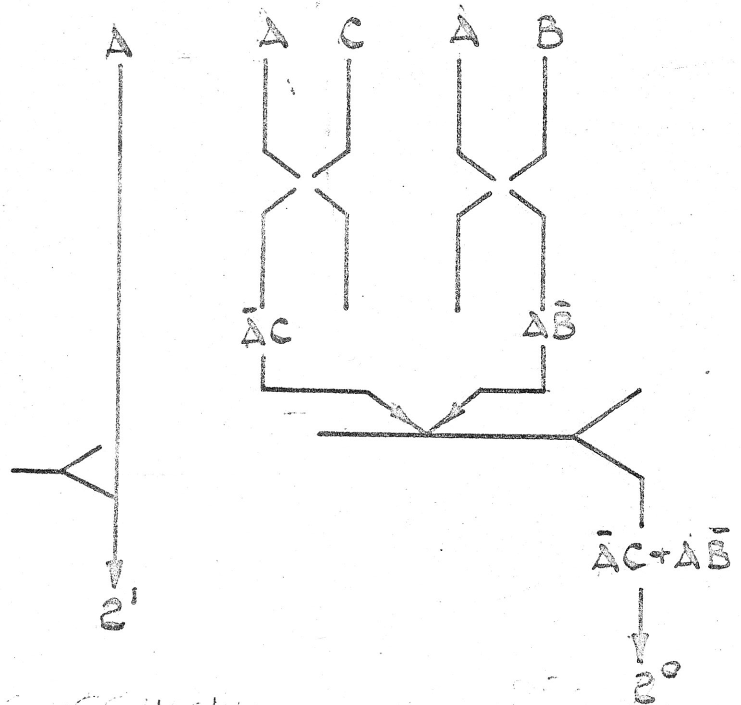


FIG - CS4/6/1C 100° DECADE DECODER

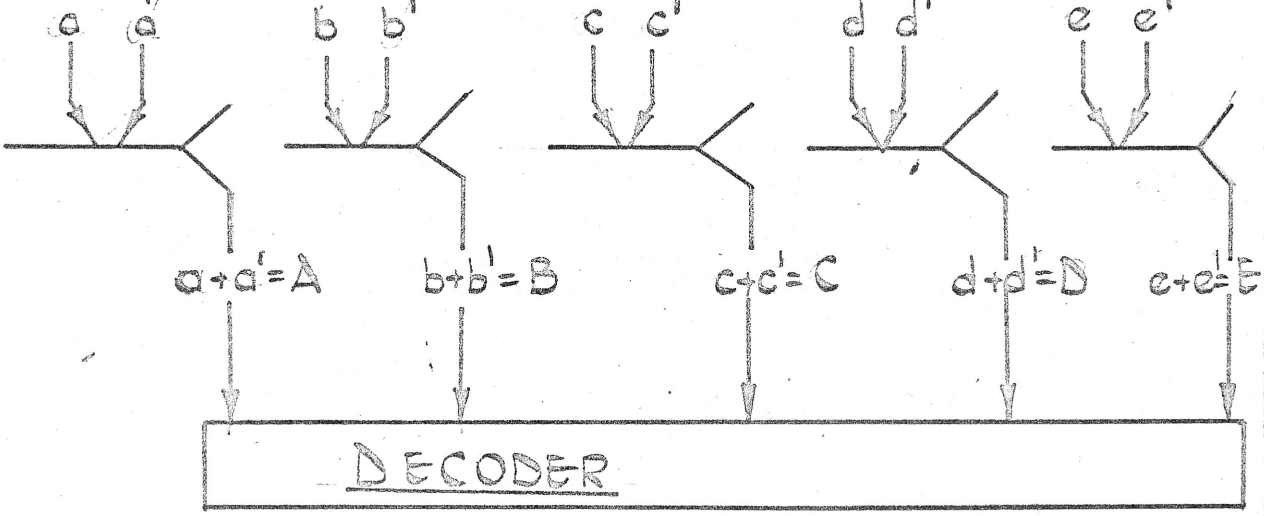


FIG CS/4/6/2C 10° DECADE ERROR DETECTOR

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN: H. BERA

APPVD.:

TITLE :- DECODING LOGIC.

DEPT. OF PRODUCTION ENGINEERING,
COLLEGE OF AERONAUTICS,
CRANFIELD, BEDFORD.

SCALE

DRAWING No: PT/CS/A4/122.

SHT. OF SHT:

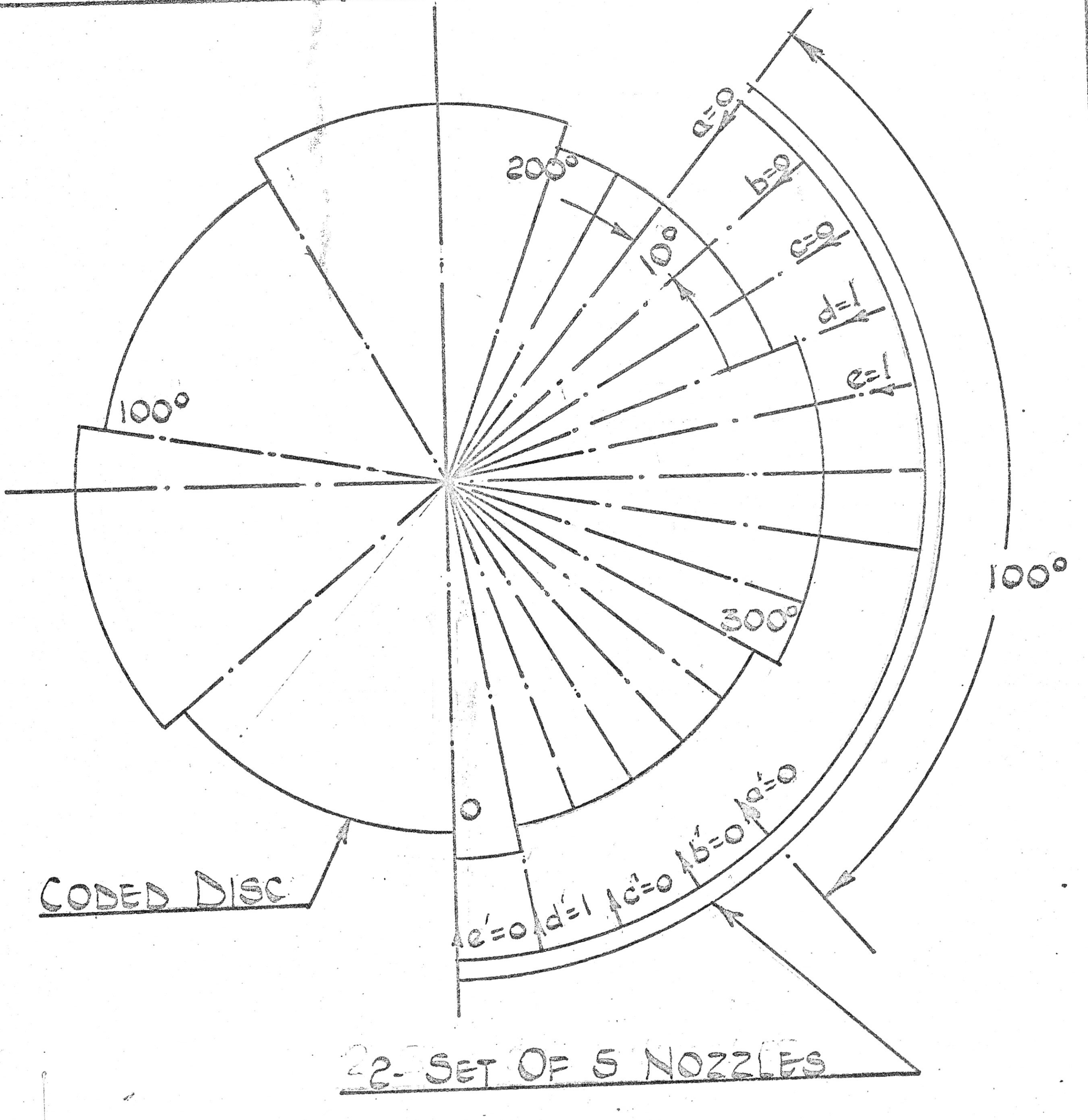


FIG - CS4/6/2B

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN	APPVD.	TITLE :-
H. BERA		10° DECADE (ROTARY ENCODER)

	SCALE	DRAWING N.
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.		PT/CS/A4/119
		SHT. OF SHTS.

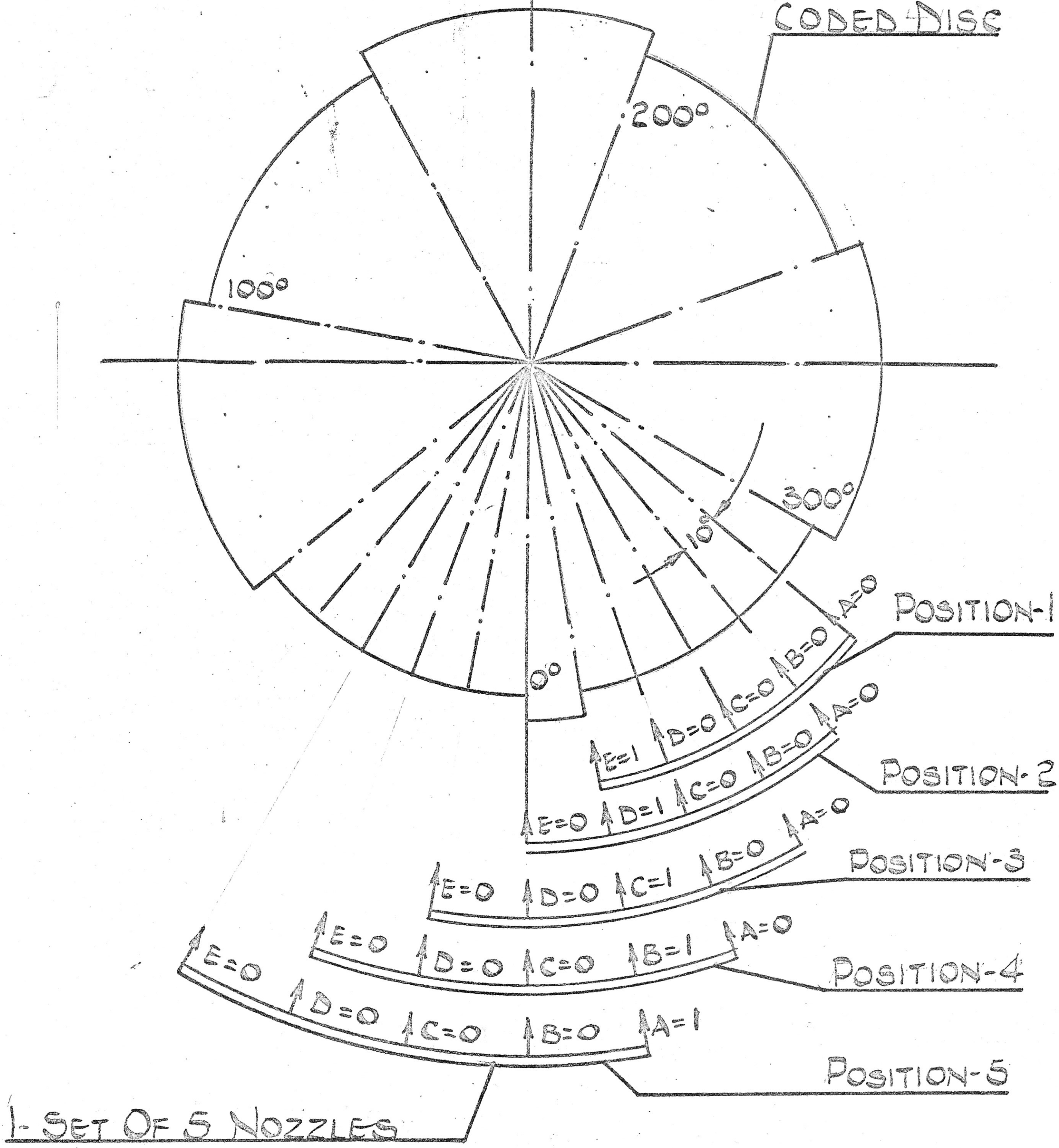


FIG - CS4/6/2A

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN	APPVD.	TITLE :-
H. BERA		10° DECADE (ROTARY ENCODER)

	SCALE	DRAWING No.
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.		PT/CS/A4/118
		SHT. OF SHTS.

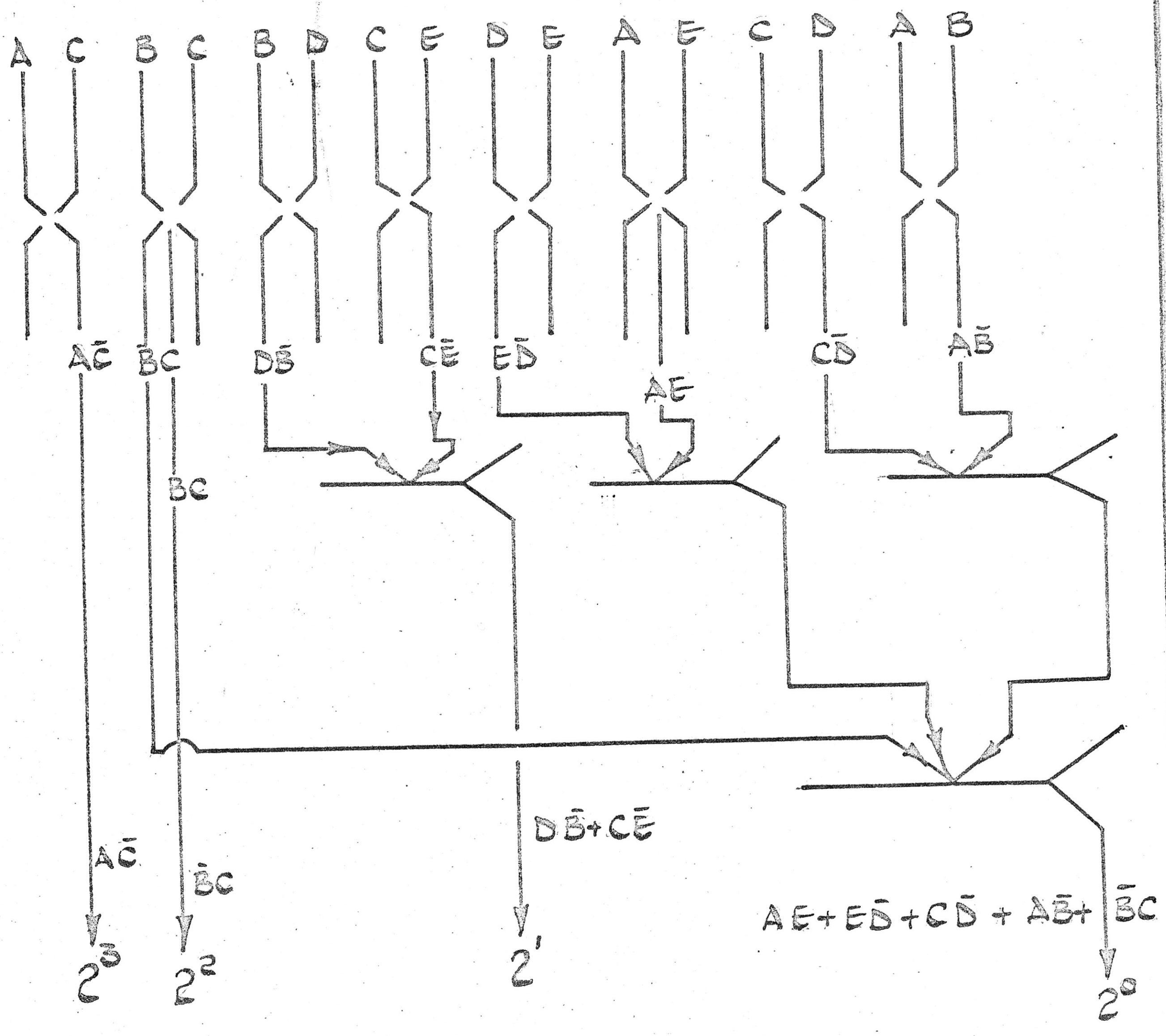


FIG CS4/6/4 10° AND 1° DECADE DECODER

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN H. BERA | APPVD. | TITLE:- DECODING LOGIC

DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.

SCALE | DRAWING No. PT/CS/A4/121 | SHT. OF SHTS

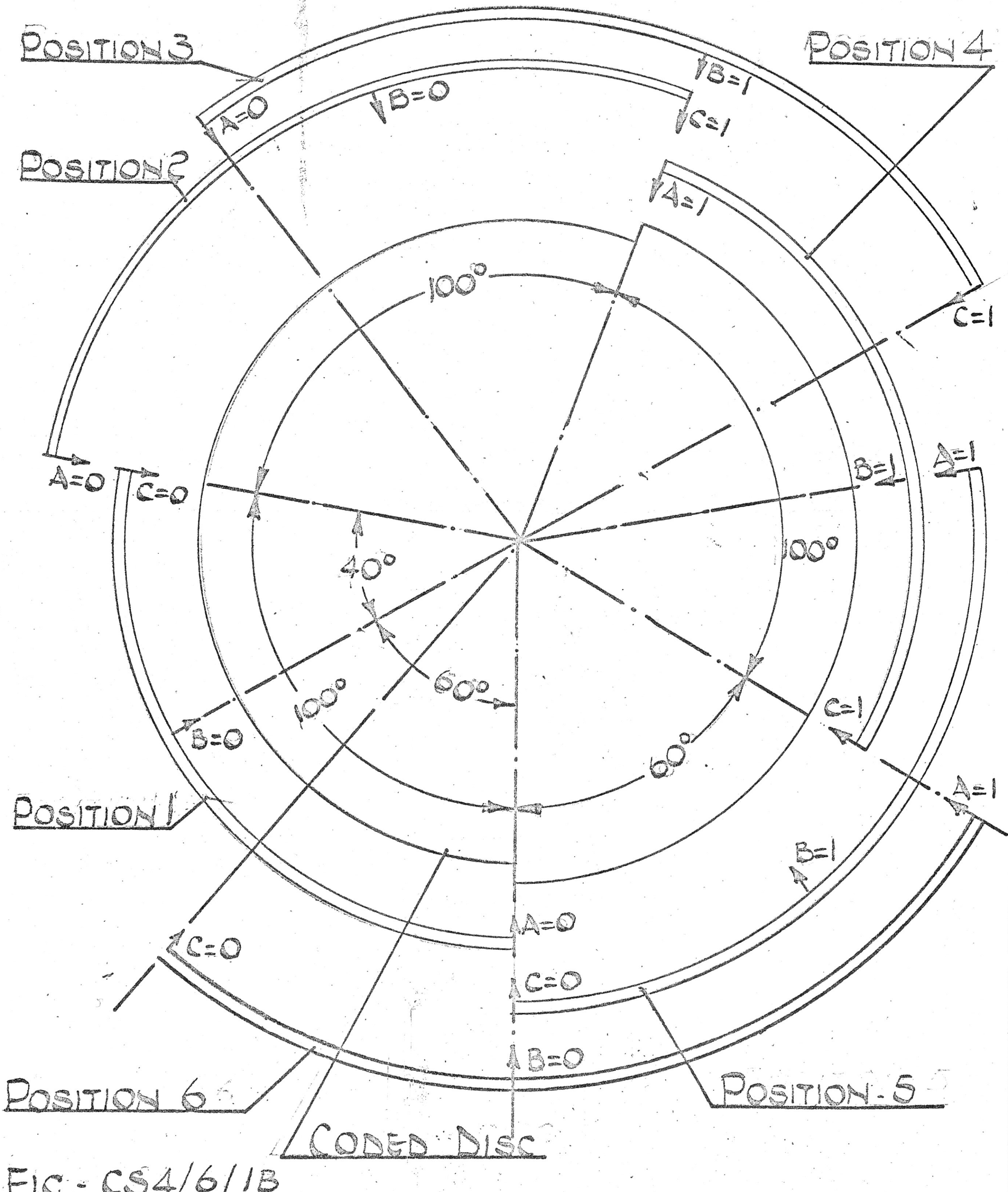


FIG - CS4/6/1B

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN	APPVD.	TITLE:-
H. BERA		100° DECADE (ROTARY ENCODER)

	SCALE	DRAWING No.
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.		PT/CS/A4/117 SHT. OF SHTS.

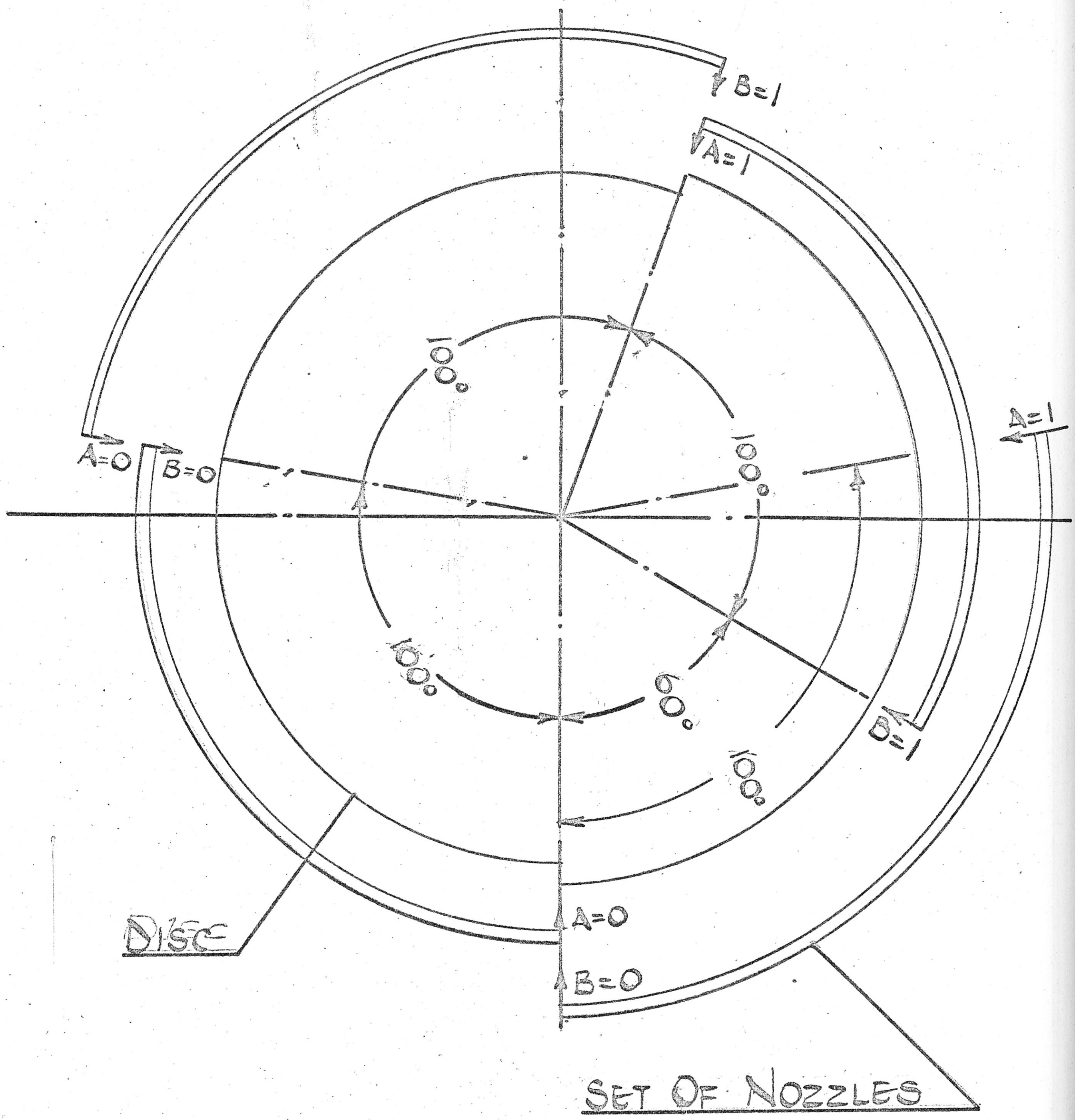


FIG-CS4/6/1A

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN	APPVD.	TITLE :-
H. BERA		100° DECADE (ROTARY ENCODER)

	SCALE	DRAWING No.
DEPT. OF PRODUCTION ENGINEERING, COLLEGE OF AERONAUTICS, CRANFIELD, BEDFORD.		PT/CS/A4/116
		SHT. OF SHT

CREEPING CODES			
NOZZLES AT	NOZZLES A	NOZZLES B	ANGLE IN DEGREE
POSITION-1	0	0	0-100
POSITION-2	0	1	100-200
POSITION-3	1	1	200-260
POSITION-4	1	0	260-360

TABLE - CS4/6/1A

NOZZLES AT	NOZZLES A	NOZZLES B	NOZZLES C	ANGLE IN DEGREE	
POSITION-1	0	0	0	0-100	0-100
POSITION-2	0	0	1	100-140	100-200
POSITION-3	0	1	1	140-200	
POSITION-4	1	1	1	200-260	200-300
POSITION-5	1	1	0	260-300	
POSITION-6	1	0	0	300-360	300-360

TABLE - CS4/6/1B

CREEPING CODES					BINARY CODES	
NOZZLES AT	NOZZLES A	NOZZLES B	NOZZLES C	ANGLE IN DEGREE	X	Y
POSITION-1	0	0	0	0-100	0	0
POSITION-2	0	0	1	100-200	0	1
POSITION-3	0	1	1			
POSITION-4	1	1	1	200-300	1	0
POSITION-5	1	1	0			
POSITION-6	1	0	0	300-360	1	1

TABLE - CS4/6/1C

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN: H. BERA
APPVD.
TITLE :- CODES

DEPT. OF PRODUCTION ENGINEERING,
COLLEGE OF AERONAUTICS,
CRANFIELD, BEDFORD.

SCALE | DRAWING N

SHT. OF SI

NOZZLES AT	NOZZLES					ANGLE IN DEGREE
	A	B	C	D	E	
POSITION-1	0	0	0	0	1	310-320
POSITION-2	0	0	0	1	0	320-330
POSITION-3	0	0	1	0	0	330-340
POSITION-4	0	1	0	0	0	340-350
POSITION 5	1	0	0	0	0	350-360 *

TABLE - CS 6:2A

DECIMAL No	5-BITS CREEPING CODES					4-BITS BINARY			
	A	B	C	D	E	W	X	Y	Z
0	0	0	0	0	0	0	0	0	0
1	0	0	0	0	1	0	0	0	1
2	0	0	0	1	1	0	0	1	0
3	0	0	1	1	1	0	0	1	1
4	0	1	1	1	1	0	1	0	0
5	1	1	1	1	1	0	1	0	1
6	1	1	1	1	0	0	1	1	0
7	1	1	1	0	0	0	1	1	1
8	1	1	0	0	0	1	0	0	0
9 *	1	0	0	0	0	1	0	0	1

TABLE - CS4/6/2B

ITEM	PART No.	DESCRIPTION	No. OFF	MTL.	REMARKS

DRAWN: H. BERA
APPVD.:
TITLE :- CODES

DEPT. OF PRODUCTION ENGINEERING,
COLLEGE OF AERONAUTICS,
CRANFIELD, BEDFORD.

SCALE | DRAWING | SHT. OF SHT

C.S.4.7 Tape Reader

A block-reader or line-reader could be used for this application, a block-reader is preferred as this allows the simultaneous reading of all information relevant to an operation.

I.S.O. standard codes with parity check should be used throughout, this allows fifty-two different lines to be used. Such a large number of different lines is not required by this application.

No tape reader has been designed for this project but a report of the design of a suitable reader is given in paper E2, 2nd Cranfield Fluidics Conference.

A probable cost is £70.

C.S.4.8 Cost Estimates of a Control Logic System

		£.	s.	d.		No. of elements
Linear:-	two axes					
	Encoders	200	-	-		40
	Decoders	56	-	-		104
	Comparators	61	-	-		98
					317 - -	
Rotary:-	one axis					
	Encoder	60	-	-		
	Decoders	20	15	-		33
	Comparators	16	5	-		26
					97 - -	
	two axes					
	Velocity control and fail safe logic	40	-	-		18
	Tape reader system	70	-	-		
	Filters	12	-	-		
	Regulators	42	-	-		
	Piping etc.	26	-	-		
					190 - -	
		Total			£604 - -	317

The number of elements shown includes all types of logic elements (including foil elements) and a blanket cost of 12/6 per element has been applied. Element costs are therefore £200. The flow consumption would be of the order of 2000 C.F.H.

C.S.4.9 Machine Management Basic Programme

The following is a machine management sequence. It must be appreciated that loops and sub-loops may be necessary depending on the particular programme. The basic layout (C.S.4/9/1) is provisional only and would require modification after appraisal of the complete machine tool system. From this management sequence a management flow diagram would be prepared. The necessary system interlocks would be in turn derived from this flow diagram.

Manual inputs could be via fluidic oak switches and push buttons or by a plug boarding system.

Obviously futher work needs to be carried out on this section; reference to this is made in the recommendations.

C.S.4.10 Recommendations and Conclusions

Further work is required on several sections of the work detailed in this report.

1. Machine management interlock and sequence requirements and programme layout must be derived in more detail.

2. From (1) the specification of the tape reader can be finalized and consideration given to the relative viability of manufacturing or buying the required unit.

3. Consideration should be given to the feasibility of continuous path control systems. This requires greater resolution on the linear and rotary axes, a sophisticated velocity control and the continuous interlocking of these functions.

4. Digital to Analogue convertors are recommended at various points in the control circuits. At the current state of the art insufficient attention has been applied to the non-moving part type of device.

5. The design of the linear encoders should be looked at with the object of reducing cost. Once the master for the code strips has been obtained, the major cost is the manufacture and assembly of component parts other than the code strip and elements.

6. Elements to detect over-run should be included on the trolley of the linear encoders. The case ends can be used to trigger the over-run elements and the outputs used to indicate system malfunction, and consequently shut the machine down.

7. Circuit manifolding and layout must be done. It is suggested that the linear comparators be placed as close as possible to the linear encoders.

8. The linear comparator network is of a completely new type and requires proving.

9. A progressive velocity decrease to zero would be preferable to the selection of a creep speed.

A low cost fluidic control system has been found to be feasible for the specification given in the technical survey. The system has been designed to be completely compatible with the rest of the machine tool system, but could be applied to other parallel axis ram operated machines.

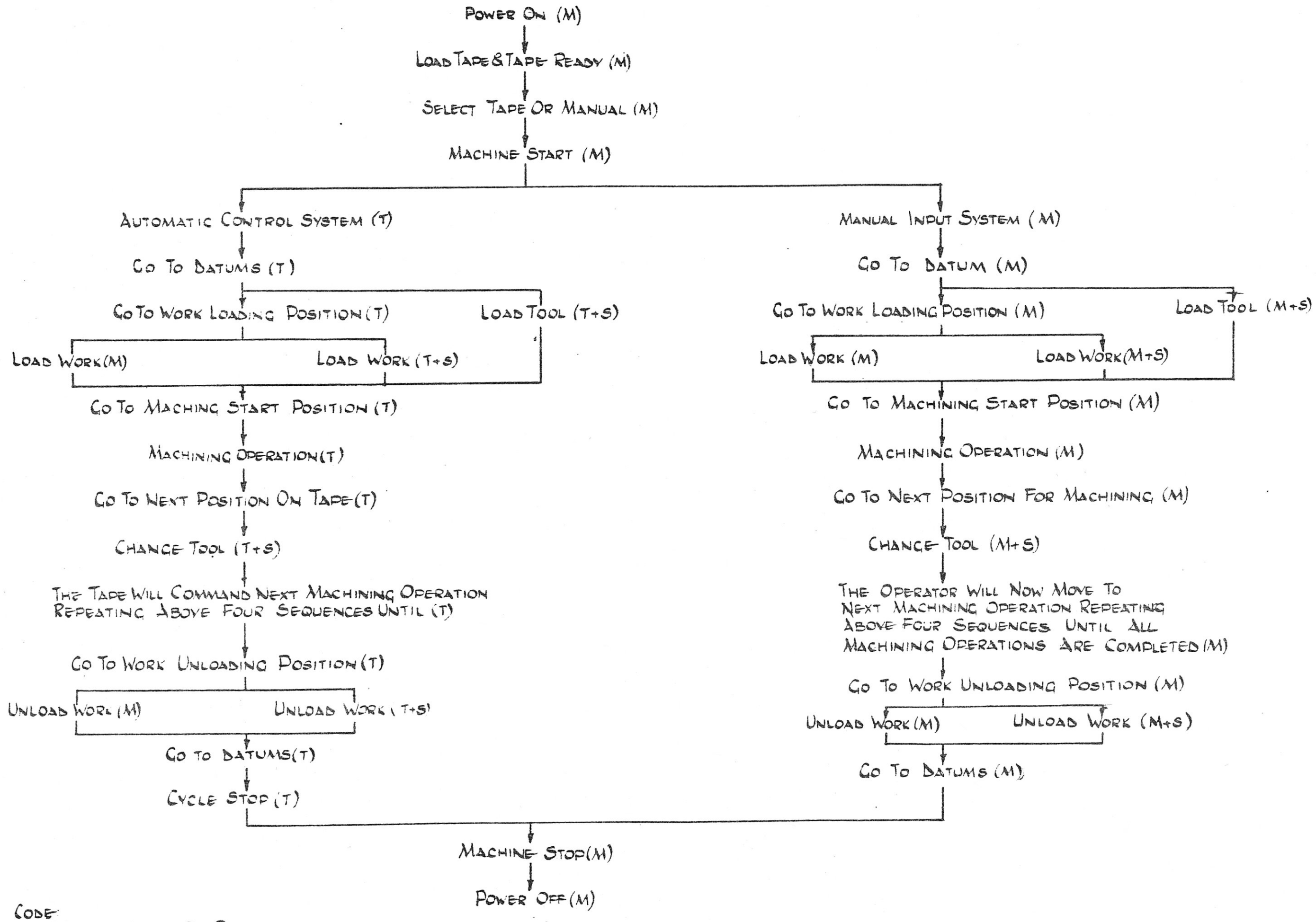
C.S.5 Cost of the Control System and Actuation

Detailed estimates are made in the different sections of the report, these giving a total as shown below.

Control Logic	£ 600
Control Actuation	£1300
	£1900

DRAWING No.

ISSUE | MODIFICATION.



Code
M = Manual Input Or Operation
T = Tape Input Or Tape Control
S = Sequencing By Built In Units

THIRD ANGLE PROJECTION

ITEM	PART No	DESCRIPTION	No OFF	MAT'L	SPEC	REMARKS

GENERAL TOLERANCE ON DIMENSIONS
MACHINED
UNMACHINED
OTHER DIMENSIONS AS STATED

JOB No. | No OF SETS REQ'D | SCALE | FINISH

DRAWN | CHK'D | APPV'D | STRESS APPV'D

TITLE:- Basic Machine Management Programme

ISSUED BY Control Logic Sub-Comm

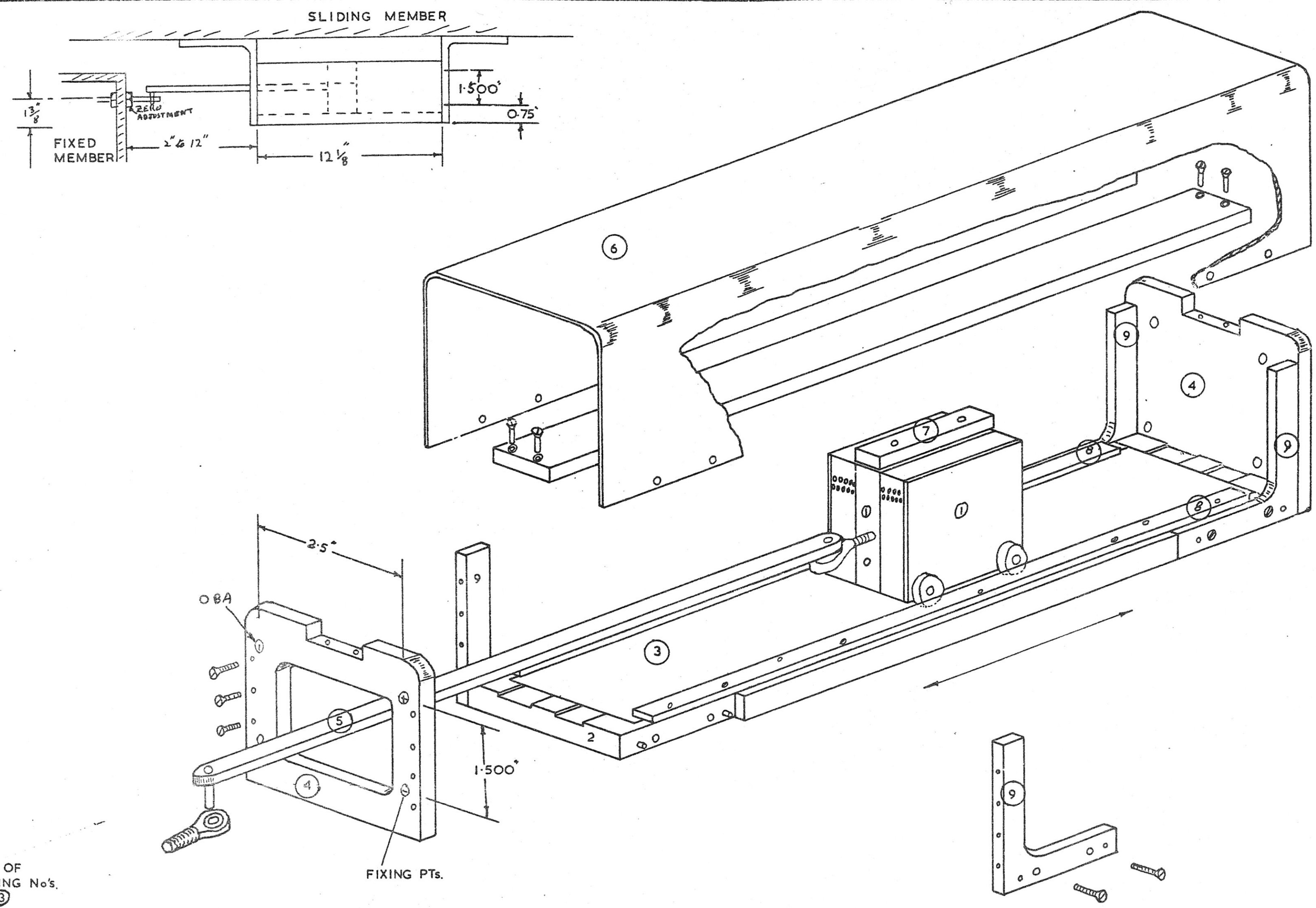
THE COLLEGE OF AERONAUTICS
CRANFIELD

DRAWING No. Fig CS4/9/1

SHT 1 OF 1 SHEETS

DRAWING No.

ISSUE	MODIFICATION.



RINGED No's.
ARE LAST DIGITS OF
RELEVANT DRAWING No's.
ie. PT/CS/C/10③
CODE STRIP

THIRD ANGLE PROJECTION

ITEM.	PART No.	DESCRIPTION	No. OFF	MAT'L.	SPEC.	REMARKS

GENERAL TOLERANCE ON DIMENSIONS
MACHINED
UNMACHINED
OTHER DIMENSIONS AS STATED
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No | No. OF SETS REQD | SCALE. | FINISH. | USED ON DRG.

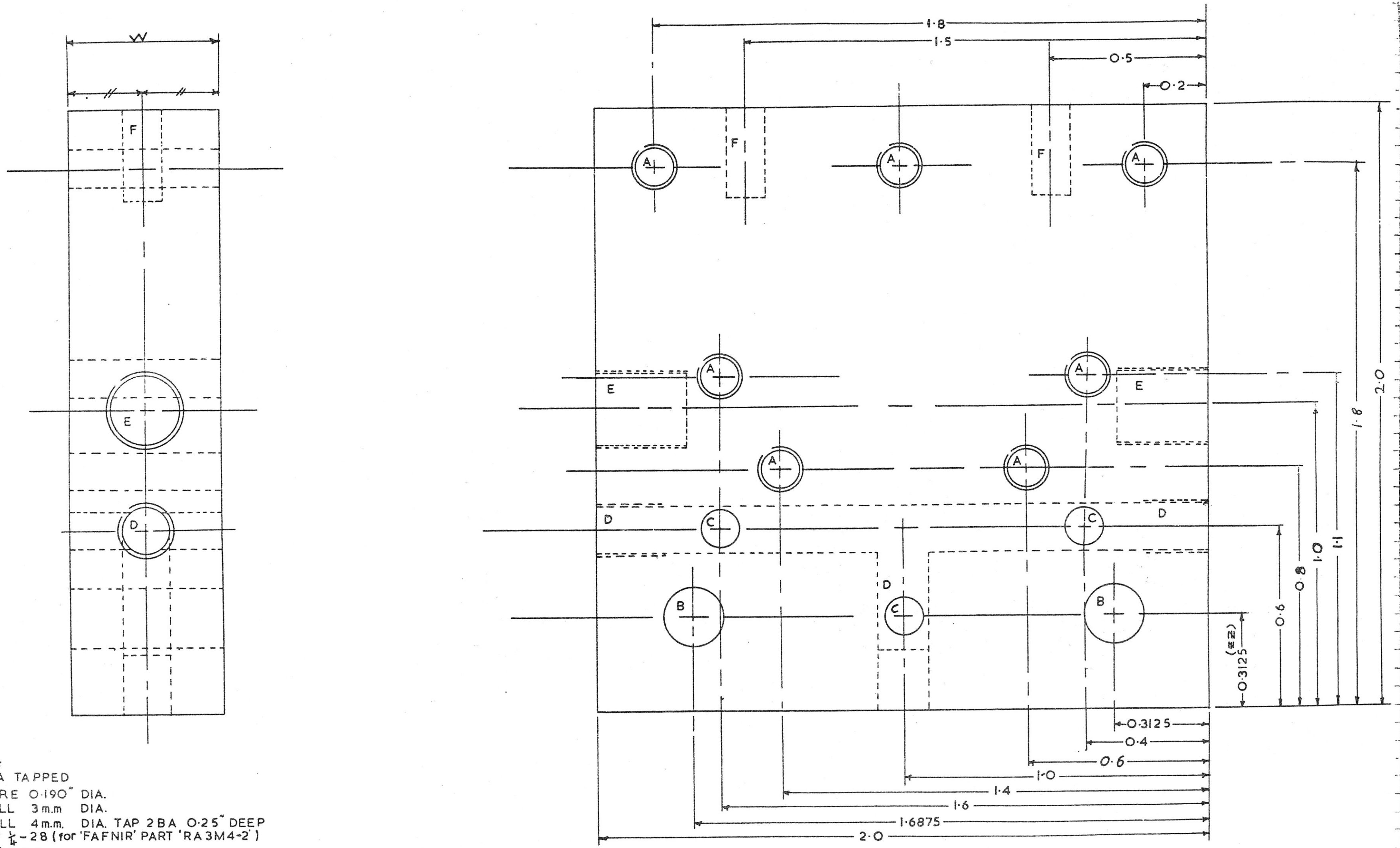
DRAWN: V.a.w. | CHKD | APPVD | STRESS APPVD

TITLE:- LINEAR ENCODER SCHEME 2
GENERAL ASSEMBLY

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD

DRAWING No. PT/CS/C/100

SHT. OF SHEETS.



DRAWING No.

ISSUE	MODIFICATION.

MANIFOLD. 1 OFF
'A' 4BA TAPPED
'B' BORE 0·190" DIA.
'C' DRILL 3 m.m DIA.
'D' DRILL 4 m.m. DIA. TAP 2BA 0·25" DEEP
'E' {TAP ¼-28 (for 'FAFNIR' PART 'RA3M4-2') 0·3" DEEP
'F' REAM 0·125" DIA. 0·3" DEEP
'W' GRIND 0·500"

END PLATES 2 OFF
'A' 3·75 m.m.
'B' 0·190"
'W' GRIND 0·050"

DIMENSION ZZ GRIND TO 0·312" AFTER ASSEMBLY
DIMENSIONS IN INCHES UNLESS STATED m.m.

THIRD ANGLE PROJECTION

GENERAL TOLERANCES ON DIMENSIONS
MACHINED ±0·001
UNMACHINED B, ZZ ±0·0001
OTHER DIMENSIONS AS STATED

JOB No. | No. OF SETS REQD | SCALE 5:1 | FINISH

ITEM | PART No. | DESCRIPTION | No. OFF | MATL STEEL | SPEC | REMARKS

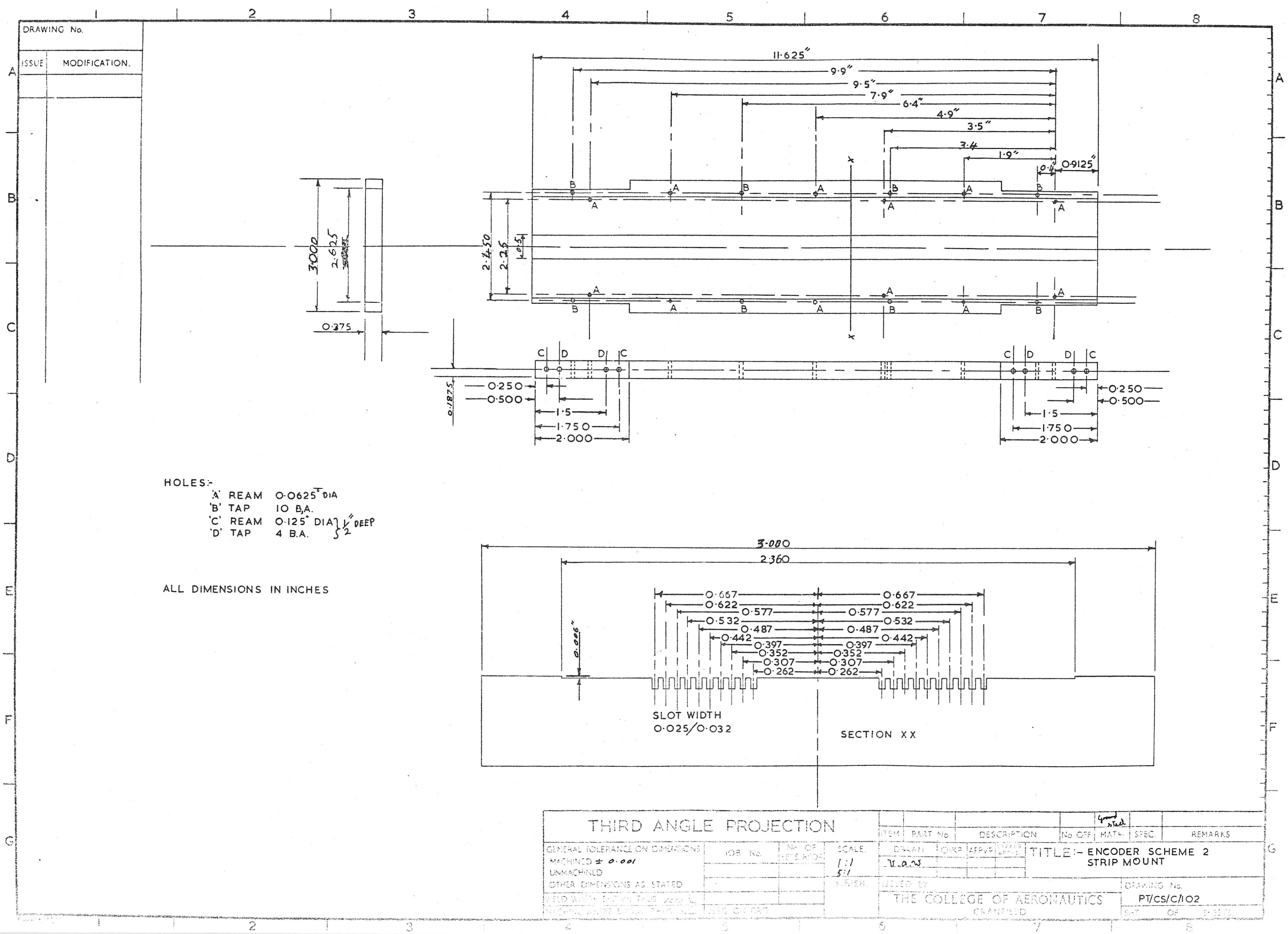
DRAWN D.A.W. | CHKD | APPVD

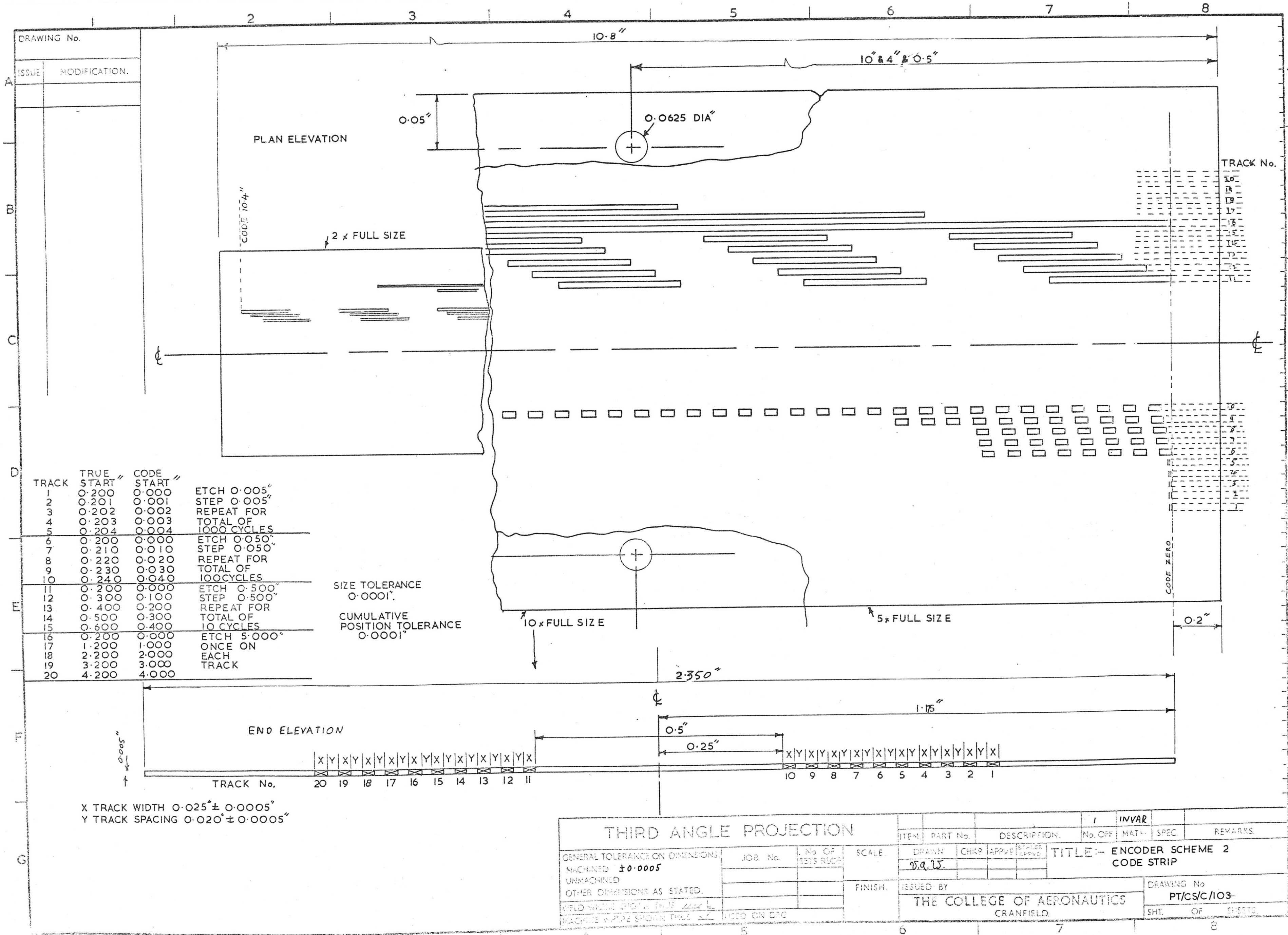
TITLE – ENCODER SCHEME 2
MANIFOLD & END PLATES

THE COLLEGE OF AERONAUTICS
CRANFIELD

DRAWING No. PT/CS/C/101

SHT OF SHEETS


DRAWING No.
ISSUE MODIFICATION.
11·625"
9·9"
9·5"
7·9"
6·4"
4·9"
3·5"
3·4
1·9"
0·9125"
3·000
2·625
2·450
2·25
0·375
0·1875
0·250
0·500
1·5
1·750
2·000
HOLES:-
'A' REAM 0·0625" DIA
'B' TAP 10 B.A.
'C' REAM 0·125" DIA } 1/2" DEEP
'D' TAP 4 B.A.
ALL DIMENSIONS IN INCHES
3·000
2·360
0·667
0·622
0·577
0·532
0·487
0·442
0·397
0·352
0·307
0·262
0·006"
SLOT WIDTH
0·025/0·032
SECTION XX
THIRD ANGLE PROJECTION
GENERAL TOLERANCE ON DIMENSIONS
MACHINED ± 0·001
UNMACHINED
OTHER DIMENSIONS AS STATED
JOB No.
SCALE 1:1 5:1
FINISH
ITEM PART No DESCRIPTION No OFF MAT'L Ground steel SPEC. REMARKS
DRAWN CHKD APPVD
TITLE:- ENCODER SCHEME 2 STRIP MOUNT
ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD
DRAWING No.
PT/CS/C/102



TRACK	TRUE START"	CODE START"	
1	0·200	0·000	ETCH 0·005"
2	0·201	0·001	STEP 0·005"
3	0·202	0·002	REPEAT FOR
4	0·203	0·003	TOTAL OF
5	0·204	0·004	1000 CYCLES
6	0·200	0·000	ETCH 0·050"
7	0·210	0·010	STEP 0·050"
8	0·220	0·020	REPEAT FOR
9	0·230	0·030	TOTAL OF
10	0·240	0·040	100CYCLES
11	0·200	0·000	ETCH 0·500"
12	0·300	0·100	STEP 0·500"
13	0·400	0·200	REPEAT FOR
14	0·500	0·300	TOTAL OF
15	0·600	0·400	10 CYCLES
16	0·200	0·000	ETCH 5·000"
17	1·200	1·000	ONCE ON
18	2·200	2·000	EACH
19	3·200	3·000	TRACK
20	4·200	4·000	

SIZE TOLERANCE 0·0001".

CUMULATIVE POSITION TOLERANCE 0·0001"

X TRACK WIDTH 0·025" ± 0·0005"
Y TRACK SPACING 0·020" ± 0·0005"

THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION.	No. OFF	MATL.	SPEC.	REMARKS.
			1	INVAR		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED ±0·0005
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD | SCALE. | FINISH. | USED ON DRG

DRAWN: D.Q.W. | CHKD | APPVD | STRESS APPVD

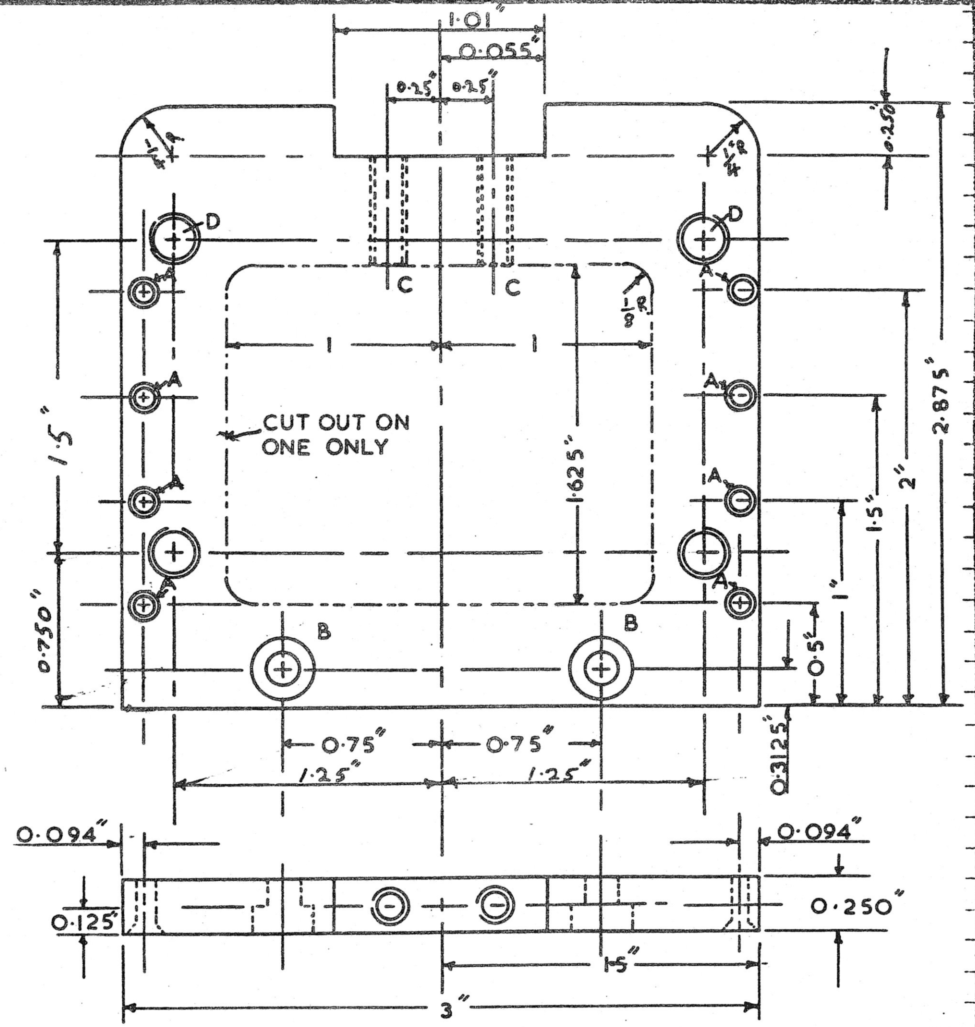
TITLE:- ENCODER SCHEME 2 CODE STRIP

ISSUED BY THE COLLEGE OF AERONAUTICS CRANFIELD.

DRAWING No. PT/CS/C/103

SHT. OF SHEETS.

DRAWING No.	
ISSUE	MODIFICATION.



'A' 2·25 m.m. COUNTER-SUNK
'B' 3·75 m.m. COUNTERBORE 0·250" DIA.
'C' TAP 4 BA
'D' TAP OBA

THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION.	No. OFF	MAT'L.	SPEC.	REMARKS.
			2 OFF	STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED ± 0·005
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No.
No. OF SETS REQD.
USED ON DRG.

SCALE. 2:1
FINISH.

DRAWN	CHKD.	APPVD.	STRESS APPVD.
V.A.W.			

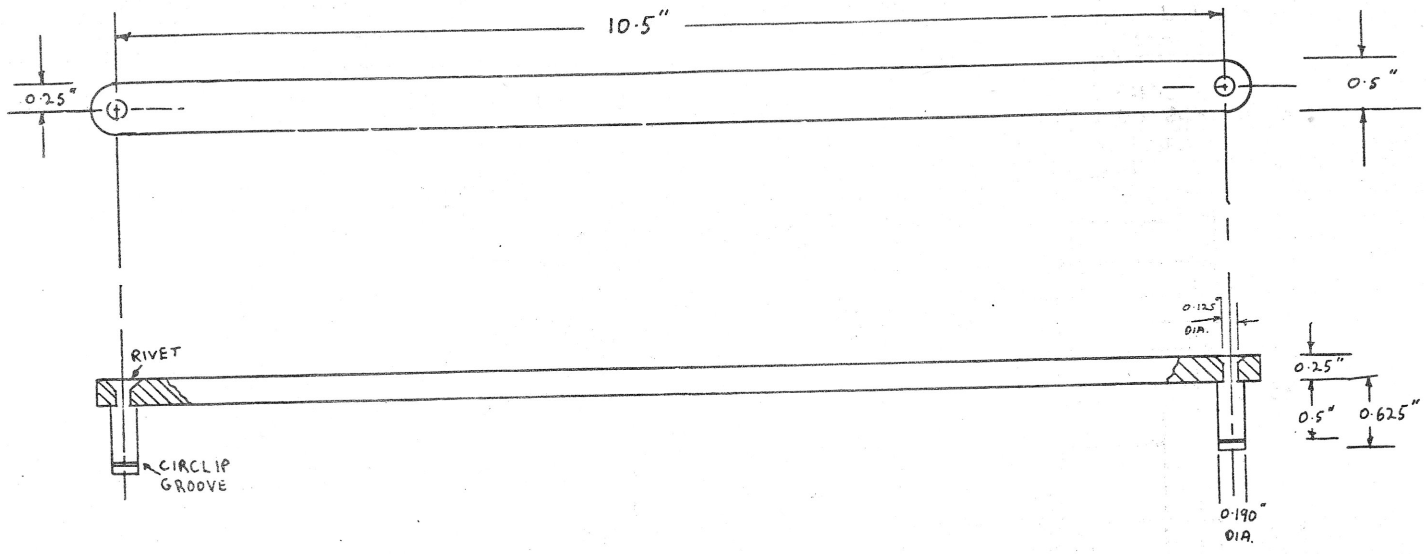
TITLE :- ENCODER SCHEME 2
CASE ENDS

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No.
PT/CS/D/104
SHT OF SHEETS

DRAWING No.

ISSUE | MODIFICATION.



THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION	No. OFF	MATL	SPEC.	REMARKS.
			1 SET	ROD:- INVAR PINS:- STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED GEN ±0·005"
UNMACHINED
OTHER DIMENSIONS AS STATED
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD | SCALE. | FINISH.
USED ON DRG.

DRAWN: V.A.W. | CHKD | APPVD | STRESS APPVD.

TITLE :- ENCODER SCHEME 2 CONNECTION ROD & PINS

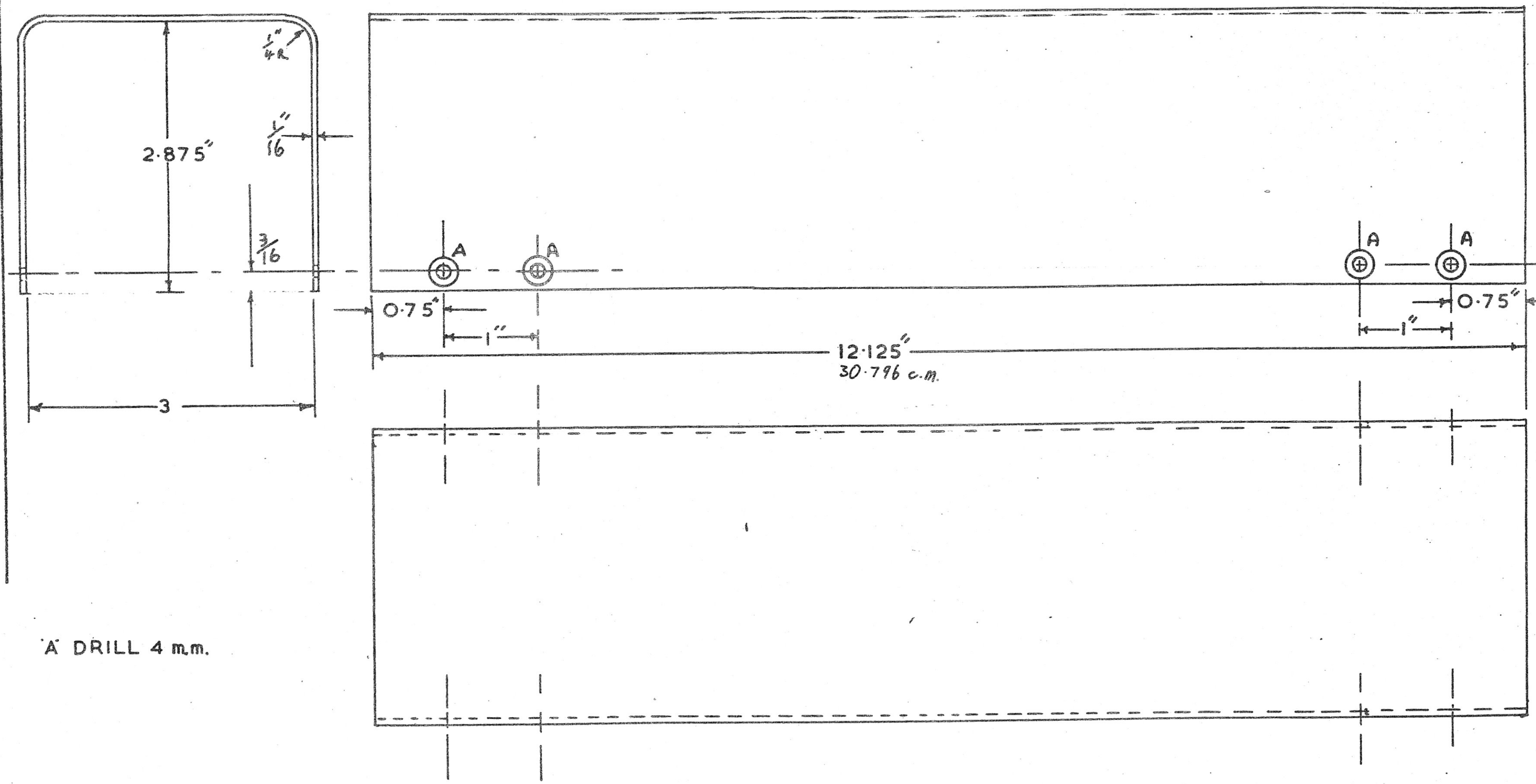
ISSUED BY THE COLLEGE OF AERONAUTICS CRANFIELD.

DRAWING No. PT/CS/D/105

SHT. OF SHEETS.

DRAWING No.

ISSUE	MODIFICATION.



'A' DRILL 4 m.m.

THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION	No. OFF	MATL.	SPEC.	REMARKS.
			1	STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED GEN. ± 0·010"
UNMACHINED HOLE ℄ ± 0·005"
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD. | SCALE. | FINISH.

USED ON DRG

DRAWN	CHKD.	APPVD.	STRESS APPVD.
V.A.W.			

TITLE :- ENCODER SCHEME 2
COVER

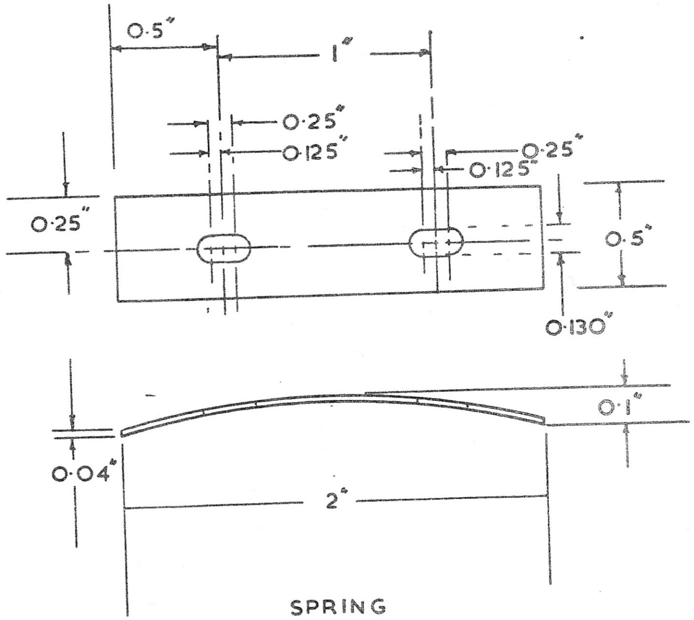
ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No.
PT/CS/D/106

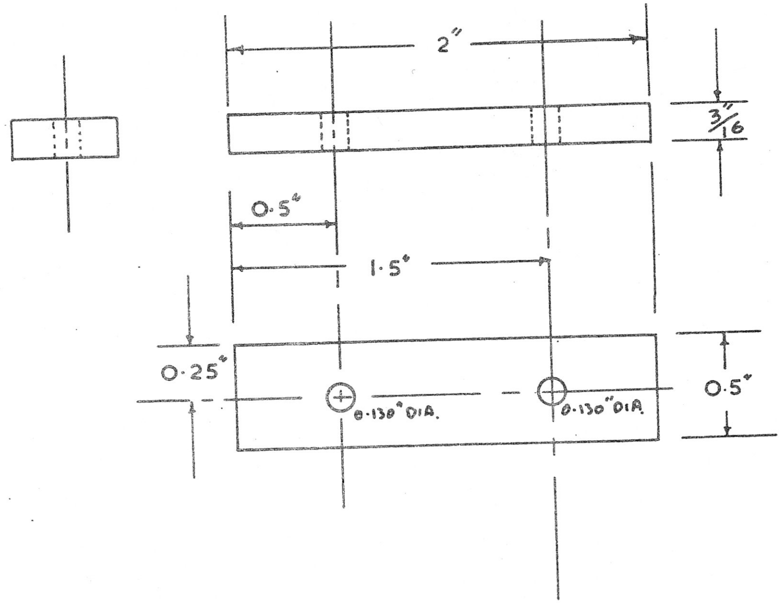
SHT OF SHEETS

R/66157 974 1A

DRAWING No.	
ISSUE	MODIFICATION.



SPRING
MAT:- SPRING STEEL



BLOCK
MAT:- NYLON

THIRD ANGLE PROJECTION

ITEM	PART No	DESCRIPTION	No. OFF	MAT'L	SPEC.	REMARKS.
			1 EACH			

GENERAL TOLERANCE ON DIMENSIONS
MACHINED ± 0.005"
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD. | SCALE. | FINISH.

USED ON DRG.

DRAWN | CHKD | APPVD | STRESS APPVD.

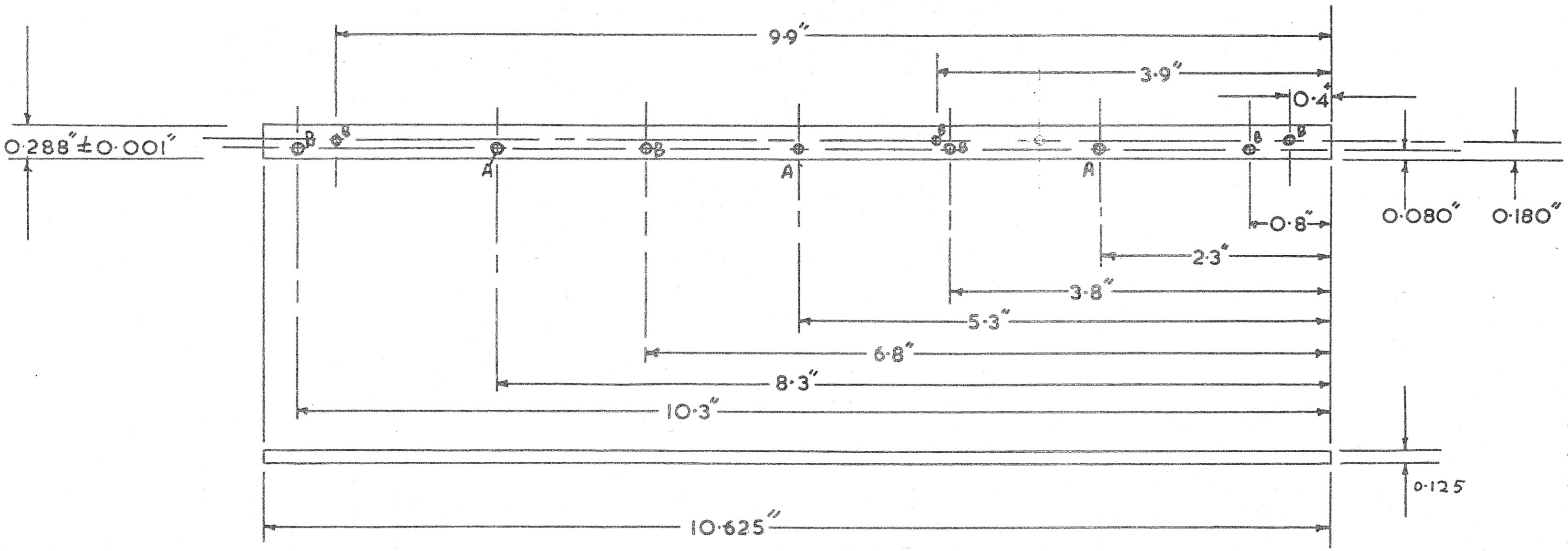
TITLE :- ENCODER SCHEME 2
BLOCK & SPRING

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No.
PT/CS/D/107

SHT. OF SHEETS.

DRAWING No.	
ISSUE	MODIFICATION.



'A' REAM 0·0625" DIA
'B' DRILL 1·8 m.m.

THIRD ANGLE PROJECTION

ITEM	PART No	DESCRIPTION	No. OFF	MATL.	SPEC.	REMARKS
			2	STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED ± 0·005
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD | SCALE. 1:1 | FINISH.

USED ON DRG

DRAWN	CHKD	APPVD	STRESS APPVD

TITLE :- ENCODER SCHEME 2 CLAMPS

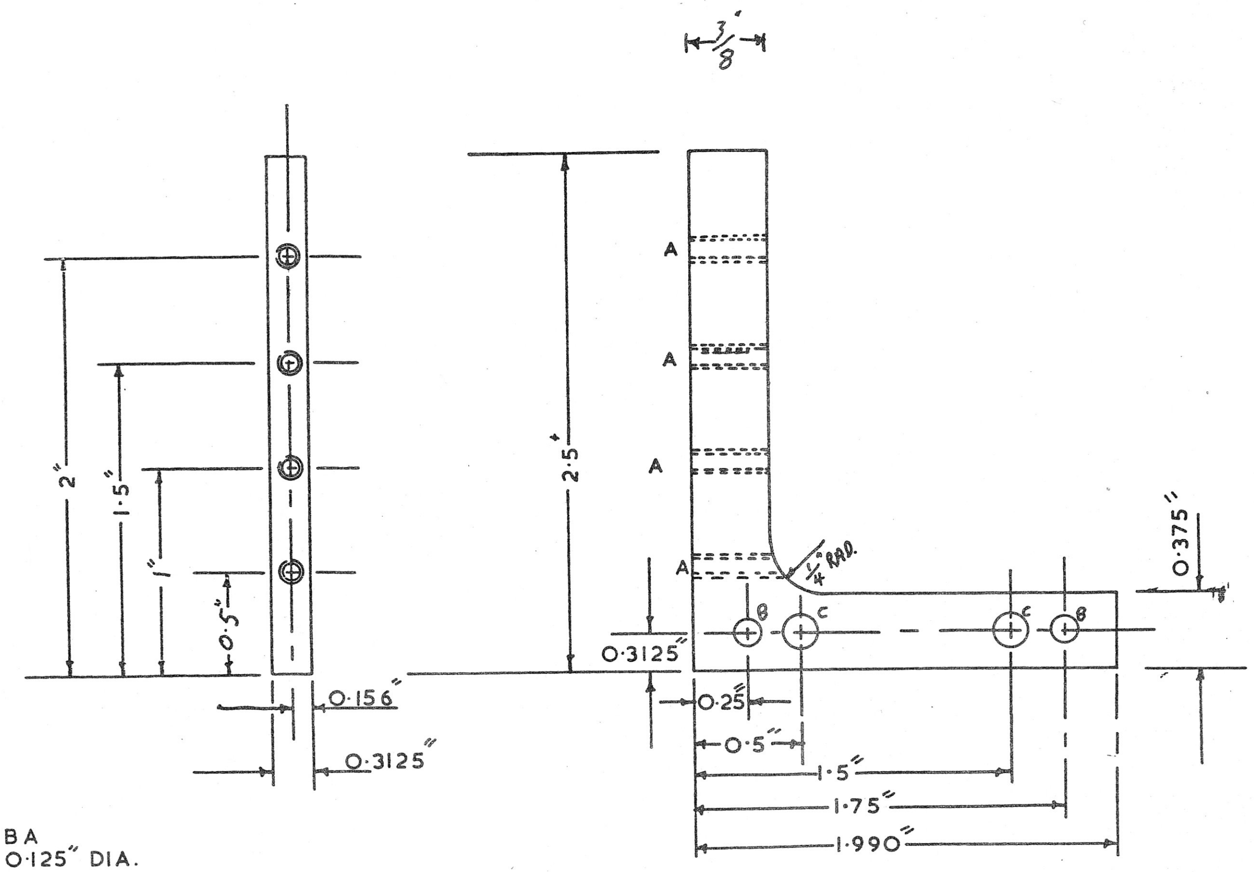
ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No. PT/CS/D/108

SHT OF SHEETS

DRAWING No.

ISSUE	MODIFICATION.



'A' TAP 8 BA
'B' REAM 0.125" DIA.
'C' 3.75 m.m. dia.

THIRD ANGLE PROJECTION

ITEM	PART No	DESCRIPTION.	No. OFF	MATL.	SPEC.	REMARKS.
			4	STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED 0.005"
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No.
No. OF SETS REQD.
USED ON DRG.

SCALE. 2:1
FINISH.

DRAWN	CHKD.	APPVD.	STRESS APPVD.
V.a.W.			

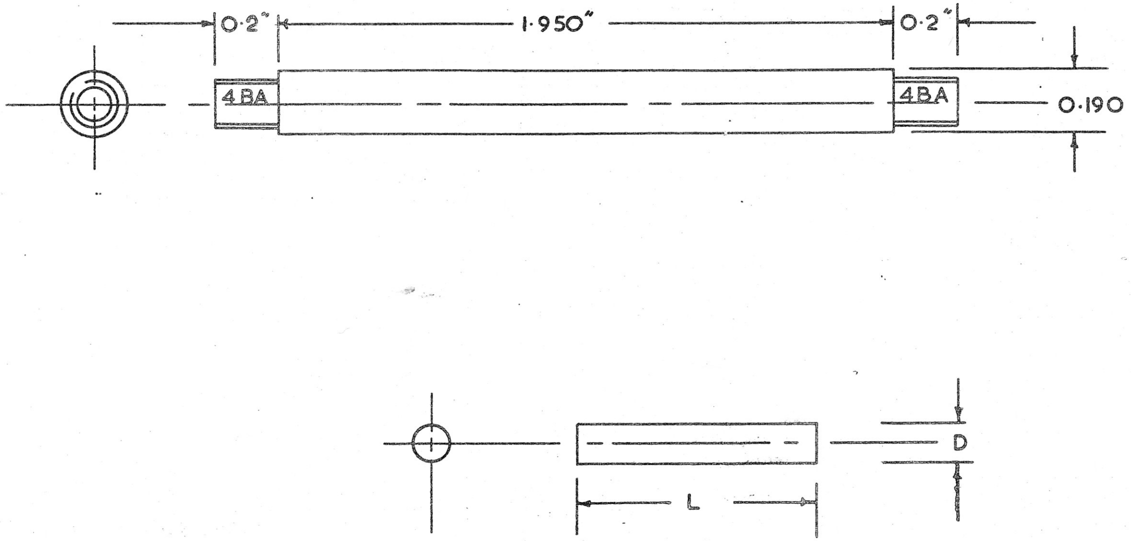
TITLE :- ENCODER SCHEME 2 BRACKETS

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No. PT/CS/D/109
SHT. OF SHEETS

DRAWING No.

ISSUE	MODIFICATION.



'D' 0·0625" 'L' 0·5" 8 OFF
'D' 0·125" 'L' 0·75" 8 OFF

THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION	No OFF	MATL.	SPEC.	REMARKS

GENERAL TOLERANCE ON DIMENSIONS
MACHINED GEN. ± 0·005"
UNMACHINED DIA. ± 0·0001"
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD | SCALE. | FINISH. | USED ON DRG

DRAWN | CHKD | APPVD | STRESS APPVD.

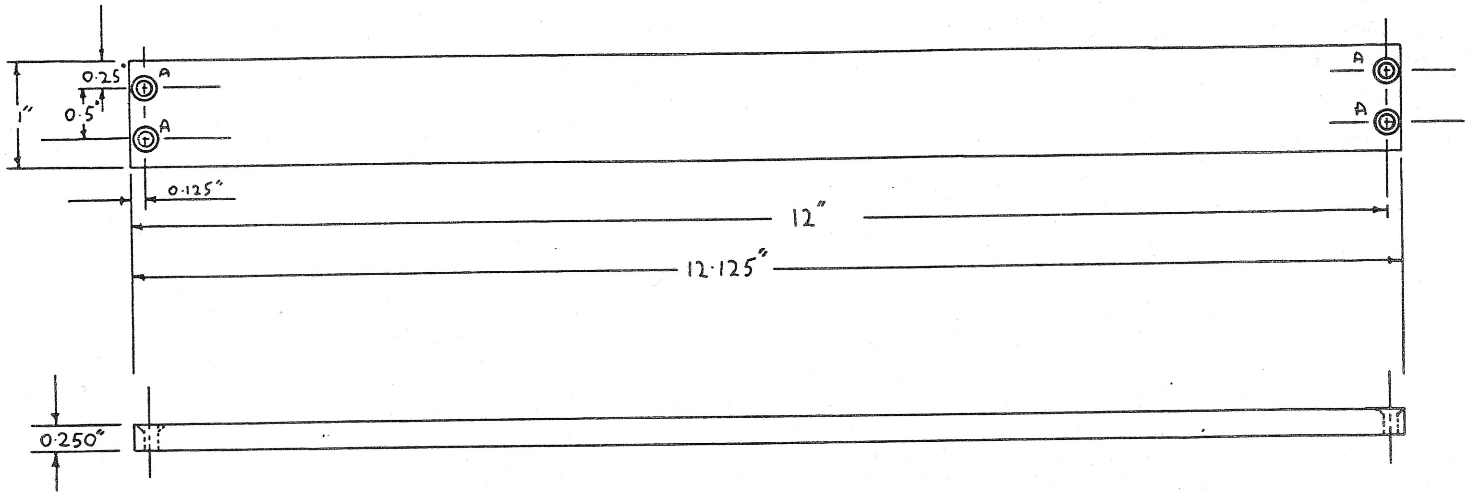
TITLE :- ENCODER SCHEME 2
DOWELS

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No.
PT/CS/D/110

SHT OF SHEETS

DRAWING No.	
ISSUE	MODIFICATION.



'A' DRILL 3·75 m.m. & COUNTERSINK

THIRD ANGLE PROJECTION

ITEM	PART No.	DESCRIPTION	No. OFF	MAT.	SPEC.	REMARKS.
			1	STEEL		

GENERAL TOLERANCE ON DIMENSIONS
MACHINED ± 0·005"
UNMACHINED
OTHER DIMENSIONS AS STATED.
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQ? | SCALE. | FINISH.

USED ON DRG.

DRAWN | CHK? | APPV? | STRESS APPVD.

TITLE :– ENCODER SCHEME 2
BAR

ISSUED BY
THE COLLEGE OF AERONAUTICS
CRANFIELD.

DRAWING No.
PT/CS/D/111

SHT OF SHEETS

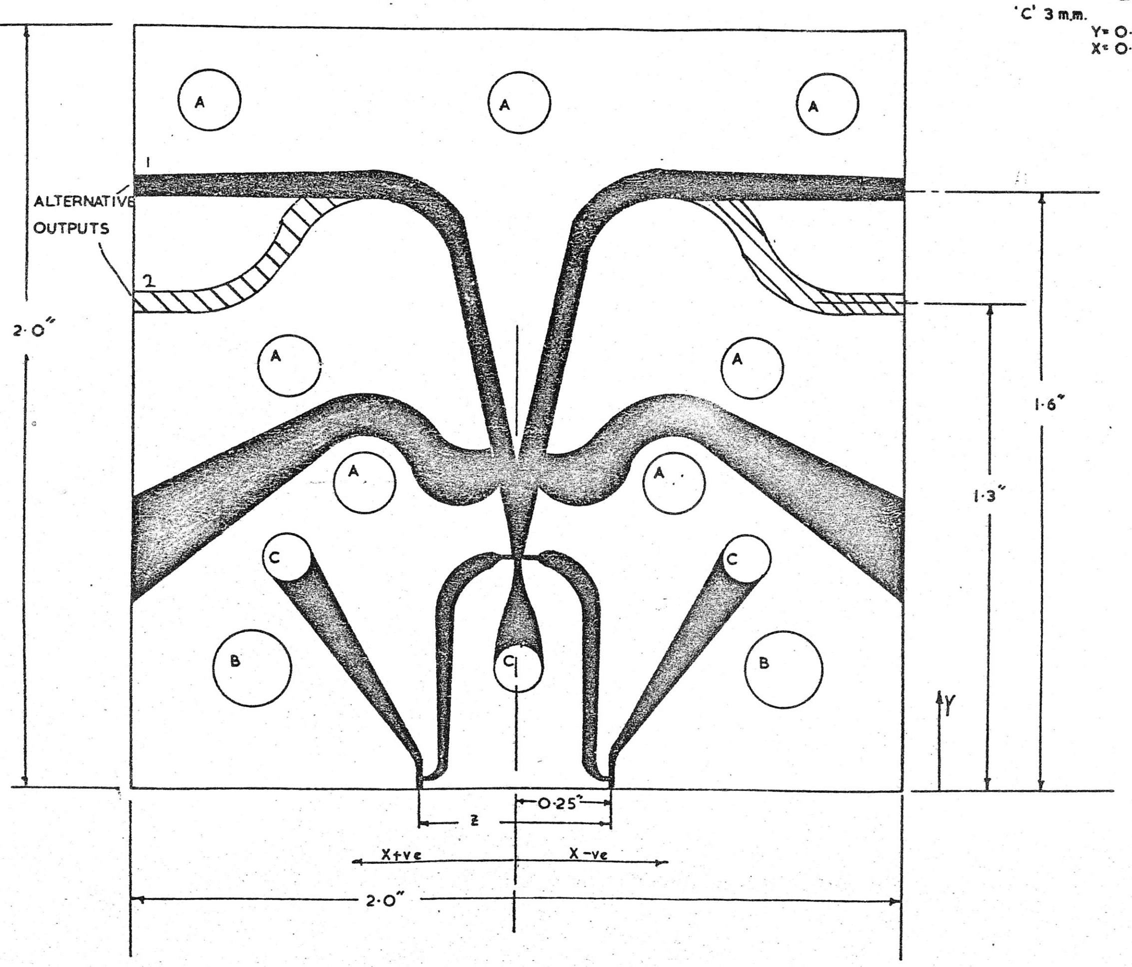
DRAWING No.	
ISSUE	MODIFICATION.

HOLES:-

'A' 3·8 m.m.
Y = 0·8" 1·1" 1·8"
X = ±0·4" ±0·6" ±0·8" & 0"

'B' BORE 0·190"
Y = 0·3125"
X = ±0·6875"

'C' 3 m.m.
Y = 0·3125" 0·6"
X = 0·000" ±0·6"



THIRD ANGLE PROJECTION.

GENERAL TOLERANCE ON DIMENSIONS
MACHINED
UNMACHINED
OTHER DIMENSIONS AS STATED
WELD WHERE SHOWN THUS
MACHINE WHERE SHOWN THUS

JOB No. | No. OF SETS REQD | USED ON DRG.

SCALE. 5:1
FINISH.

ITEM.	PART No.	DESCRIPTION.	No. OFF	MATL.	SPEC.	REMARKS.

DRAWN. | CHKD | APPVD | STRESS APPVD

TITLE:- ENCODER SCHEME 2 ELEMENT LAYOUT

ISSUED BY
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CRANFIELD

DRAWING No. PT/CS/C/114
SHT. OF SHEETS.