AN INVESTIGATION INTO MINIATURE

HYDRAULIC ACTUATION TECHNIQUES

FOR NEEDLE CONTROL ON INDUSTRIAL

KNITTING AND SEWING MACHINES

AN INVESTIGATION INTO MINIATURE HYDRAULIC ACTUATION TECHNIQUES FOR NEEDLE CONTROL ON INDUSTRIAL KNITTING AND SEWING MACHINES.

by

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S Y N O P S I S

This research has been sponsored by Courtauld's Educational Trust Fund to investigate miniature hydraulic actuation techniques, and demonstrates how linear mechanical motions can be replaced by hydraulic devices.

Basic actuators have been outlined in the author's thesis "The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine" submitted for M.TECH. Degree 1970 and as a continuation of this study, the concepts have been applied to textile machinery.

The thesis presented is in four main parts.

Part 1. The design and development of a hydraulic circular weft knitting machine.

Part 2. The construction and testing of a hydraulic lockstitch sewing machine.

Part 3. A detailed design study and analysis of pulse-generating rotary valves.

Part 4. The design of a multi-feeder hydraulic circular weft knitting machine.

Part 1 deals with the knitting machine aspect of the project consisting of verifying that a multi-actuator rotary valve system would operate with the desired time displacement profile, and in the correct sequence. This was then used as the basis for developing a ninety-six needle, single feeder hydraulic circular weft knitting machine. This prototype machine was tested to obtain an assessment as to the advantages offered by hydraulic knitting techniques.

Part 2 involved replacing the needle and thread take-up mechanisms of a lockstitch sewing machine, by two miniature hydraulic actuators, controlled by a rotary valve. The purpose of this machine was to prove that stitches could be formed successfully, thus demonstrating any beneficial features offered by hydraulic sewing devices.

Part 3 deals with the detailed design study for pulse-generating rotary valves resulting from the previous applications. This valve was a new concept in valve technology and having established its' definite potential, warranted the formation of a design procedure. The study outlines a method of optimising the torque required to rotate the bobbin by the construction of a mathematical model.

Part 4 was concerned with designing a multi-feeder hydraulic circular weft knitting machine. This machine, controlled by an integrated actuator rotary collar valve to generate pulses, demonstrated how a series of twelve knitting time-displacement profiles could be created by ninety-six actuators positioned in a circular configuration.

Thus, the research programme has been aimed at demonstrating how high speed motions, normally obtained by mechanical devices (cams, linkages) can be produced by miniature hydraulic actuation techniques. The feasibility of using these techniques has been verified by the building and testing of probably the first ever hydraulic knitting and sewing machines.

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NOTATION

		Units
θ	Angular Displacement	degrees
t	Time	sec
P	Supply Pressure	lbf/in^2
E	Exhaust Pressure	lbf/in^2
x	Linear Displacement	in.
y	Linear Displacement at right angles to the direction of x	in.
h	Lubrication Film Thickness	in.
p	Pressure Variable	lbf/in^2
U	Linear Velocity	in/sec
η	Viscosity (reyns)	$lbf.sec/in^2$
ω	Angular Velocity	rad/sec
c	Radial Clearance	in.
ε	Eccentricity Ratio	
e	Eccentricity	in.
R	Radius	in.
ϕ	Eccentricity Angle	degrees
L	Length	in.
B	Breadth	in.
$\bar{p}$	Dimensionless Value of Pressure	p
$\bar{y}$	Dimensionless Value of y	
$\bar{x}$	Dimensionless Value of x	
$\bar{h}$	Dimensionless Value of h	
W	Load	lbm.
α	Mesh Size in x direction	
β	Mesh Size in y direction	
A	Relaxation Factor	

		Units.
F	Total Force	lbf
τ	Shear Stress	lbf/in^2
σ_C	Hoop Stress	lbf/in^2
M	Modulus of Elasticity	lbf/in
μ	Poisson's Ratio	
D	Diameter	in.
σ_R	Radial Stress	lbf/in^2

1. INTRODUCTION

The research presented in this thesis has been sponsored by Courtauld's Educational Trust Fund and throughout the period of study, investigations have been directed towards the creation of novel hydraulic control systems for application to textile machinery. The work presented is in three main sections:-

(i) The design and development of a hydraulic circular weft knitting machine.

(ii) The construction and testing of a hydraulic lockstitch sewing machine.

(iii) A detailed design study and analysis of pulse-generating rotary valves, which have been used as the control mechanism in both the previous developments.

The first section of the project deals with a hydraulic knitting machine, and involves the application of basic high speed actuation techniques outlined in the author's thesis: "The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine". (18). The actuators and rotary control mechanism previously designed and tested as a single unit,had demonstrated that a time displacement profile suitable for the formation of a plain knitted stitch could be obtained by using hydraulic devices. The aim of this project was to use technical knowledge gained in previous research to create a hydraulic knitting machine.

Initially, a block of actuators simulating a knitting station had to be designed and commissioned. This study would produce information to answer the following questions:-

(iv) Would actuators operate in sequence?

(v) Would a single pulse-generating rotary valve be capable of operating at the desired speed?

(vi) Could actuators be packaged at 16 per inch?

(vii) Could the three basic knitting actions (knit, tuck and miss) be programmed into the system?

(viii) Would the actuators operate in the three different modes?

The parameters for this rig were a block of twenty-four actuators in a 16 gauge orientation, sequenced from a single rotary valve. The programming features were restricted to four individual selections per needle, before repeating the cycle. The programming aspect was limited to enable the full speed potential to be realised. This rig was fully tested in the laboratory and the needles found to function at near maximum cycling rates of 25 cycles per second. All the needles operated in sequence and could be programmed for the knit, tuck and miss modes. Having demonstrated how a series of miniature hydraulic actuators could be controlled to operate in sequence at speeds in excess of existing mechanical cam systems, the outstanding consideration still to be determined was its actual knitting capability.

It was at this juncture, using the technical experience gained in developing the multi-actuator rig, that the specification for a prototype circular weft knitting machine was compiled, taking into consideration existing hardware. The specification for the machine was:-

(ix) A machine having a single set of needles traversing continually through the knitting profile,

(x) A single feeder machine,

(xi) Ninety-six needles spaced at four to the inch in a circular configuration.

This prototype machine would provide the basis for future developments and enable the performance of a hydraulic single jersey circular weft knitting machine to be evaluated. A design study for a prototype machine was completed, being based on the existing rotary valve with a new actuator block and trix mechanism. The time displacement profile generated by this configuration, was to have six consecutive needles in the extended knitting position, followed by six needles at tuck height, with the remaining eighty-four needles in the miss (static) condition. When the hydraulic control system for the machine had been assembled, the system was tested under similar conditions to that of the multi-actuator block. This provided a visual demonstration of how the time displacement profile was formed, and its traversal round the series of actuators produced a cam-like profile. After finalising the construction of the machine and introducing minor modifications to the yarn carrier, results were obtained showing that knitting speeds previously unobtainable by mechanical cam systems were possible using hydraulic actuation techniques. After building and testing this machine further major advantages were discovered. The number of possible knitting stations per machine could be increased:- the independent movement of each needle removed the limitation imposed by the length of cam track. This feature would allow the number of needles per knitting station to be greatly reduced so enabling greater fabric production by utilising more feeders. The quality of knitting produced by the hydraulic machine at high speed was relatively good, due to the single needle knitting action characteristics made possible by hydraulic devices. Consequently, this section of the project has indicated a new technique of needle motivation for knitting machines, and while a commercial machine has yet to be developed, this new approach offers certain features that may have great potential in the textile industry.

A second development for the application of miniature hydraulic actuation techniques in textile engineering was for use on a sewing machine. Sewing machines contain a number of complex mechanisms that culminate in a linear motion. To demonstrate the feasibility of applying miniature hydraulic actuators to a sewing machine, a mechanical lockstitch machine was converted by introducing miniature hydraulic actuators to function as the needle and thread take-up mechanisms. The control for the actuators was accomplished by introducing a rotary valve at the end of the hook driving mechanism, thus establishing a phase relationship between the hook and the hydraulic motions. The designing and testing of a hydraulic sewing machine has demonstrated that a lockstitch can be formed using this technique.

In specific applications, the benefits of this technique may be as follows:-

(xii) Mechanically decoupling the hook and needle motions.

(xiii) Removing the restrictions of the throat.

(xiv) Providing a simple method of reversing the direction of individual components.

(xv) Removing restrictions imposed by mechanical mechanisms. (i.e. excessive needle movement due to the slider crank mechanism).

Two such applications could be manifest in a left-hand overlocking machine, also a moveable co-ordinate sewing head. The work undertaken and presented in this thesis serves as an introduction to hydraulic sewing techniques and provides a basis for future developments.

The necessity for the third section of this study had arisen from previous developments. These two applications of pulse-generating rotary valves established that they functioned effectively and possessed characteristics not found when using conventional hydraulic control devices. Therefore, a further study to rationalise the overall design technique for this type of valve was essential. Up to this juncture valves had been designed solely to generate the required pulses, so powering the systems. The valves had now to be analysed to optimise the overall performance in any future applications. During the testing of the two systems, it became apparent that unbalanced forces were created within the valve due to irregular pressure distribution between the rotor and the cylinder. A method of analysing the pressure distribution based upon hydro-dynamic bearing theory was evolved, and the results used to design compensating pads, so balancing the internal hydrostatic forces. A test rotary valve, designed using results obtained from this analysis, showed a marked improvement in performance. This also enabled a prediction as to the torque required to rotate the valve under particular operating conditions to be made. This analytical technique was established in the form of computer programmes, enabling the operating characteristics to be determined once the physical dimensions of the valve had been selected.

The research presented in this thesis has provided a novel and viable alternative approach to the well-established mechanical mechanisms of knitting and sewing machines. The application of miniature hydraulic actuators and pulse-generating rotary valves in these two instances will, it is hoped, stimulate interest in miniature hydraulic devices and provide encouragement for designers in attempting to obtain cyclic linear motions by using these techniques.

PART I

THE DESIGN AND DEVELOPMENT OF A HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

2. DESIGN AND DEVELOPMENT OF A MULTI-ACTUATOR BLOCK.

3. A HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

2. DESIGN AND DEVELOPMENT OF A MULTI-ACTUATOR BLOCK

2.1. INTRODUCTION

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PART 1

THE DESIGN AND DEVELOPMENT OF A HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

2. DESIGN AND DEVELOPMENT OF A MULTI-ACTUATOR BLOCK

2.1. Introduction

2.1.1. Previous Work.

In the thesis "The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine" by the author, details are given on how a miniature compound hydraulic actuator was developed to produce the desired time displacement profile required for producing a knitted loop. In conjunction with the actuating device a study was also made of the control aspect of the actuator, both in relation to the cyclic displacement and the sequencing of the actuators to each other. This resulted in the development of a rotary supply valve; the rotary valve is essentially a bobbin with slots machined into the surface such that when the bobbin is encased in a cylinder and supplied with pressurised oil, a train of pulses are generated by revolving the bobbin.

At the end of the thesis a design for a Multi-Actuator Block is briefly outlined, together with the drawings required for a twenty-four actuator system. This multi-actuator block together with its development, provides the starting point for the continuation of the research, which is culminating in the building and testing of probably the first ever hydraulic knitting machine.

2.1.2. Technical Decisions Relating to the Development of a Multi-Actuator Block.

A decision to build a multi-actuator block was taken at a technical meeting in Coventry between Mr. K. Hartley, Head of Research and Development at Courtaulds, Mr. B. Baker, a Courtaulds Engineer and Mr. T.P. Priestley and the author. It was agreed at this meeting that the application of hydraulic actuators to knitting machines should be directed towards increasing the production speed of conventional knitted fabrics with only limited pattern capabilities. To this end, a block of twenty four actuators would be built with sixteen actuators to the inch. At this juncture it was envisaged that the Department of Mechanical Engineering would be responsible for building a block of twenty four actuators which would operate in sequence and have a limited patterning facility. The knitting aspect of the project would be undertaken by Courtaulds' own knitting engineers once the actuators had been fully tested.

2.2. The Time Displacement Profile

The first consideration for any work on a knitting machine must be the time displacement profile that is required by the needle to form the knitted stitch. A knitting machine has three basic stitch forms, see figure 1 which shows the basic profiles required.

(i) The knitting movement; here the needle rises to the maximum position (AB) passing the loop already on the needle over the open latch. The needle then

A DIAGRAM TO SHOW THE TIME DISPLACEMENT PROFILE REQUIRED FROM A MINIATURE HYDRAULIC ACTUATOR TO PRODUCE A KNITTING ACTION.

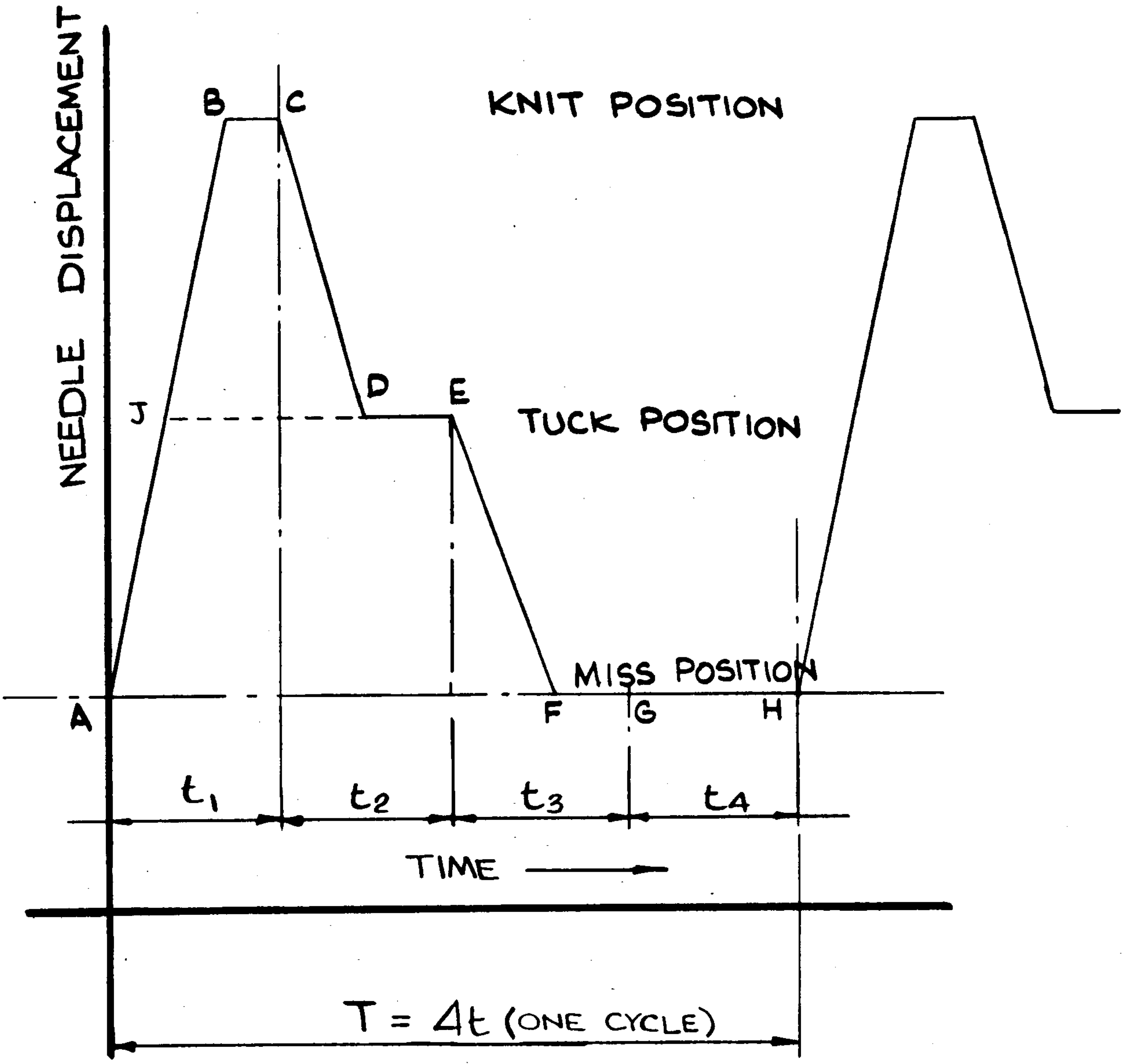


FIG 1

returns to the tuck position (CD) and dwells for a period (DE). At this stage the new yarn is introduced under the hook of the needle before the needle returns to the miss position (EF) pulling the new yarn through the old loop to form the knitted stitch.

(ii) The tuck movement; where the needle rises to approximately half the maximum stroke (AJ) and retains the old loop in the hook of the needle. A new yarn is introduced and is held together with the old loop in the hook of the needle. The tuck action can only be performed for four consecutive movements without fear of damaging the needle, after which the basic knitting action must be performed to clear the hook. The tucking action permits texture to be knitted into a fabric.

(iii) The miss movement is no movement of the needle and in effect produces a ladder, its main application being the knitting of multi-colours and the colours not required for the pattern are wrapped into the back of the fabric.

These three basic movements enable the vast majority of fabrics in current production to be knitted, consequently any device for driving a knitting needle must be capable of these three actions. However, the basic knitting action can be used on its own to produce "plain knitting" which accounts for over 50% of all knitted fabric. The demand for plain knitting is ever-increasing as:- the basic backing for bonded fabrics, and the disposable fabrics that are becoming more popular.

2.3. The Miniature Hydraulic Actuator

The basic hydraulic actuator developed for use in a circular weft knitting machine can be seen in figure 2. This actuator is a compound device with a knitting piston (5) and a tuck probe (6). The knitting piston is able to slide axially in the top bore through a displacement in the order of one inch and when the piston is in either extreme position then the hydraulic stop port 7 is open to tank, hence relieving the supply pressure. This central hydraulic stop prevents hydraulic pressure from driving the piston hard against the mechanical boundaries of the actuator, and thus prevents self destruction of the knitting piston. Similarly the tuck probe (6) can slide axially along its bore through a displacement equal to half that of the knitting piston. The hydraulic stop 8 is used to reduce the impact forces on the tuck probe. These two bores are linked by a smaller bore which allows the rod on the tuck probe to protrude into the knitting piston bore and make mechanical contact with the bottom of the knitting piston (5). The distance between the bores is such that when the tuck probe (6) is fully protruding into the knitting piston bore and in mechanical contact with the knitting piston, then the top edge of the piston is central to port 4, (the tuck position). The only other precaution necessary is to make the axial displacement of the

SINGLE PROBE MINIATURE HYDRAULIC ACTUATOR.

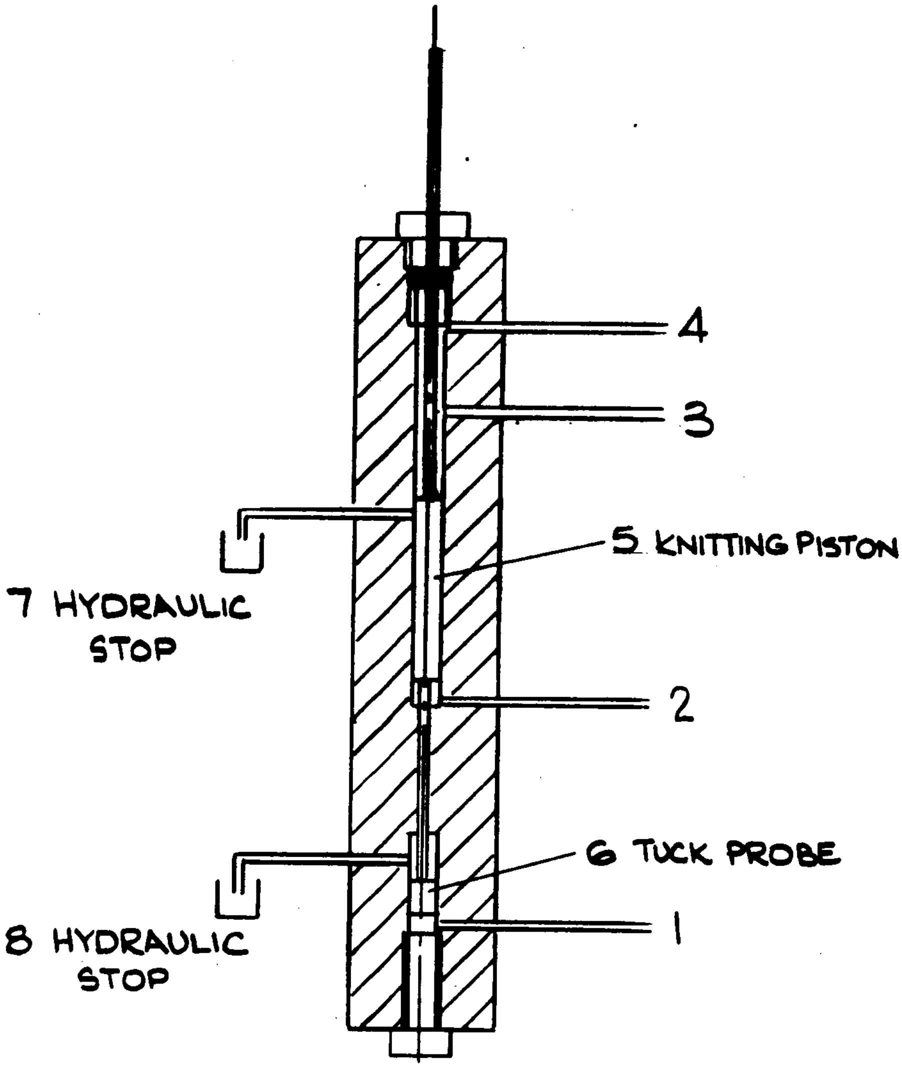


FIG 2

tuck probe of sufficient length to allow the probe rod to retract into the small bore linking the two cylinders. This prevents the tuck probe from taking the impact at the end of the knitting piston movement.

To drive the miniature hydraulic actuator through the knitting time displacement profile outlined in figure 1, and considering the hydraulic actuator shown in figure 2, the following sequence of pressure and exhaust pulses must be supplied to the actuator through ports 1,2,3 and 4. If the total cycle time of the movement is defined as $T = 4t$ and the four basic movements,

(i) Travel from the miss to the knit position AB in time t_1

(ii) Travel from the knit to the tuck position CD in time t_2

(iii) Travel from the tuck to the miss position EF in time t_3

(iv) Dwell in the miss position GH for the time t_4

are considered to take place in a period of equal duration namely time t. Then considering figures 1 and 2:

(v) In period t_1:- the knitting piston 5 and tuck probe 6 are driven out to a maximum displacement by applying pressure pulses at ports 1 and 2 and exhausting ports 3 and 4.

(vi) Period t_2:- the tuck probe 6 is kept in the extended state by maintaining the pressure at port 1 while the knitting position 5 is retracted, by applying a pressure at port 4, until it makes mechanical contact with the tuck probe 6. At this instant the knitting piston is in the tuck position and port 3 is open to exhaust hence acting as a hydraulic stop.

(vii) Period t_3:- the knitting piston 5 and the tuck probe 6 are retracted back to the miss position by applying a pressure at ports 3 and 4.

(viii) Period t_4:- the knitting piston 5 and the tuck probe 6 are held in the retracted miss position.

Thus summarising the sequence of pulses required for any period t to enable the hydraulic actuator to be driven through the knitting time displacement profile.

TABLE 1

SEQUENCE OF HYDRAULIC PULSES

TIME	PORT 1	PORT 2	PORT 3	PORT 4
t_1	Pressure	Pressure	Exhaust	Exhaust
t_2	Pressure	Exhaust	Exhaust	Pressure
t_3	Exhaust	Exhaust	Pressure	Pressure
t_4	Exhaust	Exhaust	Pressure	Pressure

To drive the miniature hydraulic actuator through the tuck profile, the knitting piston 5, in period t_1, has to be moved through a displacement AJ. This is achieved by preventing the pressure pulse at port 2 activating the knitting piston 5, allowing the tuck probe 6 to push the knitting piston mechanically into the tuck position. This can be effected by placing an on-off valve in line 2. The rest of the cycle is identical to the knitting mode.

The miss movement can similarly be performed by preventing the pressure pulses in period t_1 from reaching ports 1 and 2 again by the use of an on-off switching system in the supply line.

2.4. A Basic Rotary Valve

To obtain the sequence of pressure and exhaust pulses outlined in section 2.3. a new type of valve was developed, namely a rotary pulse-generating valve. The rotary valve consists primarily of a bobbin with radially machined slots. These slots are linked to either an exhaust sink or a pressure source. The bobbin is placed into a cylinder with ports machined to coincide with the slots in the bobbin, hence as the bobbin is rotated a series of pressure and exhaust pulses are generated at the cylinder ports. Thus for a rotary valve to drive a single actuator through the time displacement profile outlined in figure 1, the pressure and exhaust slots have to be constructed to meet the requirements specified in TABLE 1. Figure 3 shows the cross section of the slots together with a sectional view of the bobbin. These views of the bobbin indicate a method of how the slots can be linked to a pressure source, or to exhaust. A central hole is machined in the bobbin acting as a common sink. To produce an exhaust slot in the

A DIAGRAM TO SHOW THE HYDRAULIC CIRCUIT FOR A SINGLE ACTUATOR AND A ROTARY VALVE, TOGETHER WITH A SECTIONAL VIEW OF THE SLOTS REQUIRED IN THE BOBBIN TO PRODUCE THE TIME DISPLACEMENT PROFILE REQUIRED FOR KNITTING.

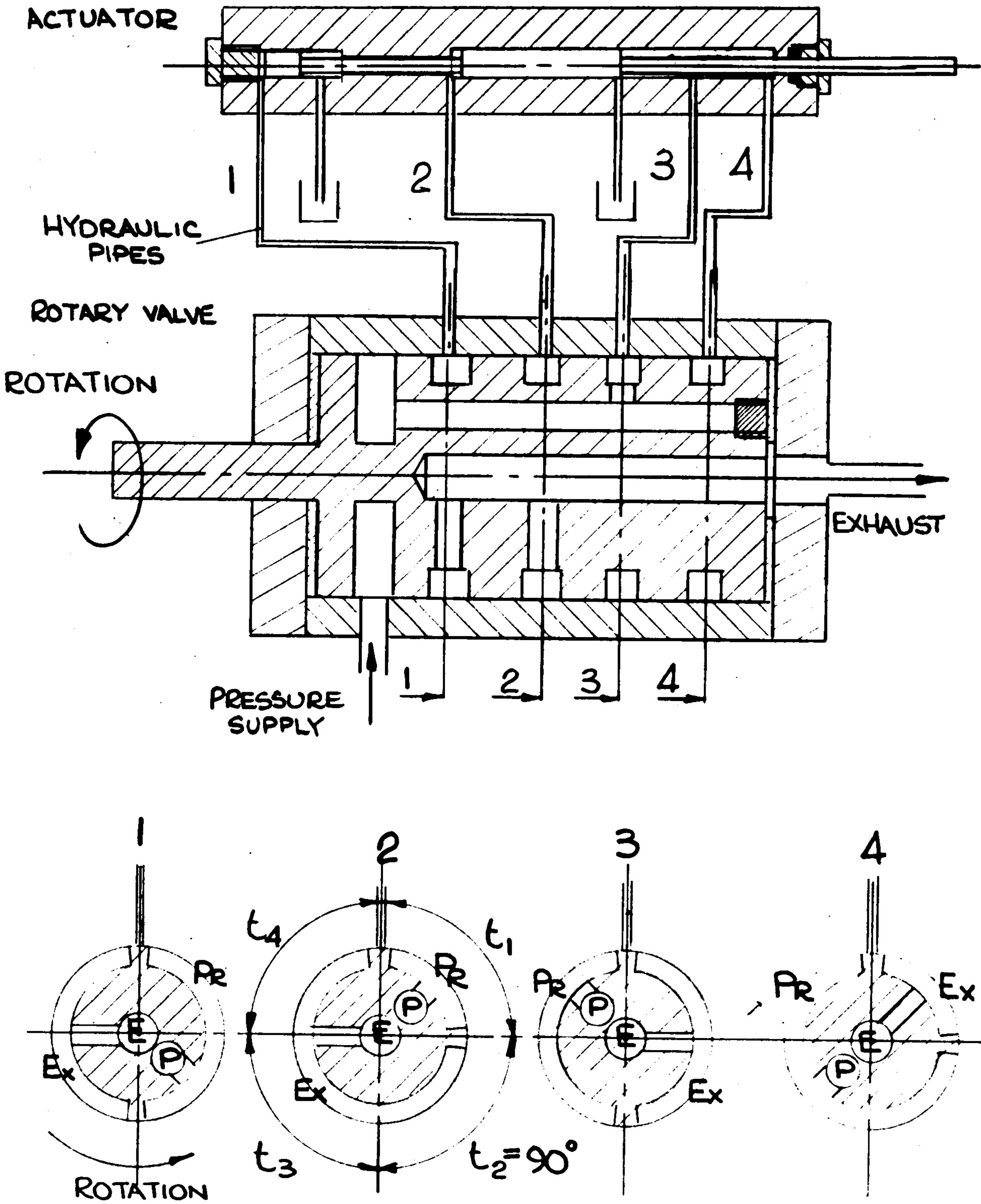


FIG 3

bobbin a radial slot is milled on the surface of the bobbin for the required duration and then linked to the central cavity by a hole drilled normal to the slot. The pressure slots are created by drilling axial holes into an annulus that is supplied with pressurised oil. The surface slots are linked to the pressurised axial holes by drilling normal to the milled slots, thus producing a pressure slot of the required duration.

This technique offers a method of generating a series of cyclic pulses that are capable of driving the hydraulic actuator through the time displacement profile once for every revolution of the bobbin.

To link the actuator to the rotary valve requires hydraulic connections between the ports on the actuator and the ports on the rotary valve. This hydraulic circuit can be seen in figure 3.

It will have been noted that in some instances where a chamber between the piston rod and the top of the cylinder contains two ports (example being ports 3 and 4 when the piston is travelling from the tuck to the miss position) then one port will be redundant. In the example outlined, pressure was maintained at ports 4 for t_3 and t_4 however it is obvious that to block port 4 during this period would have been adequate.

2.5. Variations and Extensions to the Basic Rotary Valve, Sequence Control

In the previous example the time interval required for the actuator movements were specified as being equal, thus each interval occupied 90° of the rotary valve surface. However, the time allowed for each movement can be controlled by the length of slot in the bobbin, the controlling factor being the pulse of oil required by the system. For example the time interval when the yarn is fed into the needle (DE on figure 1) can be increased by lengthening time t_2 and reducing time t_3. The overall length of the slots are calculated by expressing the duration of the pulse as a fraction of the total cycle time.

$\therefore$ Subtended Angle of the slot on the surface of the bobbin = $\frac{\text{Pulse time x } 360^\circ}{\text{Total Cycle time}}$. Thus, the overall dimensions of the displacement profile can be selected at the design stage by adjusting the actuator size and valve bobbin geometry to suit the particular application.

The idea of the rotary valve for driving more than one actuator in sequence with other actuators can be extended. If the basic rotary valve shown in figure 3 is used to drive eight actuators, each operating in sequence with a 45° phase lag between subsequent actuators, the eight sets of pressure supply ports would have to be spaced equally round the rotary valve as shown in figure 4. Thus, actuator 1 would be supplied by ports 1_1, 2_1, 3_1, 4_1, actuator 2 by 1_2, 2_2, 3_2, 4_2, etc. The governing factor as to the number of actuators that can be driven in sequence is the number of pipe

A DIAGRAM TO SHOW THE PORT ARRANGEMENT FOR THE ROTARY VALVE, TO POWER 8 ACTUATORS IN SEQUENCE

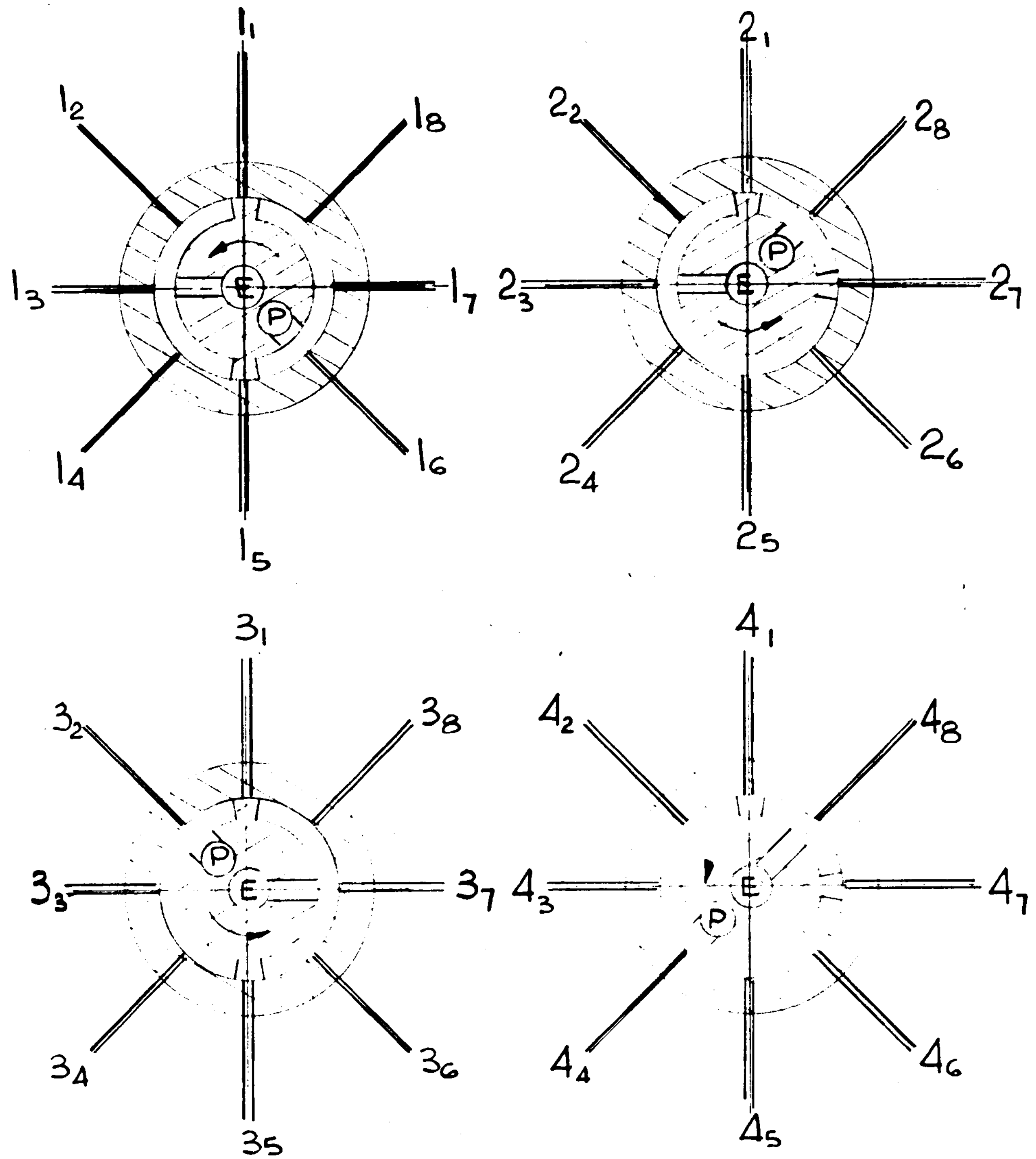


FIG 4

fittings that can physically be placed round the valve. To overcome this limitation, valves have to be made in series. For example if a knitting machine required sixty four needles to work in sequence and the maximum number of fittings that can be placed round the periphery of the valve is sixteen, then a rotary valve (which in effect is four rotary valves joined together) would have to be used. This valve would require sixteen slots, that is, four sets of the slots shown in figure 3. To produce a continual sequence of 1 - 64 needles, the sets of slots would have to be out of phase by $\theta/4$ where θ is the angle between two adjacent pipe fittings. The piping of the actuators would then be as in figure 5 (which for clarity only shows the port 1). Similar rows would exist for ports 2, 3 and 4.

To extend even further; if the knitting machine required was a six hundred and forty needle machine with ten knitting stations, then each of the ports from the rotary valve would have to supply ten actuators, thus port 1_1, would provide power for actuators (1, 65, 129, 193, 257, 321, 385, 449, 513, 577). 2_2 power for (2, 66, 130, 258, 322 etc). These examples are not intended to represent typical machines but merely to indicate how sequence control can be effected to suit all combinations of the size of the knitting station and the overall size of the machine.

2.6. Variations and Extensions to the Basic Rotary Valve, stitch selection control.

To obtain full control over the knitting needle, the aspect of selecting the needle position, that is, whether the

A DIAGRAM TO SHOW THE ARRANGEMENT OF PORT ONE FOR A ROTARY VALVE TO POWER 64 ACTUATORS IN SEQUENCE.

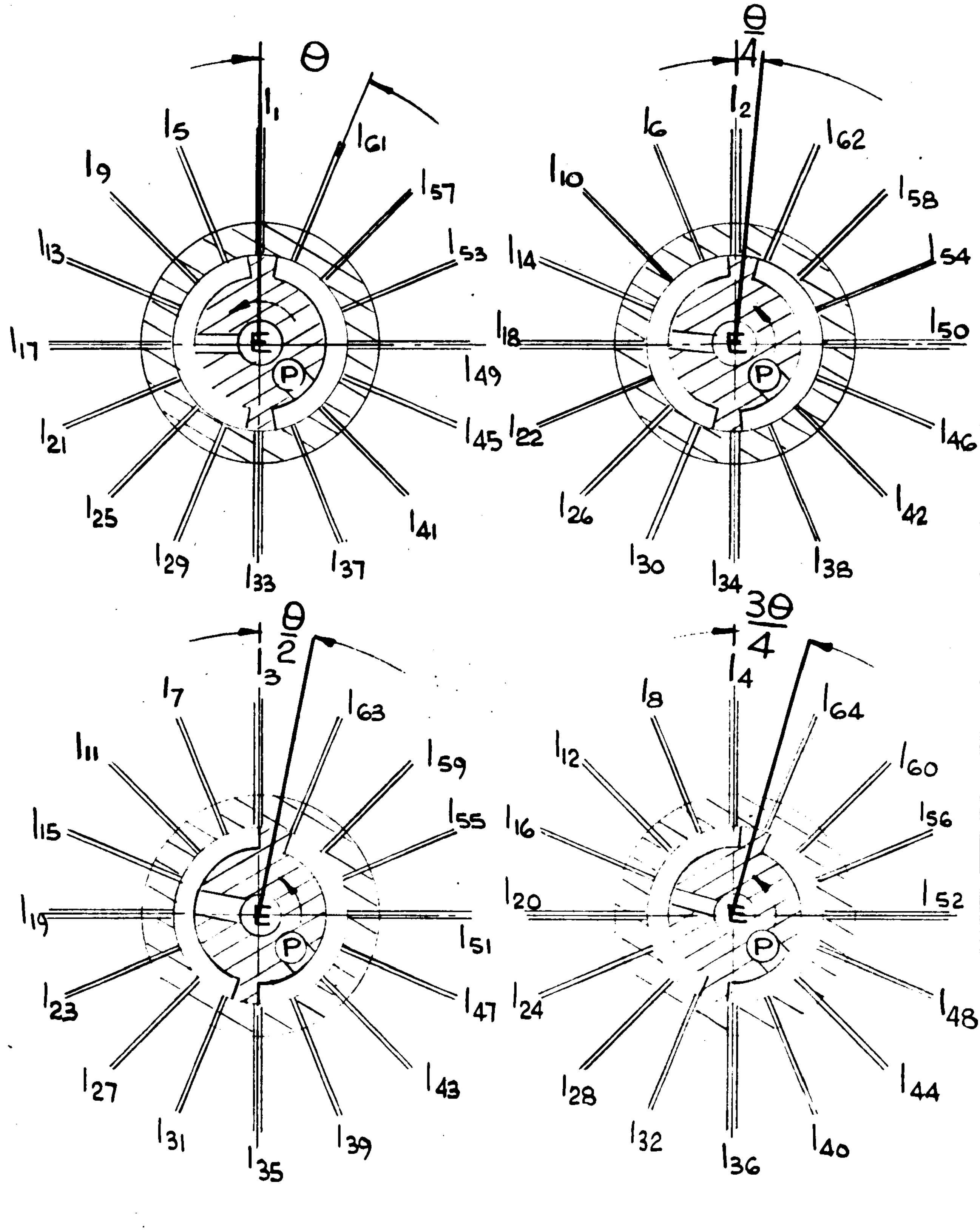


FIG 5

needle operates in the knit, tuck or miss mode, has to be considered. Up to this stage, only the plain knitting profile has been considered, consequently to obtain the tuck motion the actuator amplitude has to be constrained. This is achieved, as explained previously in section 2.3 by preventing the pressure pulse from reaching port 2. The miss stroke is similarly obtained by preventing the pressure pulses from reaching ports 1 and 2 in figure 2. The selection of the needles into a particular configuration of knit, tuck and miss combinations is known as programming.

For relatively simple programmes, i.e. stitch combinations requiring up to eight selections before repeating, the rotary valve can be used as a selecting device thus enabling fixed patterning blocks to be used.

Take, for example, a knitting machine where it is required to make four selections of knit, tuck or miss for each individual needle in sequence before repetition, then the following technique can be used. The rotary valve is designed so that it will supply oil to the actuator from four independent rotor positions for every revolution of the rotary valve. This involves machining the desired slots into a 90° segment of the bobbin thus the cycling rate of the actuator would be four cycles per revolution, with the pulses from the rotary valve being supplied from a different hydraulic circuit in each instance. A pictorial diagram for such a rotary valve can be seen in figure 6. These four separate supply paths, namely a, b, c and d for actuator ports 1 and 2 are taken via a pattern block. This consists of two steel plates with a plastic shim

A DIAGRAM TO SHOW A SECTIONAL VIEW OF A ROTARY VALVE, SUITABLE FOR PROGRAMMING REPETITIVE NEEDLE COMBINATIONS

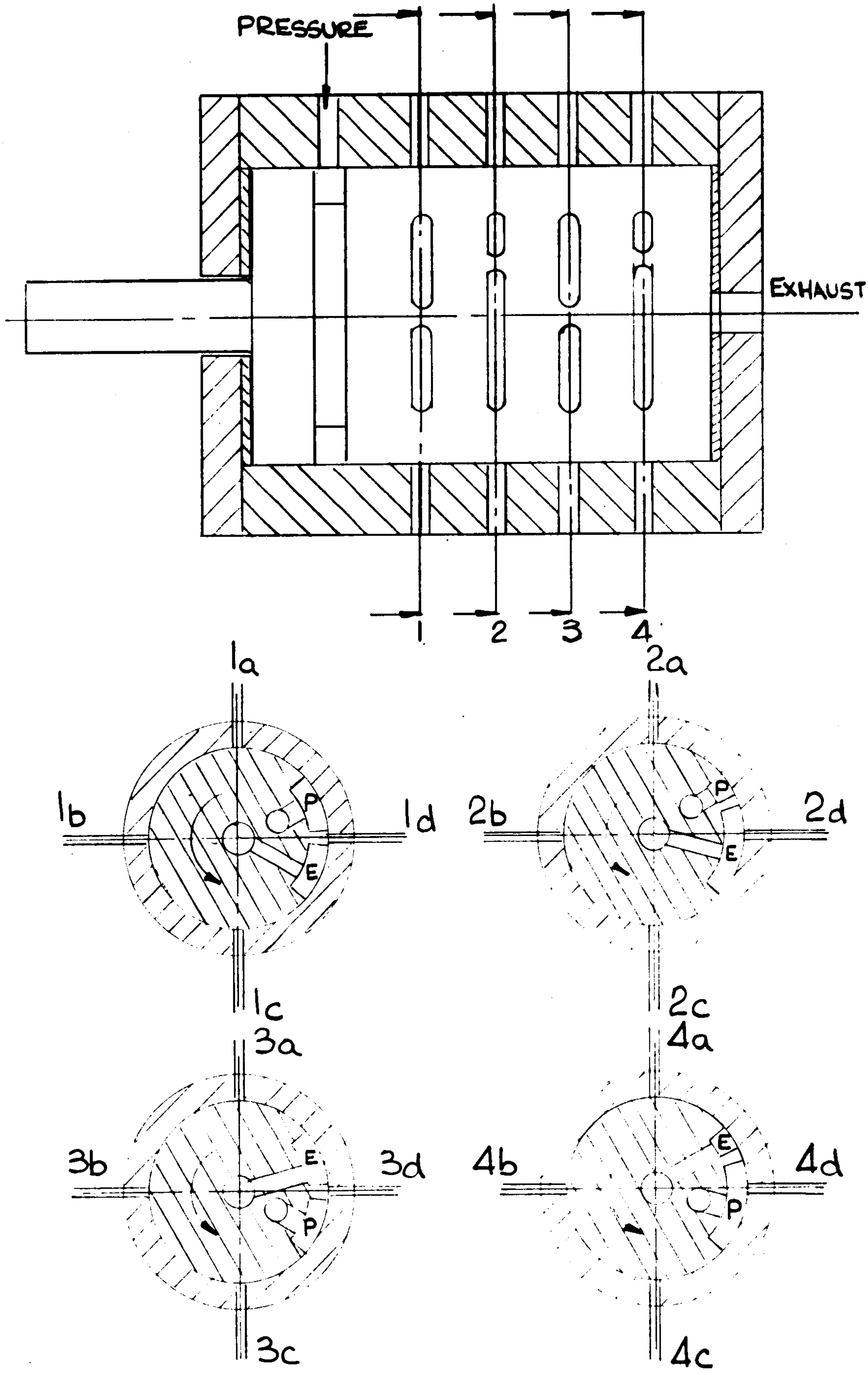


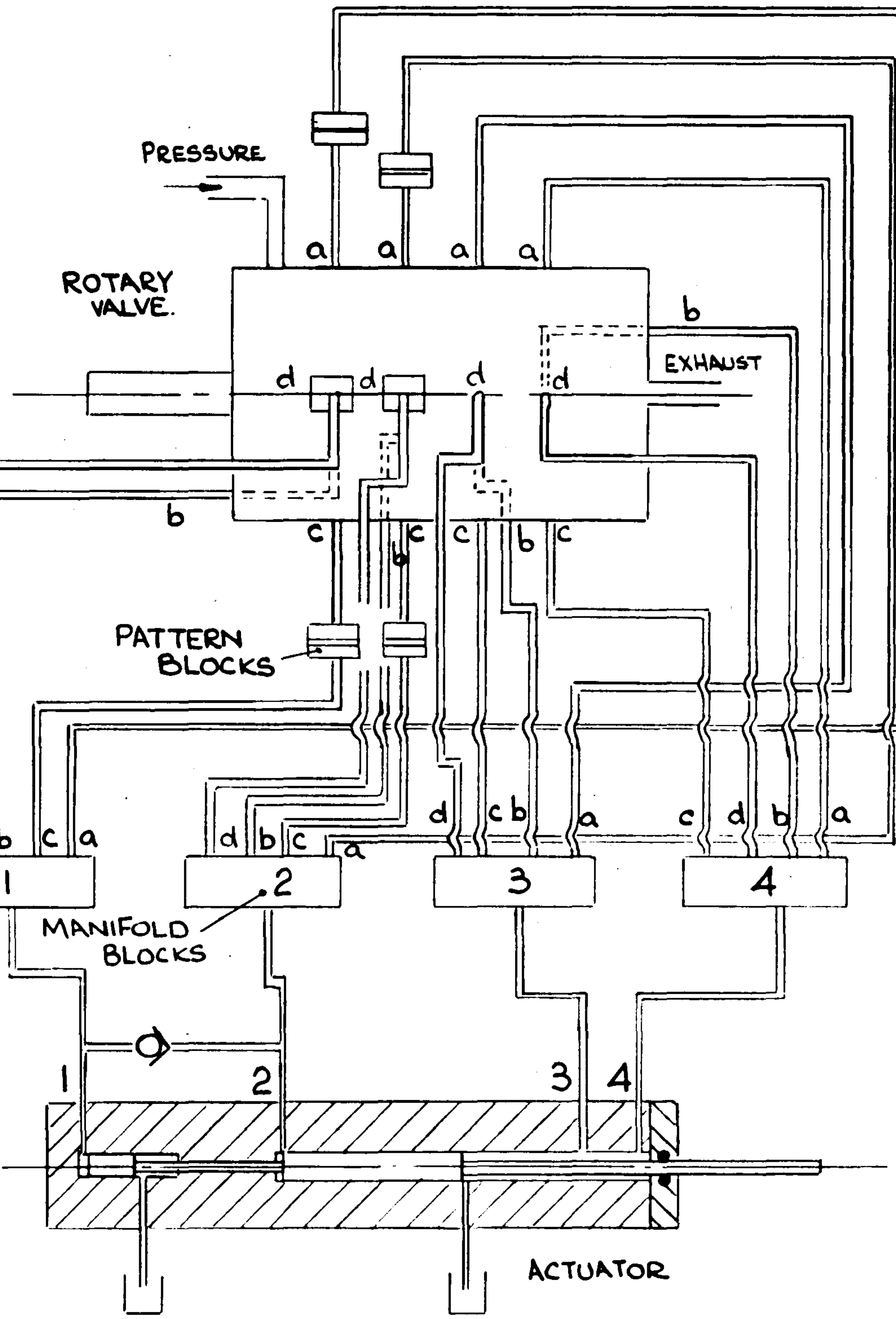
FIG 6

clamped between them; if a hole is drilled the actuator receives the pulse of oil, if the hole is omitted the supply line is blocked. The hole punching technique enables each individual needle to be programmed for four selections before repetition. Once the pressure signal has passed through the pattern block, the four separate lines can be taken to a manifold before being connected to the actuator. A circuit diagram for such a valve actuator combination can be seen in figure 7. A one-way valve has to be introduced to enable the residual oil at port 2 to be exhausted when the actuator operating in the tuck mode is returning from the tuck position. (It will have been noted that the supply line 2 has been blocked for patterning purposes and the oil trapped between the piston rod and port 2 cannot be exhausted). This technique could usefully be employed when producing mono-colour, textured fabric, where the tuck stitch is used to produce a regular surface pattern.

2.7. The Prototype Multi-Actuator Block.

The basic design for the hydraulic actuators were of the type previously tested, but to accommodate the actuators at intervals of a sixteenth of an inch a packaging exercise had to be completed. This resulted in alternate actuators being supplied with oil from opposite sides of the block, with the actuators staggered in both a horizontal and vertical plane. The orientation of the actuators is clearly defined in Drawing Numbers J.D.G. 030 and J.D.G. 031 which can be seen in figures 8 and 9. The size limitation imposed by using sixteen needles per inch prevented the use of conventional

A CIRCUIT DIAGRAM TO SHOW THE HYDRAULIC CONNECTIONS BETWEEN A ROTARY VALVE, WITH A PATTERNING FACILITY AND AN ACTUATOR.



O) ONE WAY VALVE.

FIG 7

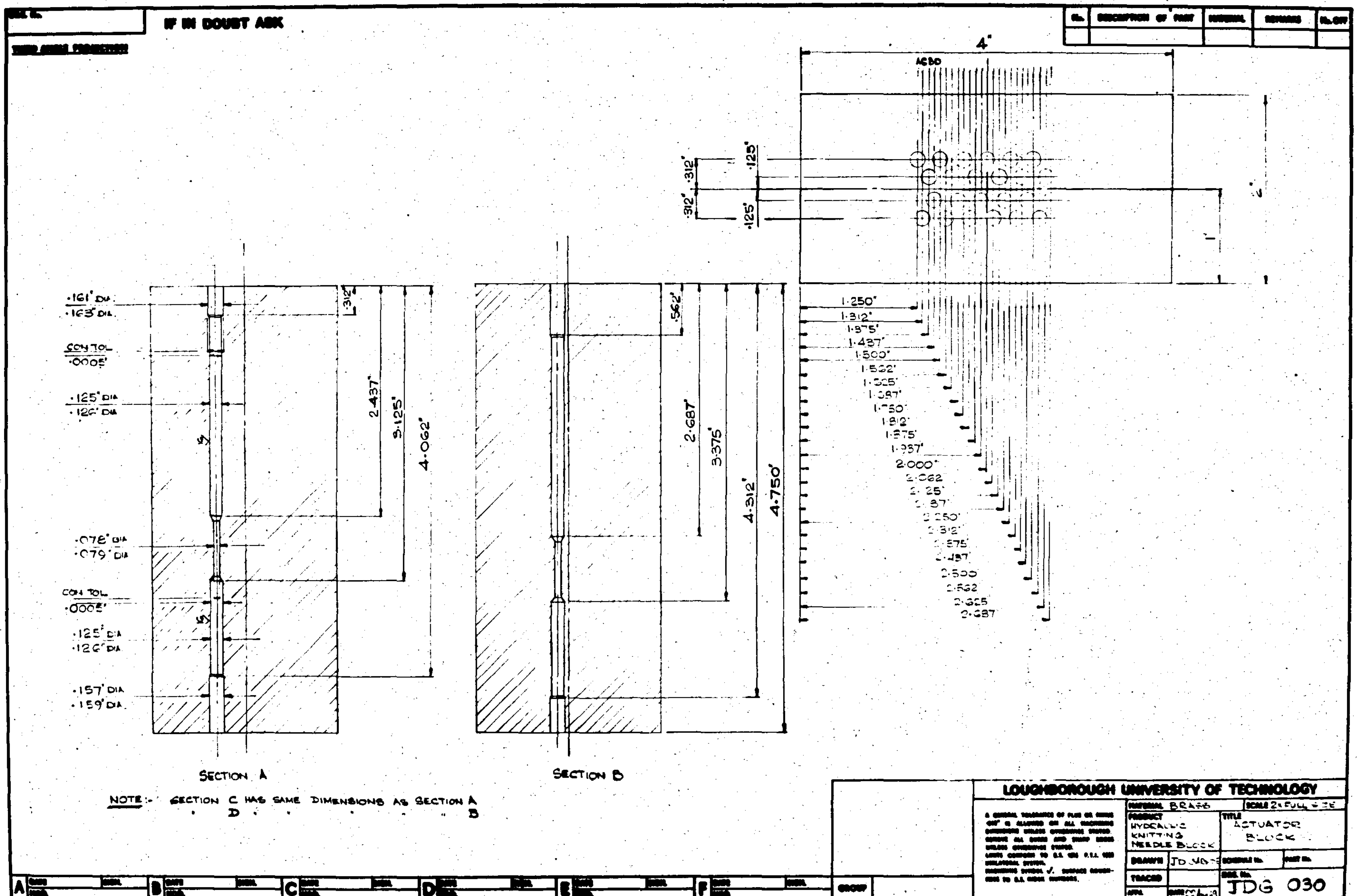

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
4"
1"
2"
·312"
·312"
·125"
·125"
1·250"
1·312"
1·375"
1·437"
1·500"
1·562"
1·625"
1·687"
1·750"
1·812"
1·875"
1·937"
2·000"
2·062
2·125"
2·187"
2·250"
2·312"
2·375"
2·437"
2·500
2·562
2·625"
2·687"
·161" DIA
·163" DIA
CON TOL
·0005"
·125" DIA
·126" DIA
·078" DIA
·079" DIA
CON TOL
·0005"
·125" DIA
·126" DIA
·157" DIA
·159" DIA
·312"
2·437"
3·125"
4·062"
·562"
2·687"
3·375"
4·312"
4·750"
SECTION A
SECTION B
NOTE:- SECTION C HAS SAME DIMENSIONS AS SECTION A
" D " " " " B
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL BRASS
PRODUCT HYDRAULIC KNITTING NEEDLE BLOCK
TITLE ACTUATOR BLOCK
DRAWN JD
TRACED
DWG No. JDG 030

FIG. 8

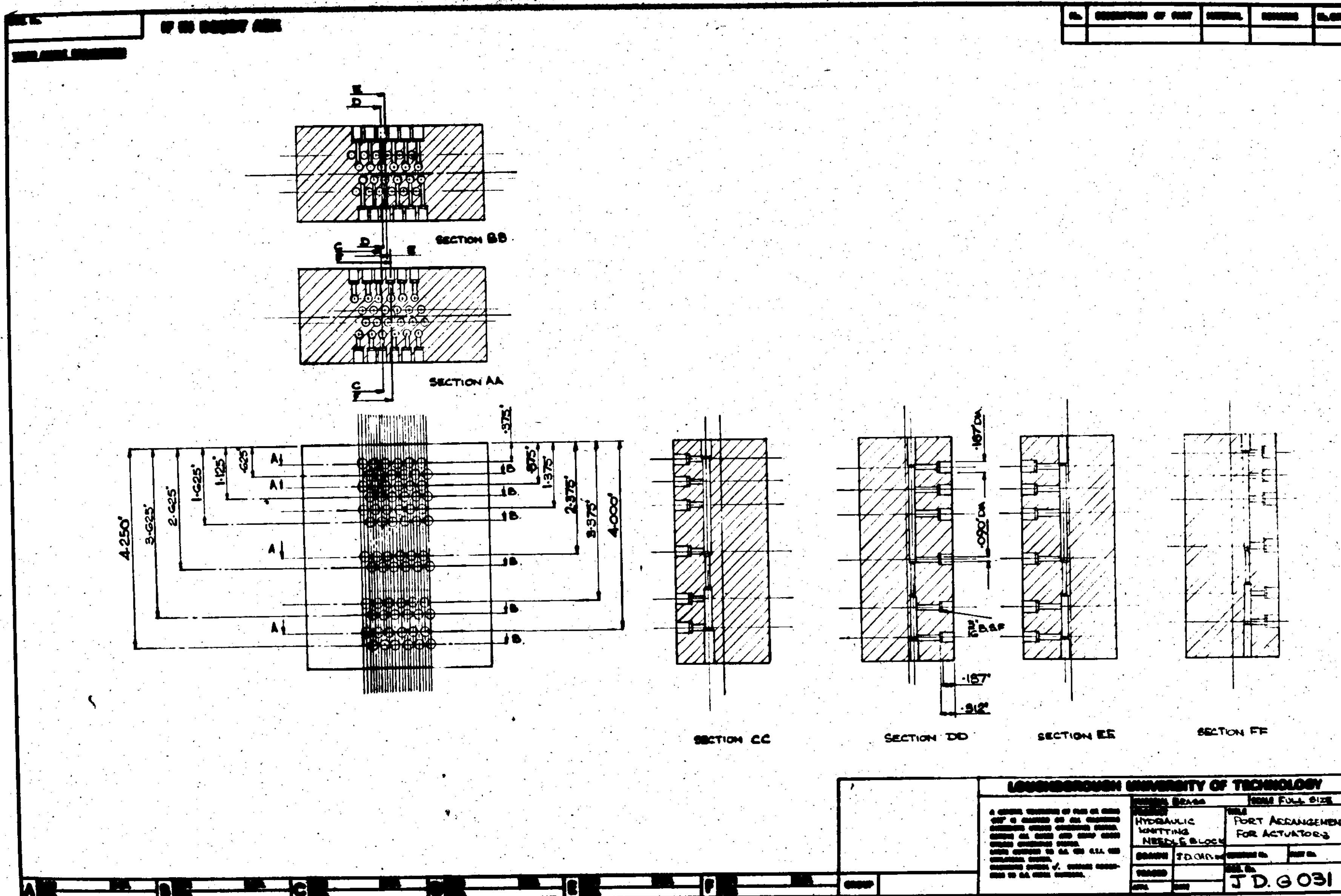

SECTION BB
SECTION AA
SECTION CC
SECTION DD
SECTION EE
SECTION FF
4·250"
3·625"
2·625"
1·625"
1·125"
·625"
·375"
·875"
1·375"
2·375"
3·375"
4·000"
·187"DIA
·090"DIA
·157"
·312"
B.S.F
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
FULL SIZE
HYDRAULIC KNITTING NEEDLE BLOCK
PORT ARRANGEMENT FOR ACTUATORS
J D. G 031

FIG. 9

pipe connections, consequently provision was made to solder the pipes directly into the block, hence brass was chosen for the actuator block. The sealing of the pistons in the actuator employed a technique developed on a final test actuator (see figure 10). The 'O' ring seal for the piston rod is held in position by a spacer of larger diameter than the cylinder bore, and a top plate fastened over the actuator with small screws. A similar spacer and cover plate was used to retain the tuck probe in position.

2.7.1. Design of the Rotary Valve.

The Rotary Valve drawings, J.D.G. 028, J.D.G. 027, J.D.G. 025 and J.D.G. 026 which can be seen in figures 11, 12, 13 and 14 respectively show its basic construction. This Rotary Valve is a complex device in that it uses the techniques outlined in sections 2.5 for sequence control and 2.6. for introducing patterning facility. The bobbin is basically three rotary valves of the type shown diagrammatically in figure 6, joined in series. Examining drawing number J.D.G. 026, figure 14, it can be seen that the three sections have been displaced by 120° so that the slots are distributed evenly round the surface of the bobbin. This displacement is principally to balance the hydrostatic pressure distribution produced by the slots, and to aid the hydrodynamic lubrication of the bobbin as it rotates in the cylinder bore. The three units consist of:-

Unit 1	sections AA, BB, LL, MM
Unit 2	sections CC, DD, JJ, KK
Unit 3	sections EE, FF, GG, HH

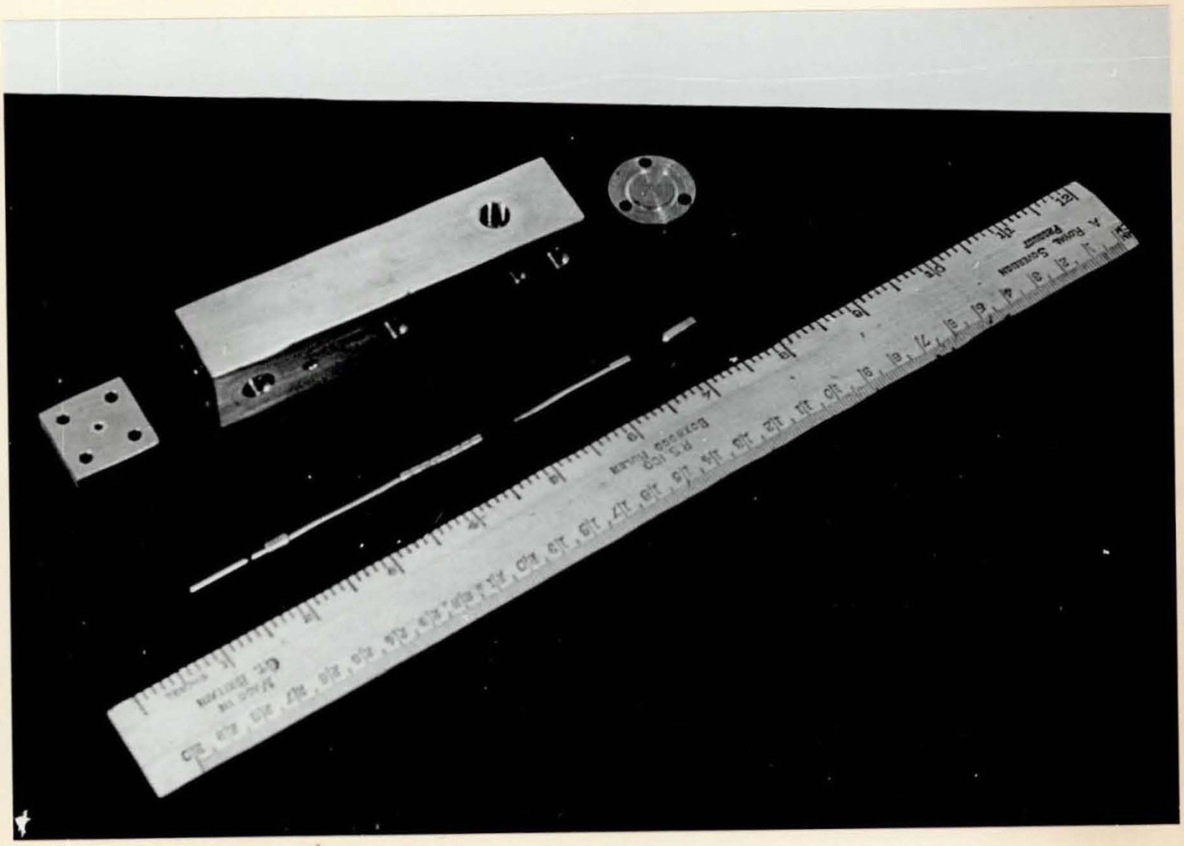
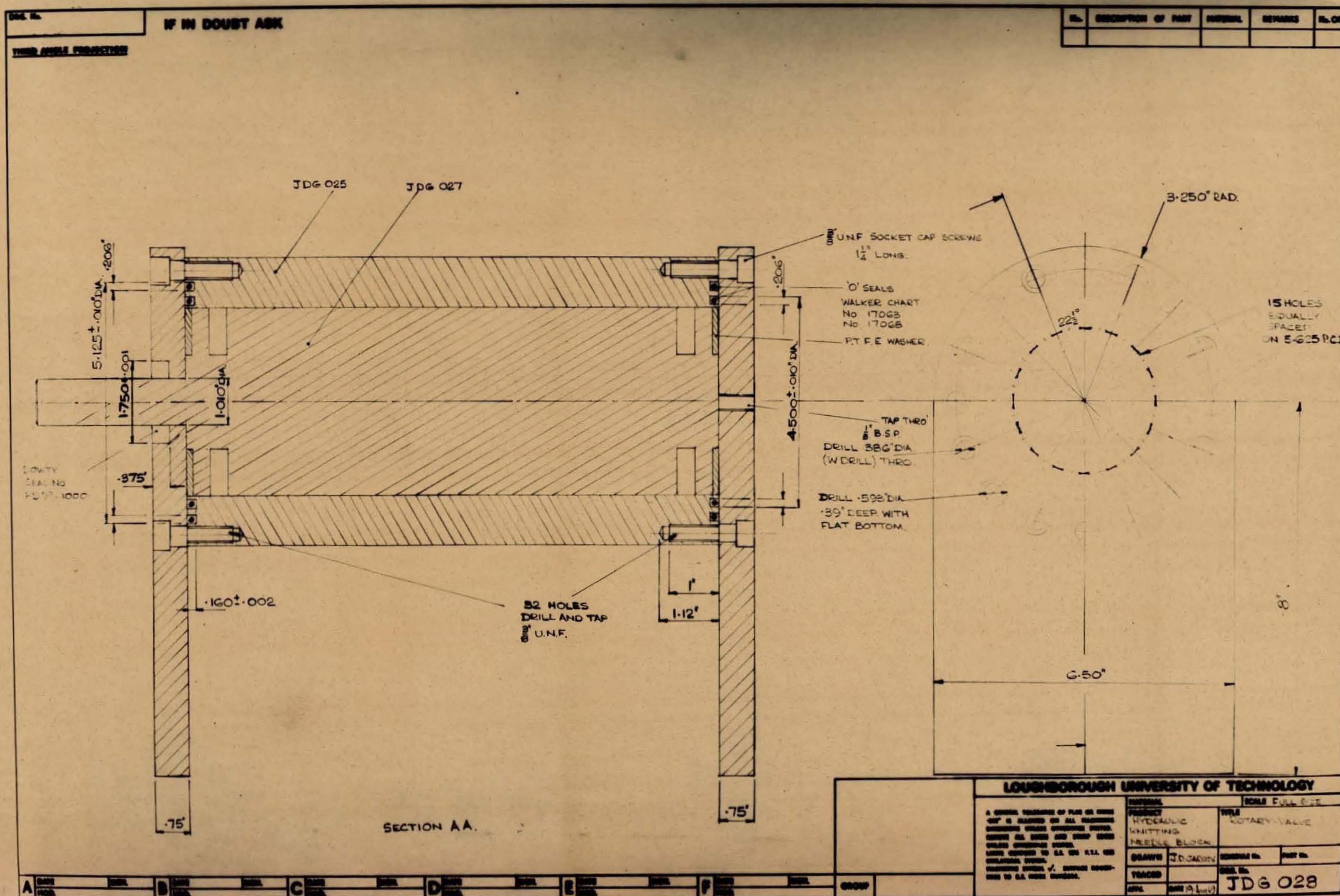


FIG. 10 FINAL TEST ACTUATOR



IF IN DOUBT ASK
THIRD ANGLE PROJECTION
JDG 025
JDG 027
3/8" U.N.F SOCKET CAP SCREWS 1 1/4" LONG.
'O' SEALS WALKER CHART No 17063 No 17068
P.T.F.E. WASHER
TAP THRO' 1/8" B.S.P.
DRILL .386" DIA (W DRILL) THRO.
DRILL .593" DIA .39" DEEP WITH FLAT BOTTOM.
32 HOLES DRILL AND TAP 3/8" U.N.F.
3.250" RAD.
22 1/2°
15 HOLES EQUALLY SPACED ON 5.625 P.C.D.
6.50"
5.125 ± .010" DIA.
.206"
1.750 ± .001
1.010" DIA.
4.500 ± .010" DIA.
.375"
.160 ± .002
1"
1.12"
.75"
SECTION AA.
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
SCALE FULL SIZE
HYDRAULIC KNITTING NEEDLE BLOCK
ROTARY VALVE
J.D. GADSBY
JDG 028

FIG. 11

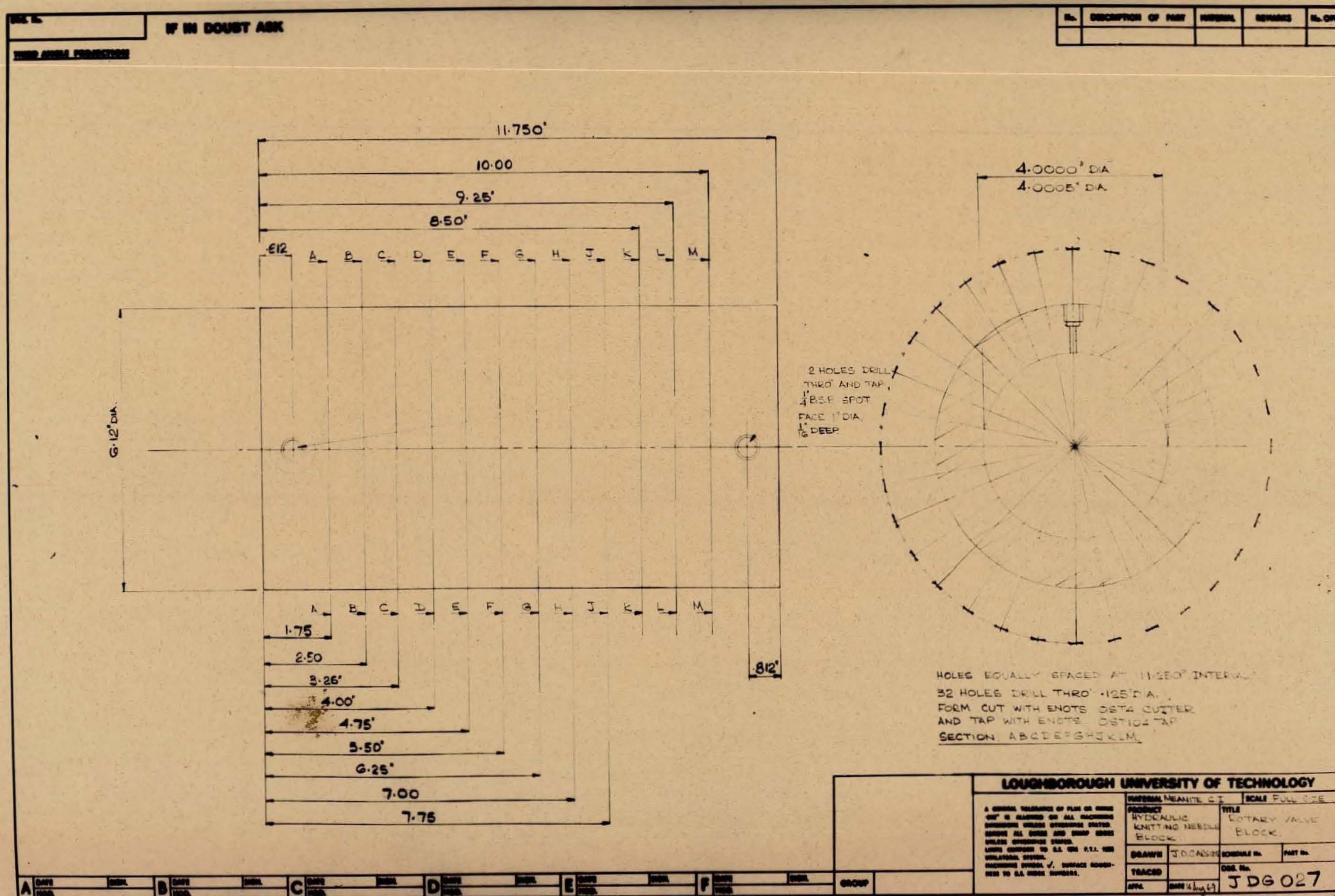

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
11·750"
10·00
9·25"
8·50"
·812
A B C D E F G H J K L M
G·12" DIA.
2 HOLES DRILL THRO' AND TAP ¼" B.S.F. SPOT FACE 1" DIA. 1/16" DEEP
1·75
2·50
3·25"
4·00"
4·75"
5·50"
6·25"
7·00
7·75
·812"
4·0000" DIA
4·0005" DIA
HOLES EQUALLY SPACED AT 11·250° INTERVALS
32 HOLES DRILL THRO' ·125" DIA.
FORM CUT WITH ENOTS DST4 CUTTER AND TAP WITH ENOTS DST104 TAP
SECTION ABCDEFGHJKLM.
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL MEANITE C.I.
SCALE FULL SIZE
PRODUCT HYDRAULIC KNITTING NEEDLE BLOCK
TITLE ROTARY VALVE BLOCK
DRAWN J.D.G.
DRG. No. JDG 027

FIG. 12

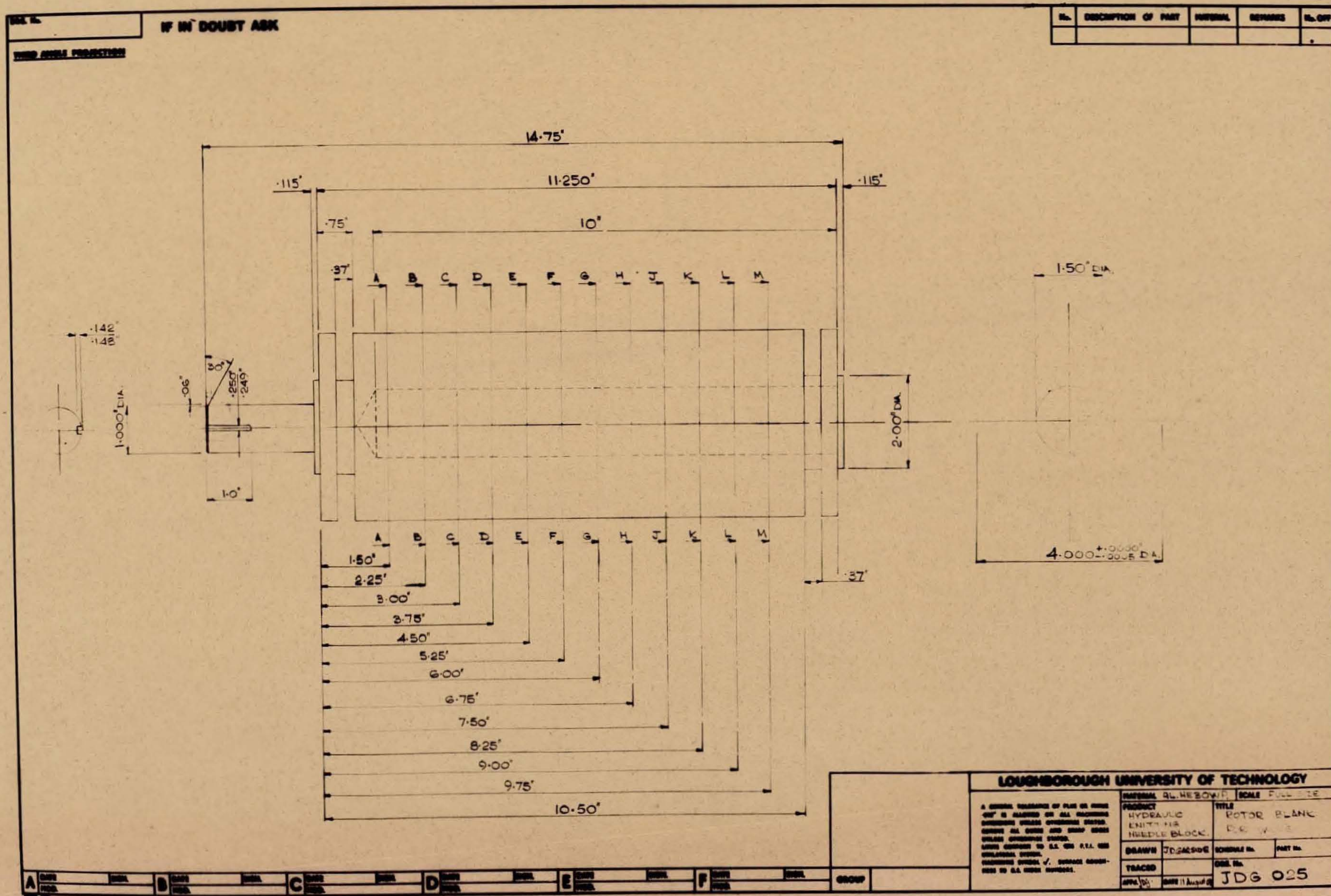

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
14·75"
·115"
11·250"
·115"
·75"
10"
·37"
A B C D E F G H J K L M
·142"
·148"
1·000" DIA.
·06"
30°
·250"
·249"
1·0"
2·00" DIA.
1·50" DIA.
4·000 +·0000" −·0005 DIA.
1·50"
2·25"
3·00"
3·75"
4·50"
5·25"
6·00"
6·75"
7·50"
8·25"
9·00"
9·75"
10·50"
·37"
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL AL. NEBOWN
SCALE FULL SIZE
PRODUCT HYDRAULIC KNITTING NEEDLE BLOCK
TITLE ROTOR BLANK
DRAWN J.D. JACKSON
TRACED
DWG. No. JDG 025
GROUP

FIG. 13

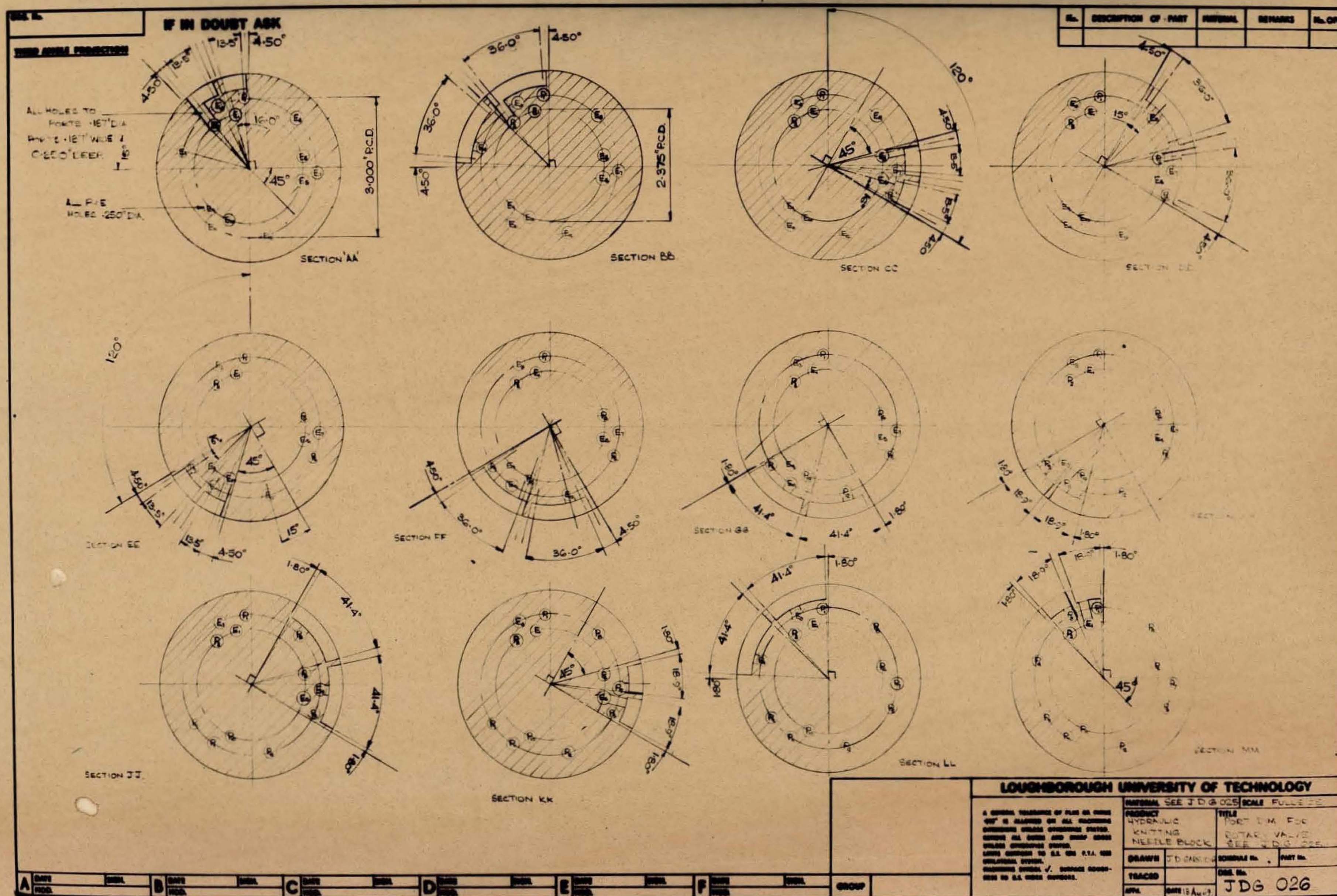

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
ALL HOLES TO PORTS ·187" DIA
ALL FIVE HOLES ·250" DIA
3·000" P.C.D.
2·375" P.C.D.
SECTION 'AA'
SECTION BB
SECTION CC
SECTION EE
SECTION FF
SECTION GG
SECTION JJ
SECTION KK
SECTION LL
SECTION MM
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL SEE J D G 025
SCALE FULL SIZE
PRODUCT HYDRAULIC KNITTING NEEDLE BLOCK
DRAWN
TRACED
DRG. No. JDG 026

FIG. 14.

Using the actuator as defined in figure 2, the actuator ports will be supplied by the following sections:-

Port 1	sections BB, DD, FF
Port 2	sections AA, CC, EE
Port 3	sections GG, JJ, LL
Port 4	sections HH, KK, MM

(It will be noticed from the drawings that the exhaust slots in the bobbin are connected to an annular exhaust groove similar to the pressure groove. The length of the exhaust slot in sections AA, CC, and EE only occupy 18.9°. These are two modifications that will be explained in section 2.8).

At each section, the maximum number of pipe fittings that could be accommodated round the periphery of the cylinder was thirty two, hence the three sections gave the total number of ninety six actuator outlets from the rotary valve. As each set of slots was confined to a rotor sector of 90°, each of the twenty four actuators could be supplied from four independent supply ports thus enabling the needles to be programmed for four movements before repeating the sequence.

The materials used to manufacture the Rotary Valve were Aluminium HE 30WP for the bobbin and Meehanite Cast Iron for the cylinder. Aluminium was chosen for its easy machining properties. Meehanite Cast Iron was chosen as it is a good bearing material suitable for use with hydraulic oil.

2.7.2. Assembly of the Rotary Valve.

The various components required for the multi-actuator block and the rotary valve were manufactured as specified on the detail drawings. Meanwhile, a suitable frame for mounting the various components was constructed. The frame had to support a variable speed motor, the rotary valve, the pattern and manifold blocks plus the actuators, consequently the frame was solidly constructed. A Carter variable speed gearbox was mounted at one end of the frame as can be seen in figure 15. This gear box is basically a variable displacement hydraulic piston pump and a fixed displacement motor. It was used for turning the rotary valve because it was readily available and possessed the required speed-torque characteristics. The rotary valve was mounted on cross plates and connected to the variable speed motor by a Fenner flexible coupling (see figure 15). The alignment of the two shafts was critical but clearance was allowed in the fixing holes to enable the rotor to be set up. Once the rotor had been positioned and found to run smoothly over relatively long periods the framework carrying the pattern and manifold blocks was positioned and bolted into place. (see figure 16). The pattern blocks consisted of two steel plates with a sheet of vinyl sandwiched between them. The manifold blocks enabled the four separate supply paths to be linked. For the actuators to operate in sequence it was necessary for the correct supply ports of the rotary valve to be linked to the correct actuators. The outlet ports, to supply a particular actuator, on the rotary valve were staggered round its periphery, so preventing symmetrical pipe formation between rotary valve and manifold blocks. As can be seen in figure 17, the path for the piping between the rotary valve and the manifold blocks is completely random, the only essential

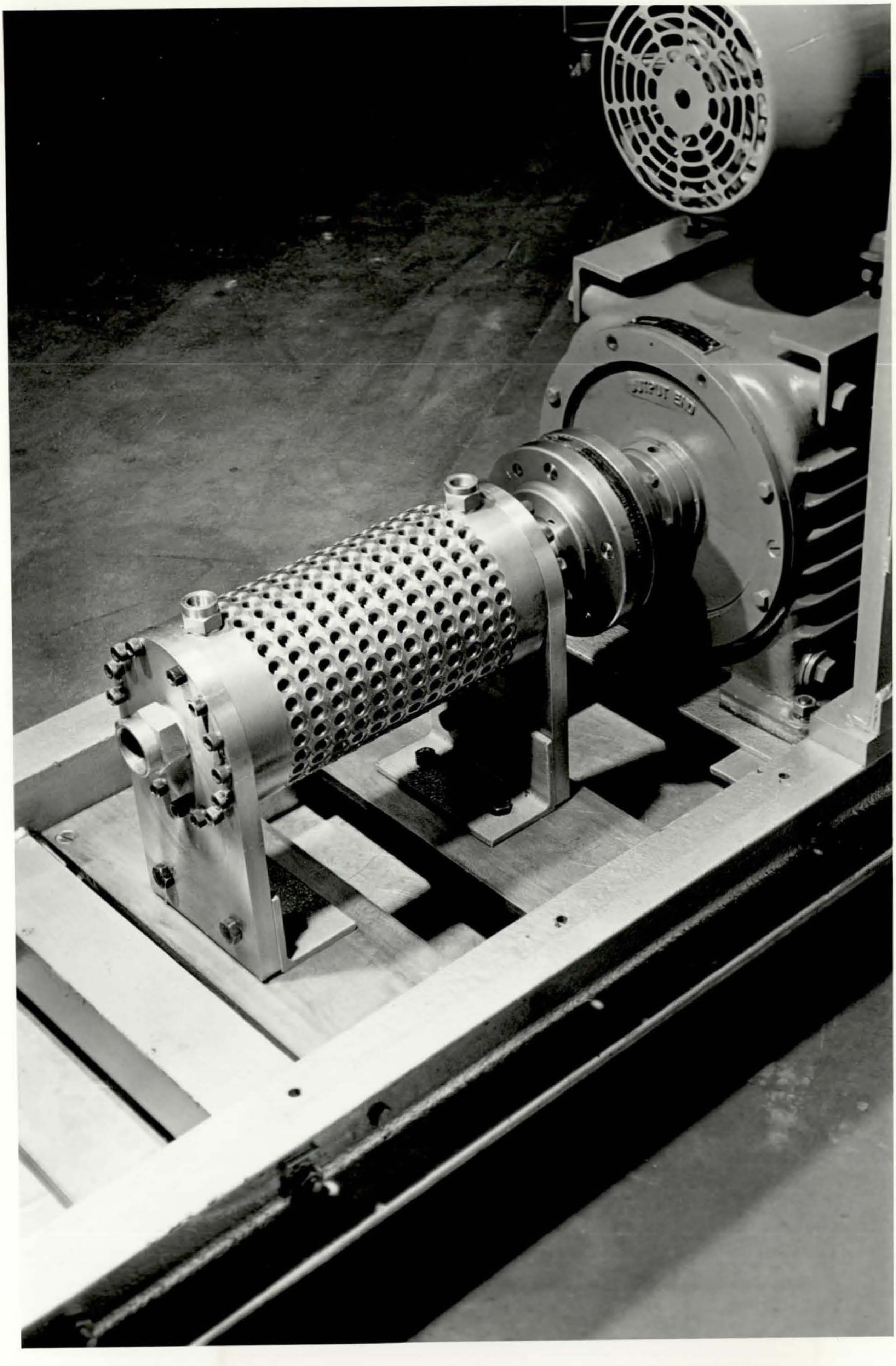


FIG. 15 ROTARY VALVE FRAME AND DRIVE MECHANISM

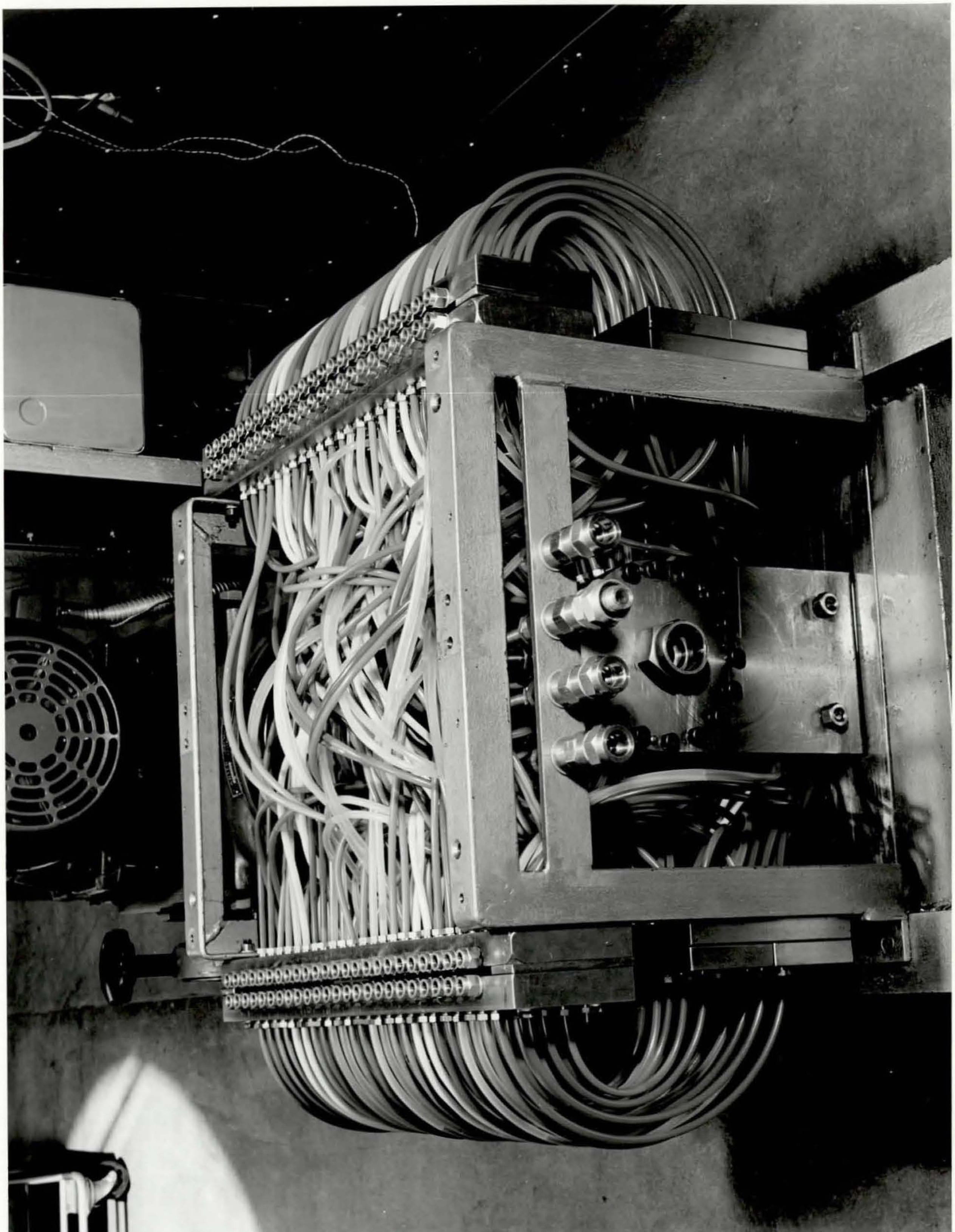


FIG. 16 FRAMEWORK FOR THE PATTERN AND MANIFOLD BLOCKS

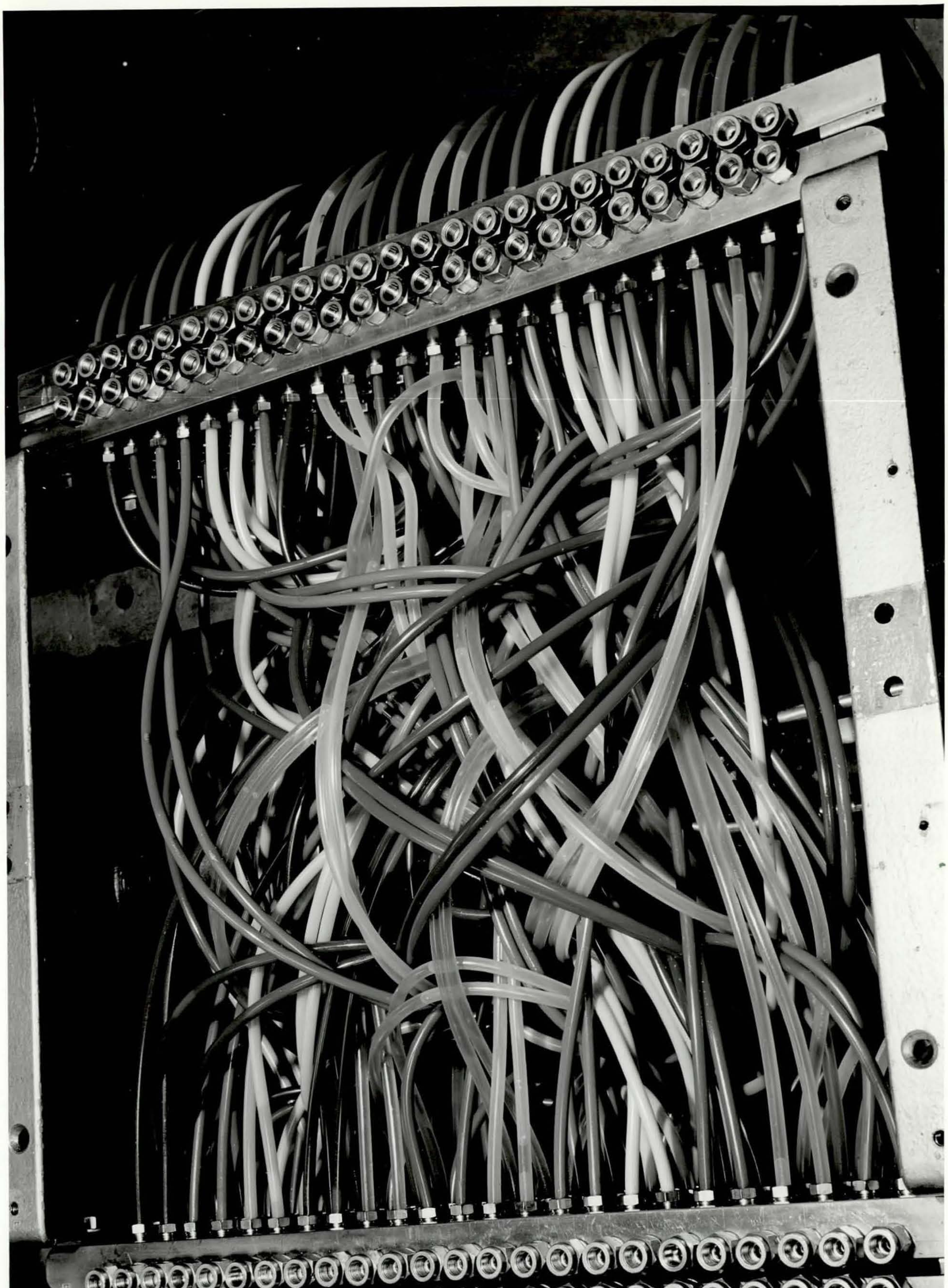


FIG. 17 FLEXIBLE PIPE CONNECTIONS

detail being that the outlet from the rotary valve was connected to the correct place in the manifold block. To simplify this task, flexible nylon tubing of .25 inches external diameter was used in five different colours. The colours were used in sequence round the rotary valve, one colour being used to link all the four separate supply paths to a particular actuator. For example, actuator one had sixteen red pipes joined into the first four positions in the manifold block, while pipes for actuator two occupied the first four positions in the manifold blocks at the opposite side of the rotor. The piping of the system up to the stage shown in figures 16 and 17 was not as complex as it might appear after gaining an understanding of the system. The use of coloured pipe provided a visual check as to actuator sequence, but no definite proof of this could be obtained until the actuators moved correctly in sequence. The next consideration was the hydraulic stops; each actuator had to be linked to exhaust via a restrictor, this being used to limit the volume of oil pumped to exhaust at the end of each movement. Previous tests on hydraulic stops had indicated that ten inches of .058 inches internal diameter pipe was sufficient to allow the static pressure to be exhausted whilst keeping the wasted energy at an acceptable level. Each actuator required two hydraulic stops so an exhaust gallery was placed alongside the manifold blocks into which all the hydraulic stops could be connected. Thus all the connections that had to be made to the actuators were grouped into blocks of six, with the sequence running from alternate manifold blocks mounted at each side of the

Rotary Valve. The pipework to each actuator could then be made symmetrical.

2.7.3. Assembly of the Actuators.

Once the rig had reached this stage of development, work assembling the actuators commenced. The knitting pistons and the tuck probes had been manufactured from Kelock 795 which is a 14% tungsten high speed steel recommended for its toughness in ardous punching operations. These components required a centreless grinding facility which was not available within the Department.

Each actuator was assembled individually, ensuring that both pistons were free to slide axially in their respective bores, before fitting the distance collar and 'O' ring, as shown in figure 10. By this stage it was realised that soldering the pipes directly into the actuator body was not feasible because of the thermal distortions that might occur. The pipes therefore, were all silver soldered, using a low melting point compound, into two face plates that could be screwed to the face of the actuator block. The pipes were allowed to protrude through the face plate and into the previously drilled actuator block. The side plates were then screwed to the actuator block with a thin Walkerite gasket clamped between the two surfaces. The assembled actuator block and pipes can be seen in figure 18. A view of the rig from both sides can be seen in figures 19 and 20. Here the completed multi-actuator rig is shown together with the variable speed drive. It will be noted that the hydraulic

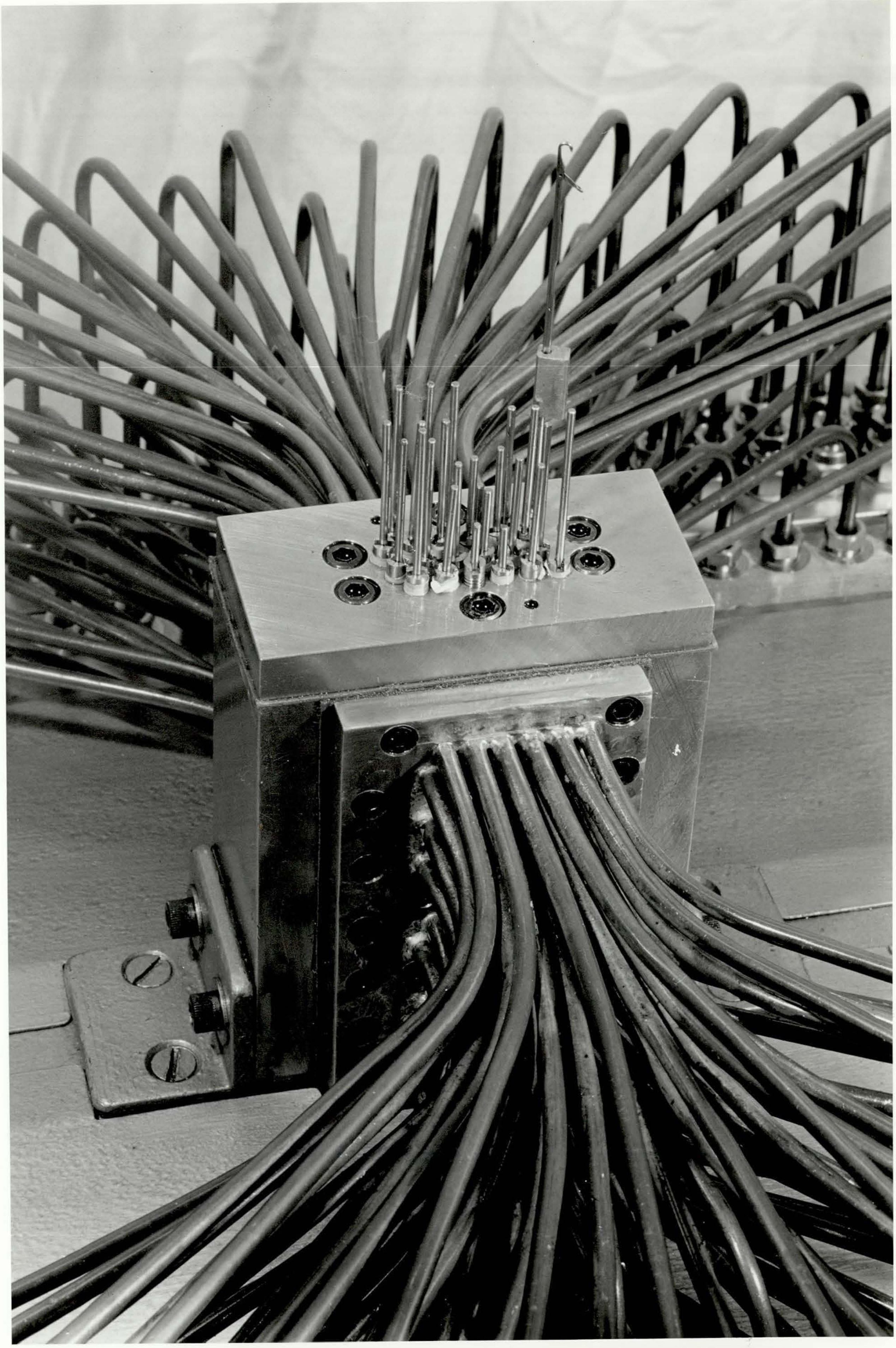


FIG. 18 ASSEMBLED ACTUATOR BLOCK

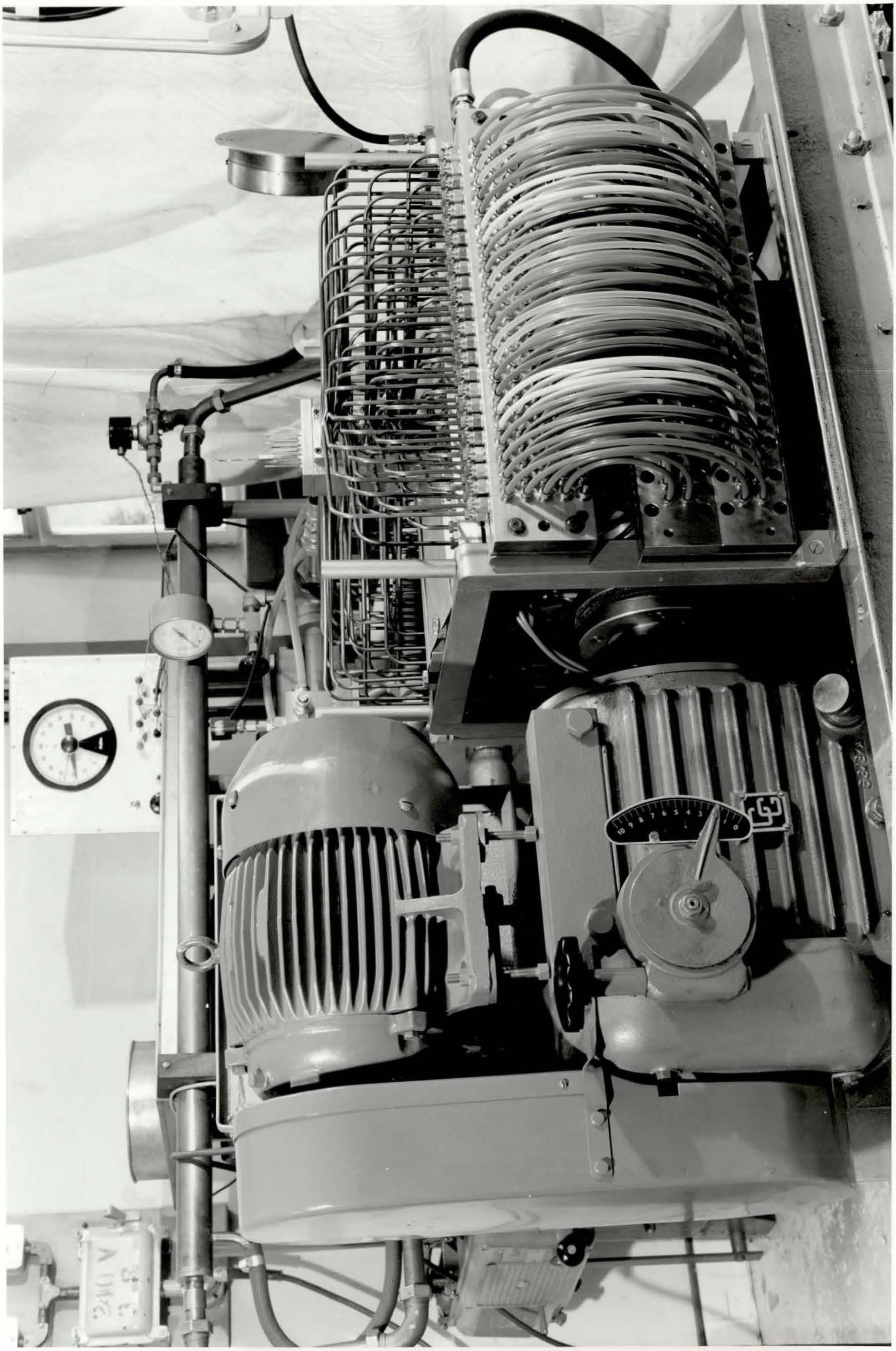


FIG. 19 COMPLETE MULTI-ACTUATOR RIG SHOWING THE SPEED CONTROL MECHANISM

FIG. 20 COMPLETE MULTI-ACTUATOR RIG SHOWING STARTER SWITCH

pipework on the outside of the rig is symmetrical and that all the links between the manifold blocks and the actuator have been formed in copper pipe.

2.8. Testing the Mulit-Actuator Block

2.8.1. The Actuators

A hydraulic power pack was connected to the rig and oil was pumped through the system with the rotor stationary to flush away any contamination and to fill the pipes with fluid. The system was then run for a considerable period with the relief valve fully open in order to check the hydraulic pipe connections. After this initial stage the relief valve was closed and the actuators were seen to move. The sequencing of the actuators was visible at the lower speeds and it was accepted that the inter-connections between the rotor and the actuators must be correct. After the initial proving run, a modification had to be made to the sealing plate on the top of the actuators. This was because a plate to cover twenty four actuators was not able to accommodate the variation in machine tolerances between actuators. This limitation was partially overcome by machining a new top plate with individual gland nuts for each actuator. This arrangement can be seen in figure 18. Here each 'O' ring can be nipped to suit the piston rod, thus allowing each actuator to be adjusted separately. Once this modification had been made, the actuator block remained unchanged throughout the tests.

2.8.2. The Rotary Valve.

The multi-actuator block was run over several long periods to ensure that the system was run in before applying any instrumentation. During this period it was noticed that when running the system at rotor speeds above 350 revolutions per minute, with a high supply pressure, the motor would stall after a few minutes. When the rotor was removed, no signs of metal to metal contact could be detected. However, the rotor was basically a bearing which relied on a hydrodynamic pressure to prevent seizure. Consequently, in order to centralise the rotor, grooves were machined at regular intervals of .25 inches. These grooves served two basic purposes:-

(i) to equalise the pressure distribution round the periphery of the bobbin hence tend to centralise it.

(ii) to ensure that a supply of oil was maintained in the surface between the bobbin and the cylinder, thus promoting a hydrodynamic bearing action.

A further modification at this stage was the introduction of an 'O' ring between the high pressure groove and the end of the bobbin. The purpose of this seal was to prevent the high pressure oil from acting on the back of the bobbin, so eliminating a hydraulic ram action, crushing the P.T.F.E. thrust bearing. Both these modifications gave improved rotary valve performance. The circumferencial grooves reduced the static torque although some of the power saved by the 'O' ring had to be consumed overcoming its frictional resistance.

No actual torque measurements were recorded, but a measure of the performance could be gauged by the pressure required to stall the rotor.

The actuators performed as predicted, but it was noticed that the supply pressure required for the rig was 50% higher than that required to run each actuator individually using a servo valve. This indicated that the pressure pulses from the rotary valve were not of the same magnitude as those of the servo valve.

2.8.3. Instrumentation.

To investigate this discrepancy, four S.E. Laboratories variable reluctance pressure transducers were used and this type of pressure instrumentation was used throughout the project for all dynamic pressure measurements. Before a value of pressure could be obtained, the pressure transducer had to be calibrated for a particular channel and galvanometer. The simplest method of calibrating the pressure transducer was to mount it onto a dead weight pressure gauge and record traces of various known pressures within the required range. Once this calibration had been performed the instruments could be readily set up, provided that the same Attenuation and Gain settings were used with the same carrier signal and galvanometer.

In this instance, the four pressure transducers were all calibrated to give the same output for a given input signal. This was achieved by adjusting the Attenuation and

Gain settings for each channel to produce the same output as the largest transducer. The details of the amplifier settings, together with the results obtained from calibrating the transducers against a dead weight pressure gauge, can be seen in Table 2.

2.8.4. Testing the Rotary Valve to check the output Pulses

The pressure pulses to one actuator were recorded at the rotary valve, the manifold block, and the actuator itself using various supply pressures and rotor speeds. A sample of these recordings can be seen in figures 21, 22 and 23. Figure 21 shows the absolute pressure pulses at the rotary valve. Here, the transducers only are connected to the supply ports, so creating a no-flow condition. The pulse into actuator port 1 shows a clear step with a differential pressure of 180 lbf/in^2, though it will be noticed that the low pressure state is at a level of 70 lbf/in^2. The peak at the front edge of the pulse shows the time lag between actuators supplied by the same slot. The leading pulse is larger because no pressure can be leaked through the hydraulic stop. The pulse into the bottom of the knitting piston, i.e. at port 2, is also very sharp on the leading edge, with a drop in pressure when the second knitting piston is supplied with hydraulic fluid. As before, the lower state of the pulse is still at 70 lbf/in^2. On examining the pulses into ports 3 and 4, it will be noticed that the pulse remains in the high state after the port is cut off, indicating that leakage between the rotor and the cylinder wall is negligible, proving

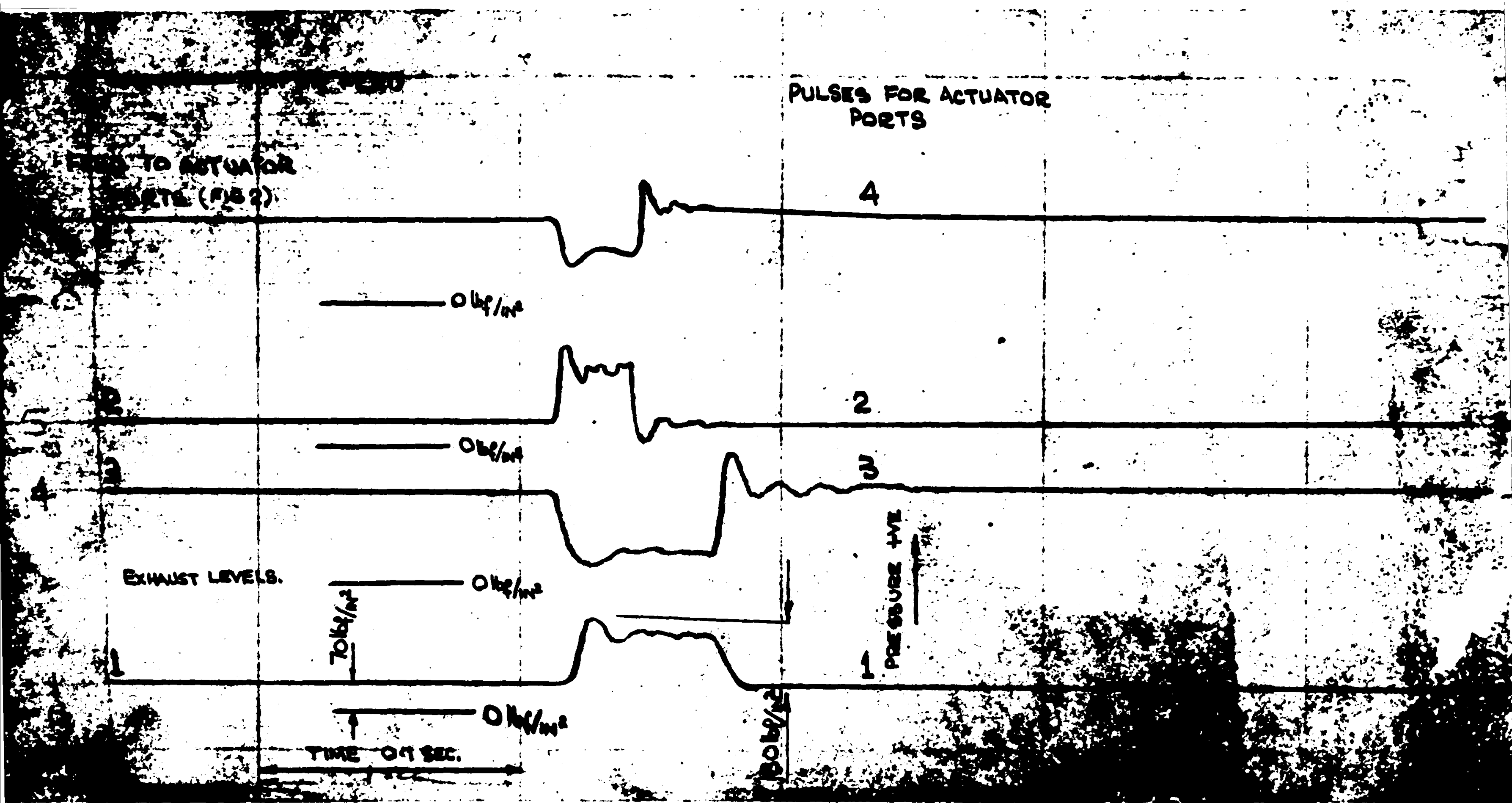
TABLE 2.

CALIBRATION OF PRESSURE TRANSDUCERS

Carrier System Channel	1	4	5	6
Pressure Transducer				
Range 0 - lbf/in^2	5000	3000	2000	2000
S.E. Lab. Number	56280	64188	58178	58739
Galvanometer				
Type	B450	B450	B450	B450
S.E. Lab. Number	7408	3809	3805	55548
Carrier System Settings				
Balance	5.42	5.71	5.2	4.7
Attenuation	0	0	9	9
Gain	5.6	8.4	6	6

Pressure lbf/in^2 applied to transducer	Distance from 0 lbf/in^2 on U.V. Trace. in.
0	0
100	.27
200	.55
300	.82
400	1.10
500	1.37
600	1.63
700	1.88

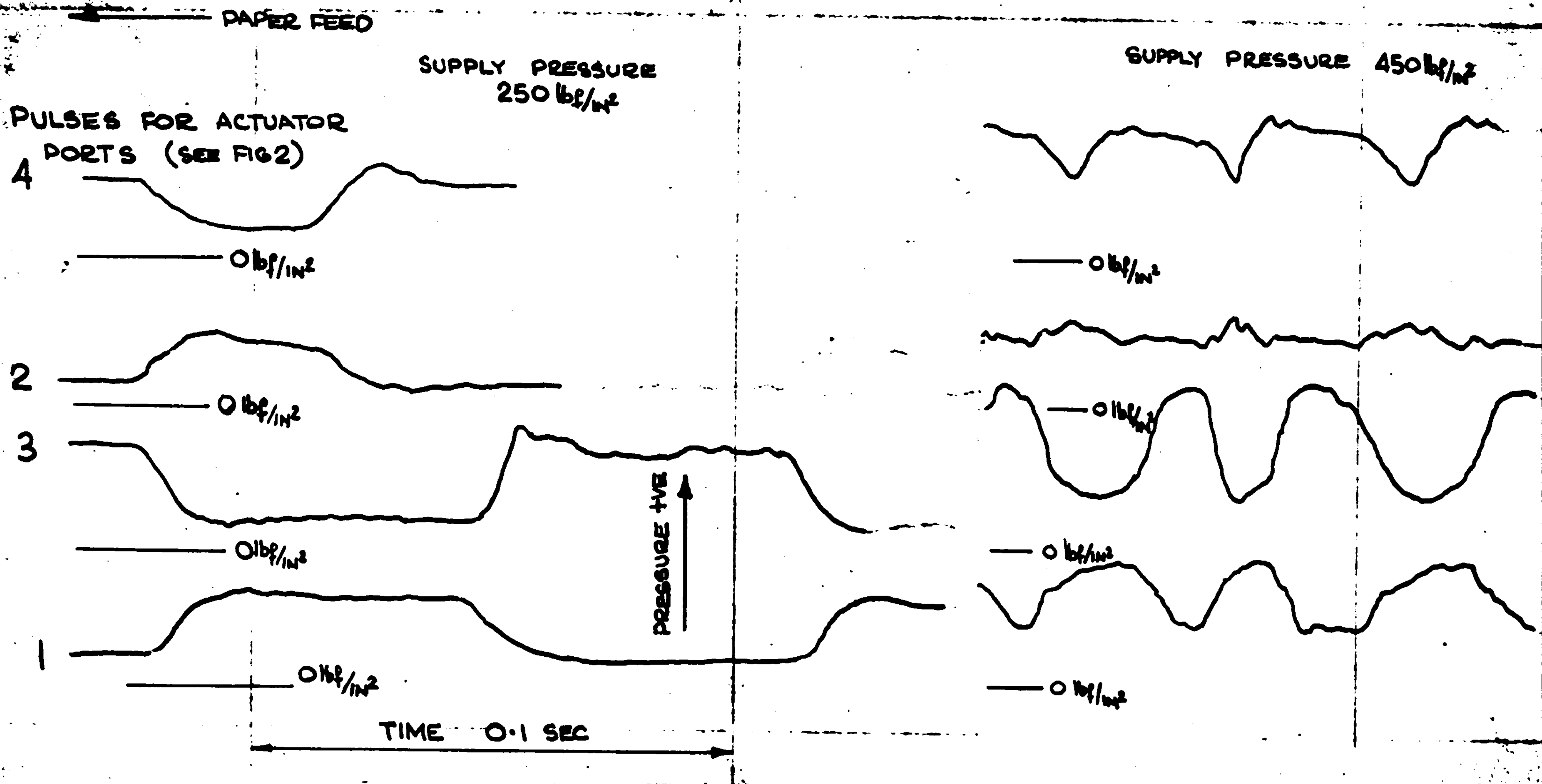
The above table gives a straight line graph of gradient 366 lbf/in^2 in.



51

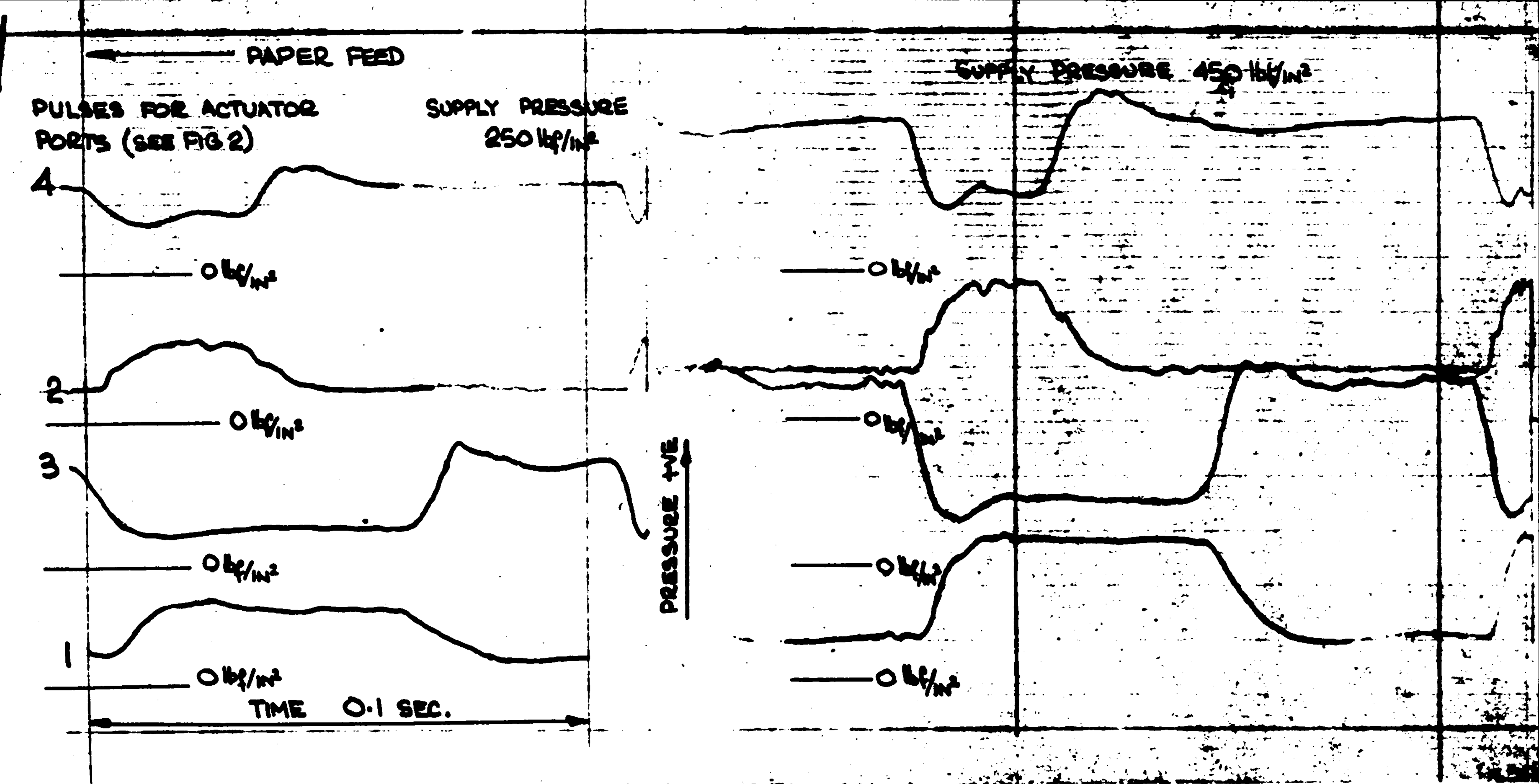
U.V. RECORDING OF THE PULSES DISTRIBUTED BY THE ROTARY VALVE SHOWING THE PULSES GENERATED AT THE CASING OF THE VALVE WITH A SUPPLY PRESSURE OF 250 lbf/in²

FIG. 21



U.V. RECORDING OF THE PULSES DISTRIBUTED BY THE ROTARY VALVE SHOWING THE PULSES FROM THE SEPARATE SUPPLY PATHS AS THEY APPEAR AT THE MANIFOLD BLOCK

FIG. 22



U.V. RECORDING OF THE PULSES DISTRIBUTED BY THE ROTARY VALVE SHOWING THE PULSES AS THEY APPEAR AT THE INLETS TO THE ACTUATOR

FIG. 23

that the bobbin must be running centrally within the cylinder and the machining tolerances required between rotor and cylinder satisfactory. Again it can be seen that the pulse steps from a pressure level of 70 lbf/in^2 to 250 lbf/in^2 which is a pressure differential of 180 lbf/in^2. Figure 22 shows the pressure pulses generated at the four points round the rotary valve as they are supplied to the manifold block. It will be noticed that all the pulses appear to have the same form, indicating that all the sectors on the rotor are functioning identically, and a method of using different supply paths to introduce a patterning facility is feasible. These traces also show that the variation in the length of pipe between the rotary valve and manifold block does not have a significant effect upon the size and shape of the pulse. It can also be seen that whilst the pressure differential of the pulse is well defined, the low pressure level is almost 100 lbf/in^2 in some instances. These results can be seen in Table 3 for 360° of valve rotation. Figure 23 shows the pressure pulses at the actuator; this trace is very similar to figure 22, the only significant point being that both traces were recorded at the same supply pressure of 250 lbf/in^2. The pulses at the actuator have been attenuated due to the pressure drop required to deliver the oil through the smaller diameter pipe. This series of tests showed that the rotary valve was capable of supplying sharp pressure pulses required by the actuators, and in the correct sequence.
The only feature that was not satisfactory was the back pressure of 80 lbf/in^2 in the exhaust line. This high exhaust pressure level indicated that the exhaust side of the rotary valve was

TABLE 3

TYPICAL VALUES FOR THE PRESSURE LEVELS FOR THE PULSES DISTRIBUTED BY THE ROTARY VALVE TO A SINGLE ACTUATOR PER REVOLUTION OF THE ROTARY VALVE, WITH A MAIN SUPPLY PRESSURE OF 250 lbf/in^2.

ANGLE OF VALVE ROTATION degrees.	PRESSURE LEVEL lbf/in^2			
	ACTUATOR PORT 1	ACTUATOR PORT 2	ACTUATOR PORT 3	ACTUATOR PORT 4
0	210	210	50	90
22.5	175	20	70	240
45	50	50	240	210
67.5	50	50	210	210
90	210	210	50	90
112.5	175	20	70	240
135	50	50	240	210
157.5	50	50	210	210
180	210	210	50	90
202.5	175	20	70	240
225	50	50	240	210
247.5	50	50	210	210
270	210	210	50	90
292.5	175	20	70	240
315	50	50	240	210
337.5	50	50	210	210

restricting flow of fluid back to the reservoir. This necessitated a higher supply pressure to obtain the pressure pulse differential required to operate the actuators.

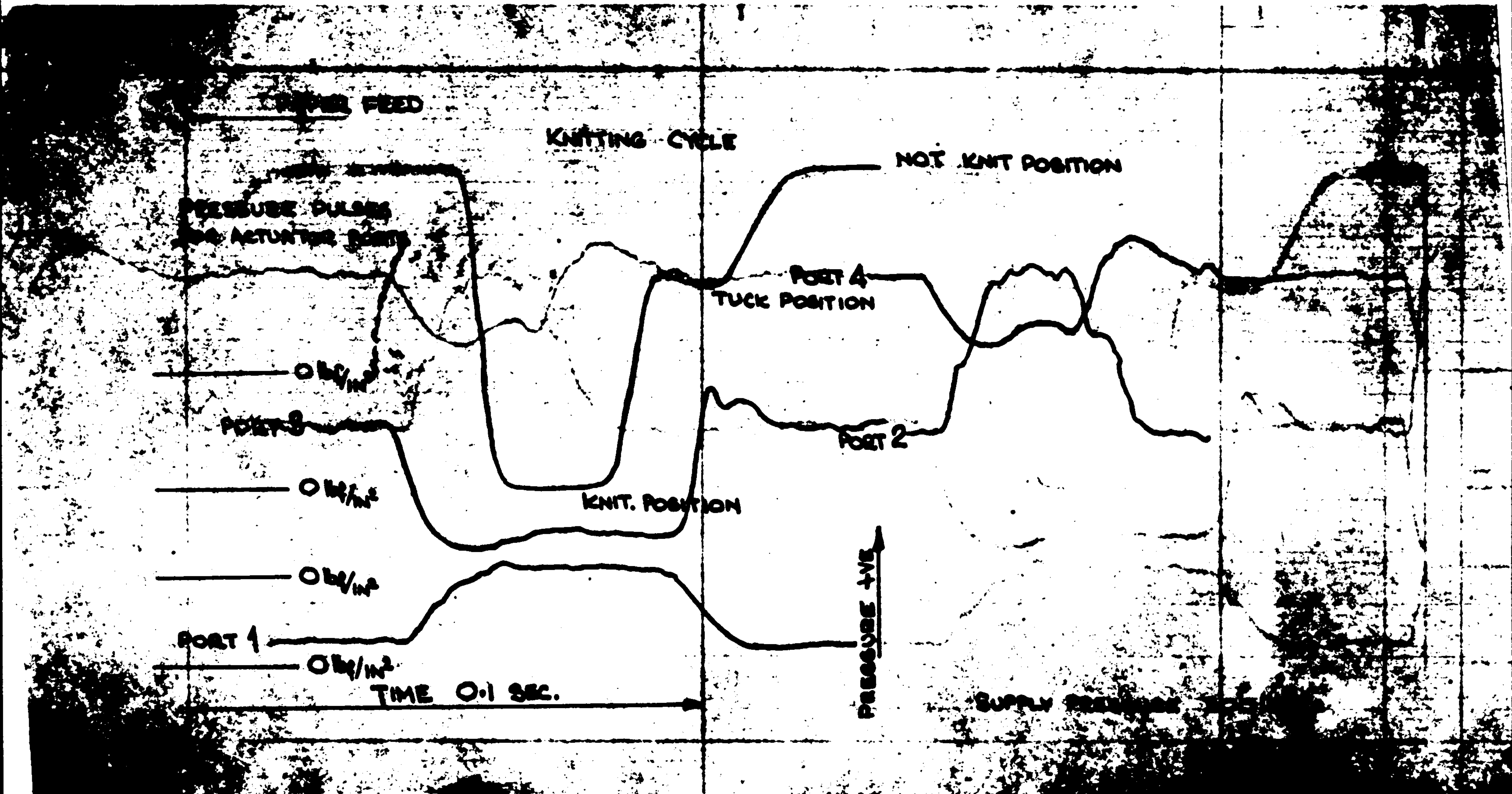
2.8.5. Modifications to the Rotary Valve.

This back pressure had to be reduced by introducing larger exhaust passages within the bobbin of the valve, and it was at this stage that the central exhaust hole was machined down the centre, and the oil exhausted through the rotor and the back plate of the valve. This can be seen in figure 24 which shows the components for the modified rotary valve. This central exhaust passage had the further advantage of simplifying the rotor design. This one large diameter passage replaced the twelve axial holes required previously. Another modification to the rotor at this stage was to extend the exhaust slot in sections AA, CC and EE to the form shown in figure 6 (section 2.7). Until this point when the actuator was returning from the tuck to the miss position the exhaust oil at port 2 had been taken via the one-way valve to the exhaust in port 1. However, the introduction of the longer exhaust slot at port 2 meant that for plain knitting, the one-way valve could be omitted.

The rig was finally reassembled in its modified form and a displacement transducer was placed over an auxilliary actuator so that the time displacement profile of the actuator could be checked against the required profile of figure 1 and compared with the results obtained in previous tests on individual actuators and rotary valves. The trace obtained from this test

FIG. 24 COMPONENTS FOR THE MODIFIED ROTARY VALVE

can be seen in figure 25. Here the time displacement profile for the actuator can be seen at 10 hertz, together with the pressure pulses at the manifold block. This figure shows that an actuator can be driven through the required time displacement profile to produce a knitted loop with a conventional latch needle. A demonstration of the multi-actuator block, operating at various speeds up to 40 hertz, can be seen in the first part of the film entitled "Development of a Hydraulic Knitting Machine". This film was taken as a permanent record of the prototype multi-actuator rig and shows the general layout of the system together with views of the actuators. The actuators were filmed at various operating speeds, showing the shape of the time displacement profile for each individual needle and how they operate in sequence. Even at cycling rates as high as 30 hertz, the general wave pattern does not alter. It is envisaged that the film will be viewed as a complementary item to this thesis because it illustrates the achievements and results of the project in a significant manner.



U.V. RECORDING OF PRESSURE PULSES GENERATED AT THE ACTUATOR BY A MODIFIED ROTARY VALVE, TOGETHER WITH A TRACE OF THE TIME DISPLACEMENT PROFILE FOR THE ACTUATOR.

FIG. 25

3. A HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

3.1. INTRODUCTION

3.1.1. Technical Decisions relating to the Development of a Hydraulically-actuated Circular Weft Knitting Machine

3.1.2. Basic Concept of the Machine

3.2. DESIGNING THE CIRCULAR WEFT KNITTING MACHINE

3.2.1. Specification for the Knitting Machine

3.2.2. Relating the Specification to a Prototype Knitting Machine

3.2.3. The Yarn Feed

3.2.4. Fabric Tensioning Mechanism

3.2.5. Detail design for the Actuator Block

3.2.6. The Knitting Trix

3.2.7. The Yarn Carrier

3.2.8. Detail design for Auxilliary Items

3.2.9. Assembly of the Hydraulic Knitting Machine

3.2.10. Piping the Hydraulic Circuits

3.2.11. Initial tests on the Hydraulic Actuators

3.2.12. Introduction of the Knitting Mechanism

3.3. TESTING THE CIRCULAR WEFT KNITTING MACHINE

3.3.1. Initial attempts to produce a Knitted Fabric

3.3.2. Modifications to the Yarn Carrier

3.3.3. Film of the Hydraulic Knitting Machine

3.3.4. Description of the Completed Machine

3.3.5. Comments on Machine Performance

3.4. THE HYDRAULIC POWER PACK

3.5. FINDINGS AND RECOMMENDATIONS FOR THE APPLICATION OF HYDRAULIC ACTUATION TECHNIQUES TO KNITTING MACHINES

3.5.1. Future areas of Research

3.5.2. Further practical usage of Hydraulic Actuation Techniques as applied to Knitting Machines.

3.0 A HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

3.1. Introduction

3.1.1. Technical Decisions Relating to the Development of a Hydraulically actuated Circular Weft Knitting Machine

The building of the Multi-actuator rig completed a stage of the project as initially defined in section 2.7. The Department of Mechanical Engineering had built a block of twenty four actuators suitable for a knitting machine. These actuators operated in sequence and possessed a limited patterning facility. A technical demonstration and discussion was held at the University between Mr. K. A. Hartley Mr. W. Betts, Mr. Maidens, Mr. F. Carrotte, Mr. T. Priestley and the author, to assess the potential of the project and attempt to define its commercial application. It had always been agreed that the knitting aspect of the project would be undertaken by knitting experts within the Courtauld's Group, hence the specification of a sixteen gauge block of actuators. However, once confronted with the hardware that eliminated the use of cams on a weft knitting machine, confusion arose as to how it should be applied. A Courtaulds decision was not forthcoming and finally put the onus onto the Department to make the decision as to the next stage of the project. At a technical discussion between Mr. T. Priestley and the author, a decision was made to answer the only outstanding question:- "Can a hydraulic actuator system be made to knit?". The task of producing a hydraulic knitting machine had been declined by Courtaulds which left the Department to build a Hydraulic Circular Weft Knitting Machine, and prove the system.

3.1.2. Basic Concept of the Machine

Up to this stage in the project the technological involvement with knitting had been very limited, because it had been understood that Courtauld's would fulfil the knitting aspects. However, this situation had changed, the Department now deciding to produce a prototype hydraulic knitting machine. A general survey of knitting machines was made, which included informal discussions with Mr. F. Carrotte, Technical Director of Kirkland Engineering Ltd, and Mr. H. Wignall, Head of the Textile Department at Leicester Polytechnic. It was evident that hydraulic actuators could not be applied directly to a sophisticated production machine, and it was suggested that a hand-operated hose machine might form the basis for the initial design. These hand machines have been used by hose manufacturers since the early 1900's and while basic in form and construction, operate on the same principle as present day machines. A Griswold hand machine was donated by Mr. F. Carrotte to the Department. This was a single feeder, six gauge $4\frac{1}{2}$ inch diameter weft knitting machine. The needles were located round a circular trix and held in position with a split collar. Round the base of the trix ran a single cam mechanism which moved 25% of the needles in sequence through the knitting profile. The yarn carrier had to be rotated in phase with the knitting cam, because using this configuration, the needles were held stationary. The yarn carrier was attached to the cam mechanism and being a single feeder machine permitted the creel to remain stationary. The yarn was fed to the carrier from a feeder located on the central axis of the machine, twelve inches above the needles. The take-down mechanism of the Griswold was a fabric clamp onto which weights could be attached. This simple method of tensioning the fabric gave an even pull. The machine was hand operated driving a bevel gear mechanism to the cam and yarn carrier. Three basic adjustments could be made to the machine:-

(i) Yarn tension, by means of a pivoted counter-balance weight or light spring on the creel.

(ii) Fabric tension, by applying more weights onto the fabric via the clamp.

(iii) The stitch length, by moving the stitch cam relative to the trix. This adjustment could be made externally to the cam machine via a graduated lever on the body of the machine.

The yarn and fabric tensioners were found to be flexible provided that the yarn strength was not exceeded. The yarn tensioner was basically to prevent uncontrolled vibration modes being set up in the yarn as it unwound from the bobbin. The fabric tensioner produced a restraint on the needle latches, consequently the tension had to be sufficient to hold the latches open but to not impart a high frictional resistance to needle motion. It was found that the fabric tensioner could also be used to vary the knitting density i.e. applying more tension increased the length of the knitted loop so producing a slacker fabric. The stitch length needed critical adjustment and even a slight movement of the cam altered the quality of the knitted fabric. To start the machine knitting, a piece of knitted fabric had to be hooked over the needles, and by using a long stitch length the new yarn was introduced via the carrier. The knitting could be improved by opening the non-functioning latches and removing loops from needles operating in pairs. Within four or five revolutions of the machine, all the needles could be made to knit, and the cam controlling the stitch length could be adjusted in conjunction with the fabric tension to produce the desired quality of plain knitting. This machine indicated the problems associated with knitting and highlighted the areas requiring special attention when designing a knitting machine. This machine was examined in detail and considerable time was spent becoming competent in basic knitting techniques.

The configuration of the Griswold machine was suitable for applying the concepts derived from the hydraulic multi-actuator system to an actual knitting machine:-

(iv) The needles did not move radially; thus the actuators could be machined into a block and fixed to the frame of the prototype machine.

(v) The creel arrangement was simplified by using a single feeder.

(vi) The knitted fabric leaves the machine as a cylindrical tube, thus requiring only a simple tensioning mechanism.

(vii) The circular configuration of the needles permits the yarn carrier to be driven by, and in phase with, the rotary valve.

(viii) The rotary valve developed on the multi-actuator block could be used without any modifications.

(ix) The actuators would be applied to a circular weft knitting machine which is how they were intended to be used at the outset of the project.

3.2. Designing the Circular Weft Knitting Machine

3.2.1. Specification for the Knitting Machine.

Prior to any design work on the knitting machine, a specification for the type of machine had to be formulated. Using the experience gained in designing the multi-actuator block, and using the Griswold knitting machine, a specification for the prototype hydraulic knitting machine was derived and stated as:-

(i) A circular configuration using a coarse gauge.

(ii) A single feeder.

(iii) No patterning facility; all needles producing plain knitting.

(iv) Standard latch needles.

Before embarking upon the design, the components of the multi-actuator block were evaluated with a view to using them in the prototype knitting machine. Various components such as the actuator and manifold blocks could not be employed but the basic chassis, variable speed gear box and rotary valve could be used. As explained in section 2.7. the rotary valve was used for sequence control and also for introducing a patterning facility of up to four selections, thus giving a total of ninety-six actuator outlets. If all these outlets were connected to an individual actuator, then the system would operate in a similar manner to the Griswold machine with only 25% of the actuators operating at any particular instance. Using the rotary valve with no patterning facility would provide a system for driving ninety six actuators in sequence. The needle gauge then had to be selected. Using the actuator form already tested and considering the manufacturing problems involved in high density actuator packaging, the application problems were simplified by machining all actuators to a common depth on the same pitch circle diameter. The piston diameter for the actuator was .125 inches and experience with the multi-actuator block indicated that for relative ease of packaging, the actuators should not be pitched closer than .25 inches. In order to produce a machine capable of knitting, this .25 inch pitch allowed tolerances on the machining limits, and allowed standard pipe fittings to be used in the actuator block.

With due reference to these facts, the gauge of the machine was specified as 4, with ninety six needles on a 7.625 inch diameter pitch circle.

An aim of the design was to utilise standard latch needles of the correct gauge. The four needles selected were two plate needles type F.264 and F.327 and two wire needles type H.9 and F.13. The wire needle type F.13 (complete specification GROZ-BECKERT Aha 78.130 Gl) was selected because it had the longest straight section after the latch, thus making it more adaptable.

3.2.2. Relating the Specification to a Prototype Knitting Machine.

The needle characteristics were studied to assess the relative displacements required to form a stitch. Each needle works like a crochet hook. The cycle begins with a stitch in the hook; the needle is then pushed until the stitch passes behind the latch on the body of the needle. The needle is then returned to the tuck position, with the stitch still behind the latch, and new yarn is fed into the hook. The needle is returned to the miss position whence the latch is closed by the original stitch as it is dropped from the needle. To form the stitch, the needle required a minimum of .875 inches of movement, consequently using the same actuator displacements as had been used throughout the project, (that being a relative movement of one inch for the knit position and half an inch for the tuck movement) would form the stitch and allow flexibility in determining the stitch length.

Therefore, extending the specification, the machine conforms to:-

(v) A circular configuration with ninety six needles spaced at .25 inch intervals on a 7.625 inch diameter pitch circle.

(vi) Twenty four needles to be in operation at any one time; six in the extended knit position, six in the tuck position, and the remaining twelve held in the miss position.

(vii) Four gauge needles type F.13 to be used.

(viii) Use to be made of the existing rotary valve as a sequencing device.

(ix) The dimensions for the miniature hydraulic actuators to be identical to those in the multi-actuator block.

With the above parameters determined, the two outstanding factors to be considered before the actual design could be evolved was the yarn carrier and the fabric tensioning mechanism.

3.2.3. The Yarn Feed.

The yarn carrier had to revolve in phase with the needles in knitting profile and since the rotary valve sequenced the needles, the obvious solution was to take a drive directly from the rotary valve shaft. This ensured that the carrier would remain in phase with the needles and any increase in rotor speed would automatically be transferred to the yarn carrier. This carrier had to be placed above the actuator feed pipes thus making the top of the actuator block a bearing face. The only outstanding problem was how to transmit the rotary valve drive from a vertical plane to a horizontal plane round the needles. Several methods were considered, from a complete gear train to an electrical stepping motor. When actual hardware was considered, the most convenient method seemed to be two stepped belts and a small right angled gear box. However, when considering stepped timing belts no reason could be found for not attempting to twist the belt drive through 90° by using two auxilliary idler gears, so eliminating the bevel gear box. Modern timing belts have nylon bracing so should not fatigue under the twisting action. Using this drive mechanism involved mounting a stepped pulley on the rotary valve shaft, and a similar sized pulley to revolve on the top of the actuator block, so keeping the 1:1 speed ratio. Two idler pulleys had to be mounted at the intersection of the horizontal and vertical planes of the two major pulleys, following the established flat belt techniques. The use of the stepped timing belt would ensure the phase relationship between the rotary valve and the yarn carrier, regardless of the rotational speed of the valve.

3.2.4. Fabric Tensioning Mechanism.

The only fact still to be considered was the fabric tensioning mechanism. The simplest tensioning device would have been a series of pulleys and a weight hung onto the fabric, but this technique limits the quantity of fabric that can be knitted before weight adjustment. A more satisfactory device would be a pair of adjustable rollers to pull at a predetermined tension. Various slipping clutch mechanisms were considered, both mechanical and electro-magnetic, but these were not suitable from a torque rating and consistency aspect. An alternative mechanical method would be to turn the rollers in phase with the yarn carrier, through a variable ratio gear box that could be adjusted to suit the required fabric being knitted. When this method was examined it was evident that the idler gears were in phase with the yarn carrier so could be used as a drive for the fabric tensioning gear box. Numerous small variable ratio gear boxes were available but one, the Zero-Max, was the most suitable because the reduction ratio could be adjusted from 4 : 1 through the range to infinity. The drive from the idler gear to the gear box could be made with a vee belt because the timing did not warrant the sophistication of a stepped belt. These concepts were investigated by sketching the various possibilities on large sheets of paper, until a feasible orientation had been evolved. These ideas were roughly formulated; taking into account the individual limitations imposed by the various components. Having arrived at a suitable sketched design, the components were drawn to scale in relationship to each other to form layout drawings. These provided a basis for fixing component size thus ensuring that the basic elements could be assembled when manufactured.

3.2.5. Detail Design for the Actuator Block.

The first detail drawing to be made was that of the actuator block. (see figures 26 and 27). Figure 26 (drawing no. J.D.G. 042) shows the overall dimensions of the actuator block which was to be manufactured in Meehanite Cast Iron. The annulus shown in figure 26 had to be machined from solid as this was more convenient and time-saving than manufacturing a pattern for an individual casting. The solid Meehanite rod was centrifugally cast which helped to minimise the possibility of blow holes. Figure 27 (drawing no. J.D.G.043) shows the actuator dimensions; the actuators are basically the same as those developed in the first research project, with refinements being made to the sealing of the knitting piston rod and to the positioning of the supply ports. When the multi-actuator rig


LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
DRAWN
TRACED
GROUP

FIG. 26

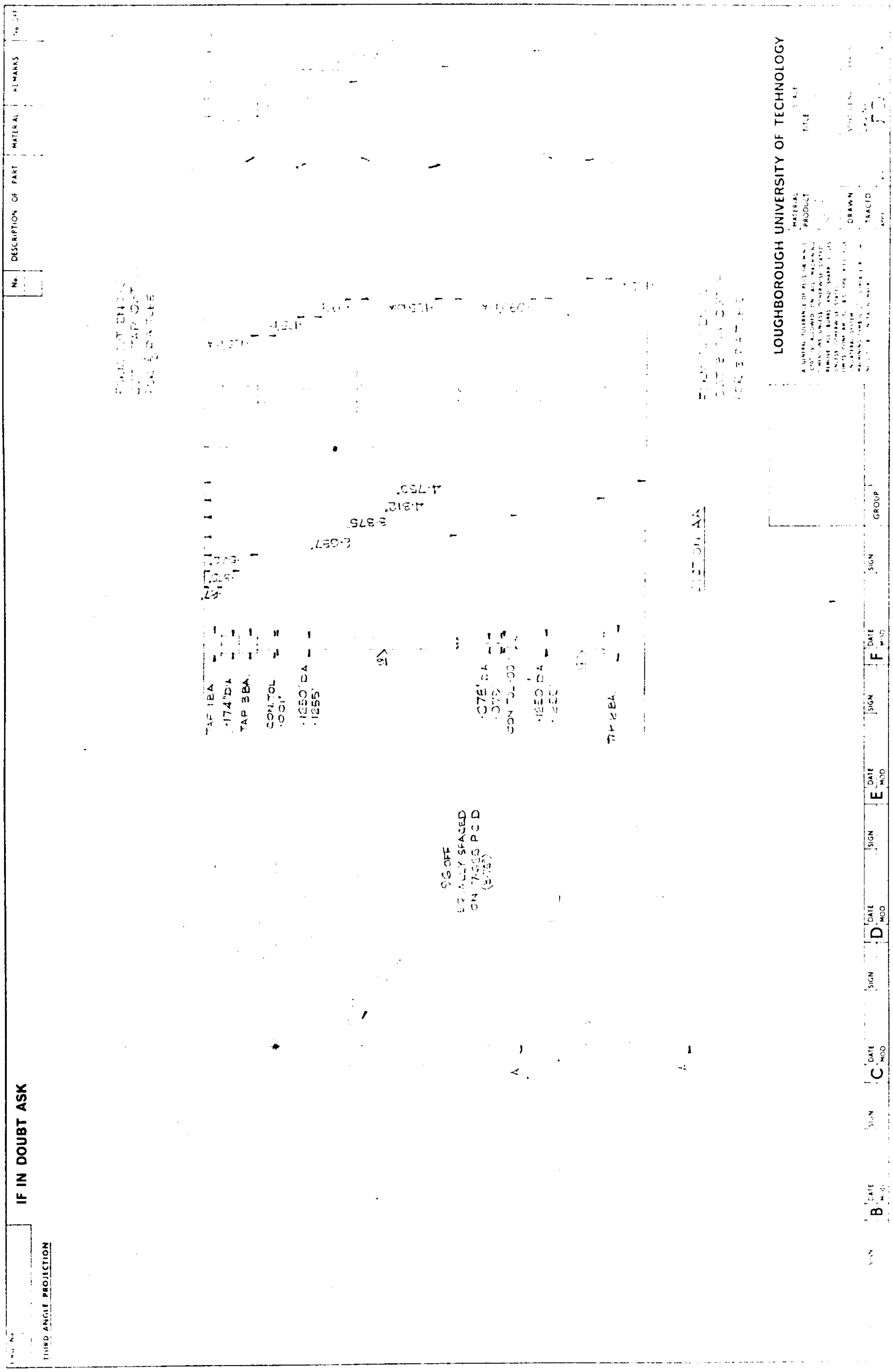

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
TAP 8BA
CON.TOL
EQUALLY SPACED

FIG.27

was dismantled, the actuator block was examined for signs of design weakness and it was found that the 'O' rings were being sheared by the continual impact of the knitting piston. To prevent this impact being transmitted to the mechanical seal, a small grub screw was introduced to absorb the shock. The 'O' ring protected by two small washers, was retained by a gland nut which was screwed down to make contact with the face of the actuator block. The actuator supply ports had long drilled passages enabling conventional pipe fittings to be used. It was the spacing required for the .187 inch diameter pipe fittings that determined the overall diameter of the actuator block. (The pipe fittings used were Enots standard fittings for .187 inches diameter tube, with the hexagon machined away and replaced by two small flats.) The ports for the hydraulic stops were machined to take the .125 inch diameter tube fittings in order that the restrictors could be attached directly to the block. The step at the bottom of the actuator block was to provide clearance for the bottom sealing screws; the top recess acted as a bearing for the stepped tooth pulley used to drive the yarn carrier. The central hole in the actuator block allowed the knitted fabric to pass through and acted as a knitting trix locator which had to be suspended in such a manner that it could be moved relative to the actuators. Figure 28 is a photograph of the actuator block viewed from the top and figure 29 shows the reverse side.

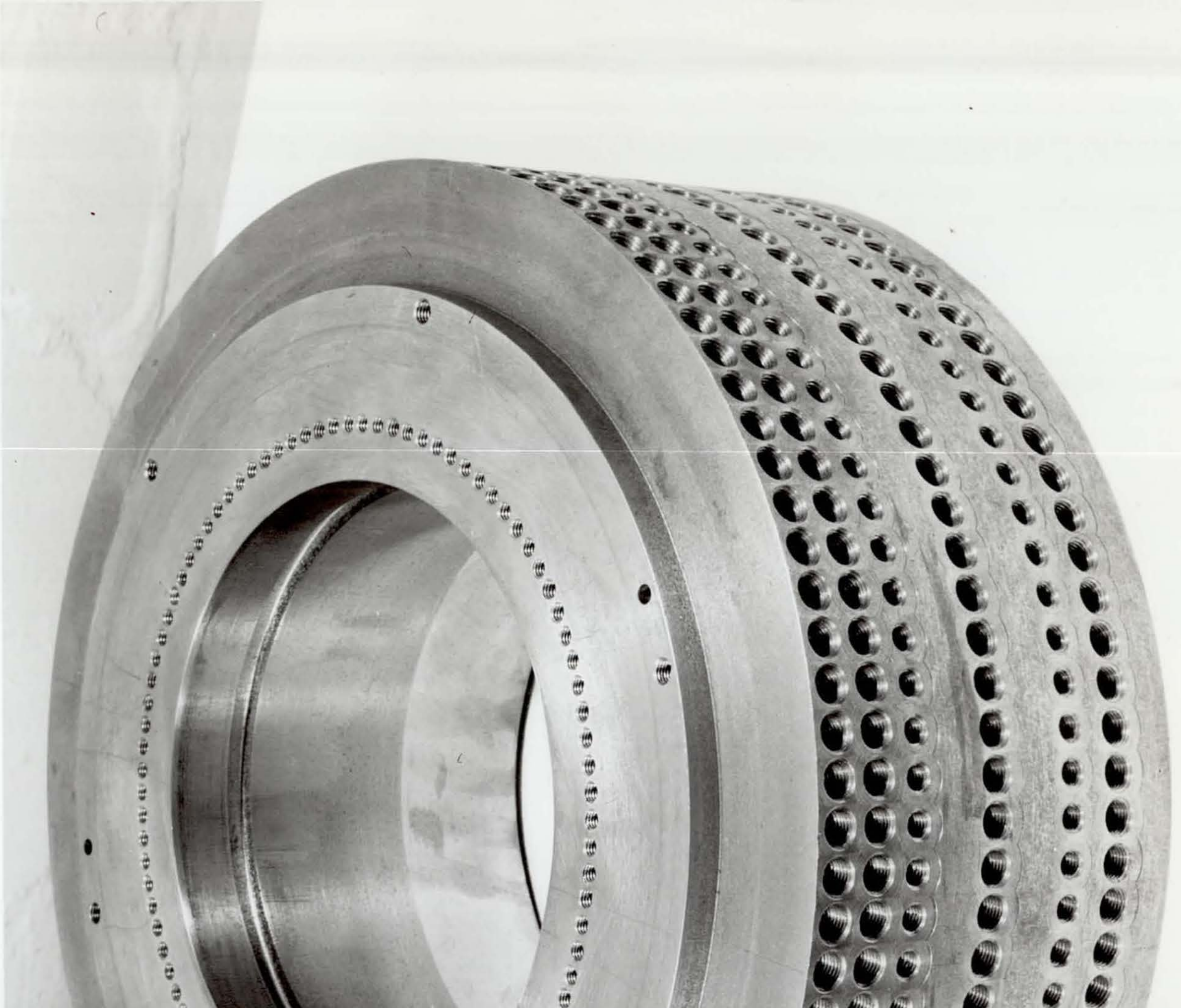


FIG. 28 ACTUATOR BLOCK VIEWED FROM THE TOP

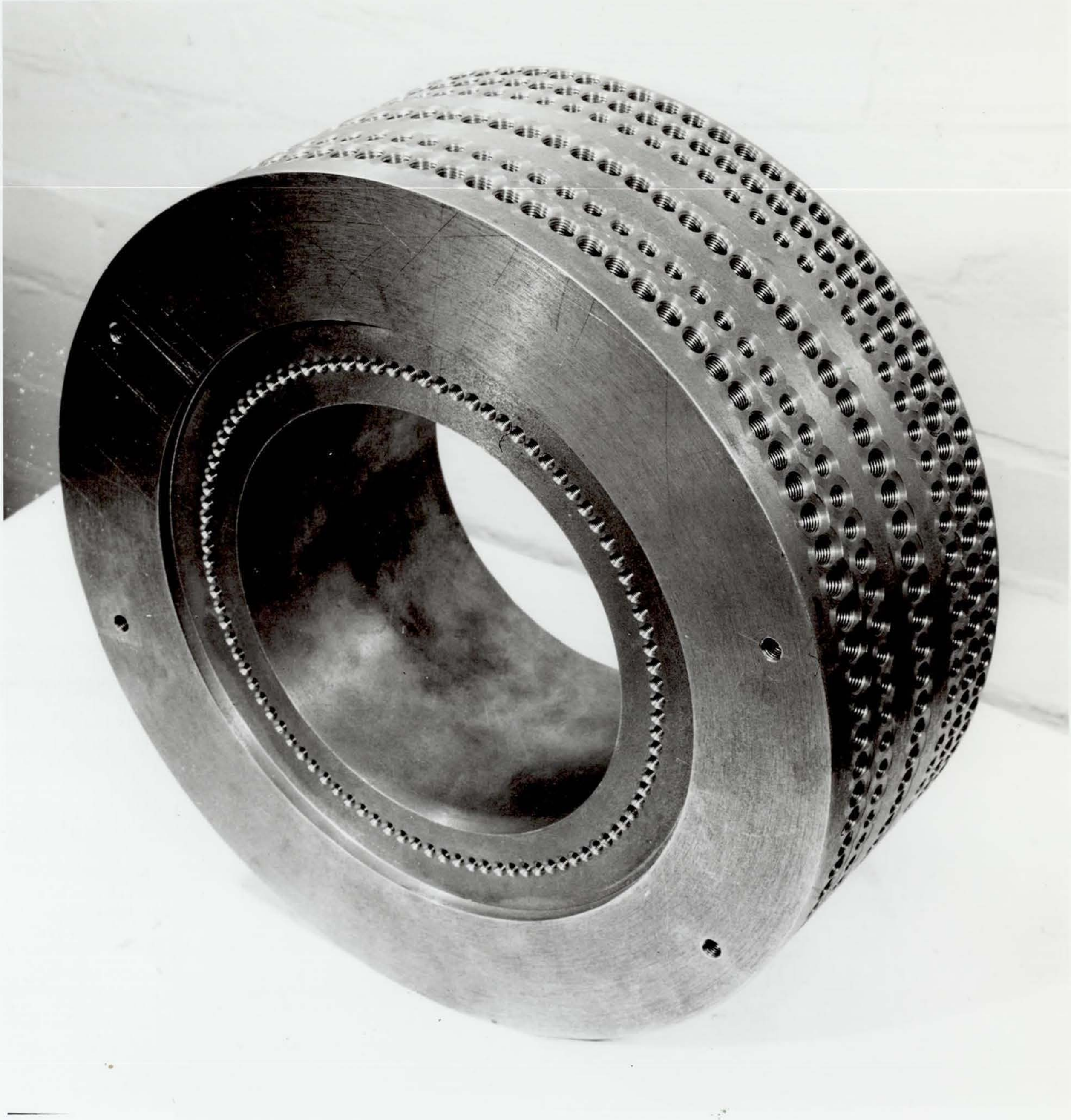
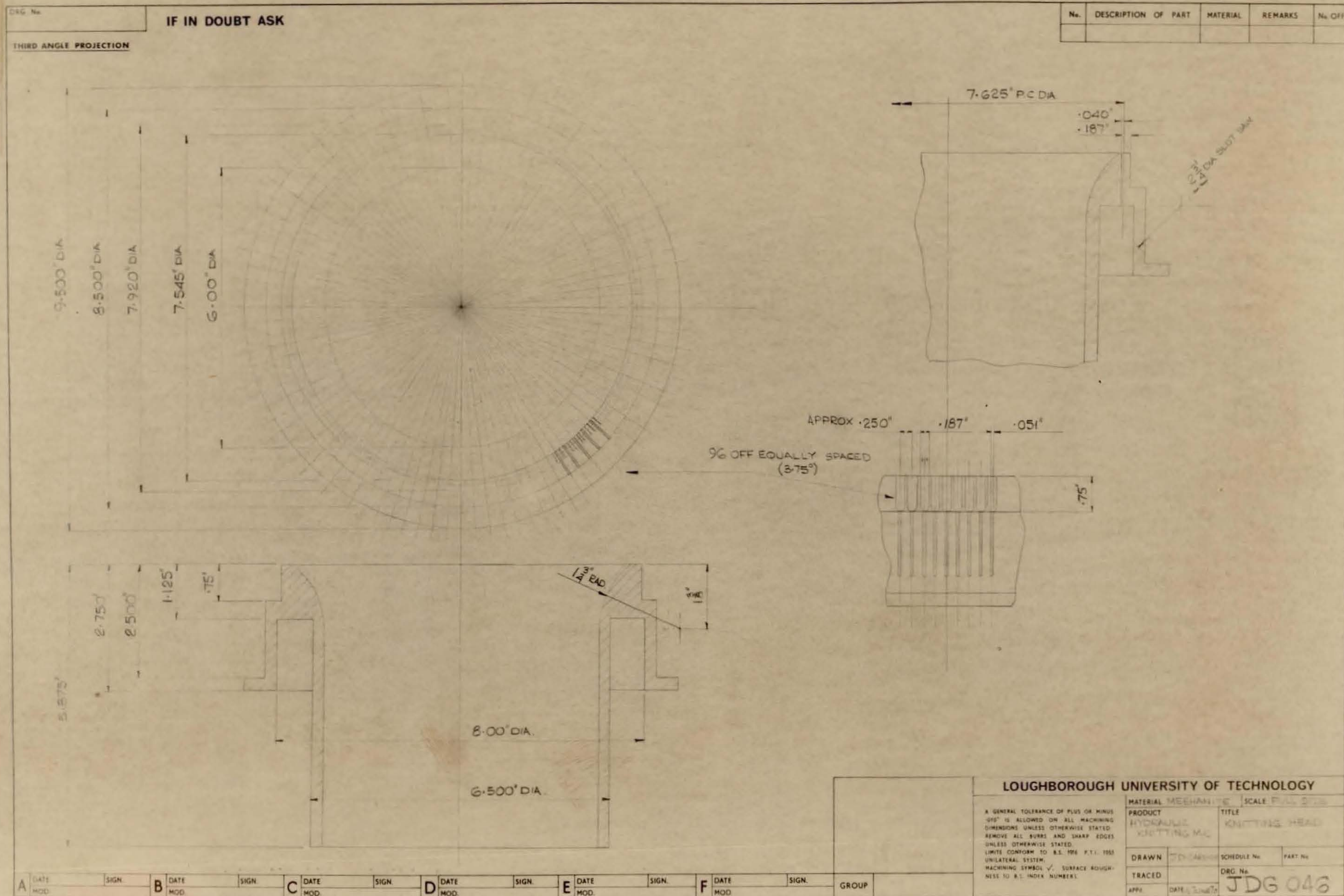


FIG. 29 ACTUATOR BLOCK VIEWED FROM THE BASE

3.2.6. The Knitting Trix.

The complementary component to the actuator block was the knitting trix. The form of the trix was modelled on the Griswold machine, and since the needles had already been selected, the section required to form the knitted stitch was a straightforward design exercise. A method of locating and fixing the trix relative to the actuator block created problems. The slots in the trix had to correspond to the positions of the actuator while allowing movement of the trix relative to the actuators in a vertical plane. When producing fabric, the centre of the trix would be inaccessible so any axial adjustment had to be made externally. The method finally adopted was to extend the edge of the trix radially and use three dowel posts in the actuator block to locate and constrain the trix while three screwed rods provided adjustment relative to the actuator block. The detailed design for the knitting trix can be seen in figure 30 (drawing no. J.D.G. 046). Again the material specified was Meehanite Cast Iron. The needle slots were cut with a slitting saw and the knitting trix formed using an end-mill. Figure 31 shows the completed trix, the central body being a sliding fit in the actuator block. Figure 32 shows the trix as viewed from the underside. A recess was machined to allow a coupling joining the actuator and needle, to be fitted without fouling the trix. These figures illustrate the complexity of the machining required to produce the trix mechanism. The remaining components required for the actuator block and knitting trix can be seen in figures 33 and 34. Figure 33 (drawing no. J.D.G. 044)



DRG. No.
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. | DESCRIPTION OF PART | MATERIAL | REMARKS | No. OFF
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8·500" DIA
7·920" DIA
7·545" DIA
6·00" DIA
7·625" P.C DIA
·040"
·187"
APPROX ·250"
·187"
·051"
96 OFF EQUALLY SPACED (3·75°)
·75"
2·750"
2·500"
1·125"
·75"
5·875"
8·00" DIA.
6·500" DIA.
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL MEEHANITE
SCALE FULL SIZE
PRODUCT HYDRAULIC KNITTING M/C
TITLE KNITTING HEAD
DRAWN
TRACED
DRG. No. JDG 046
GROUP

FIG. 30

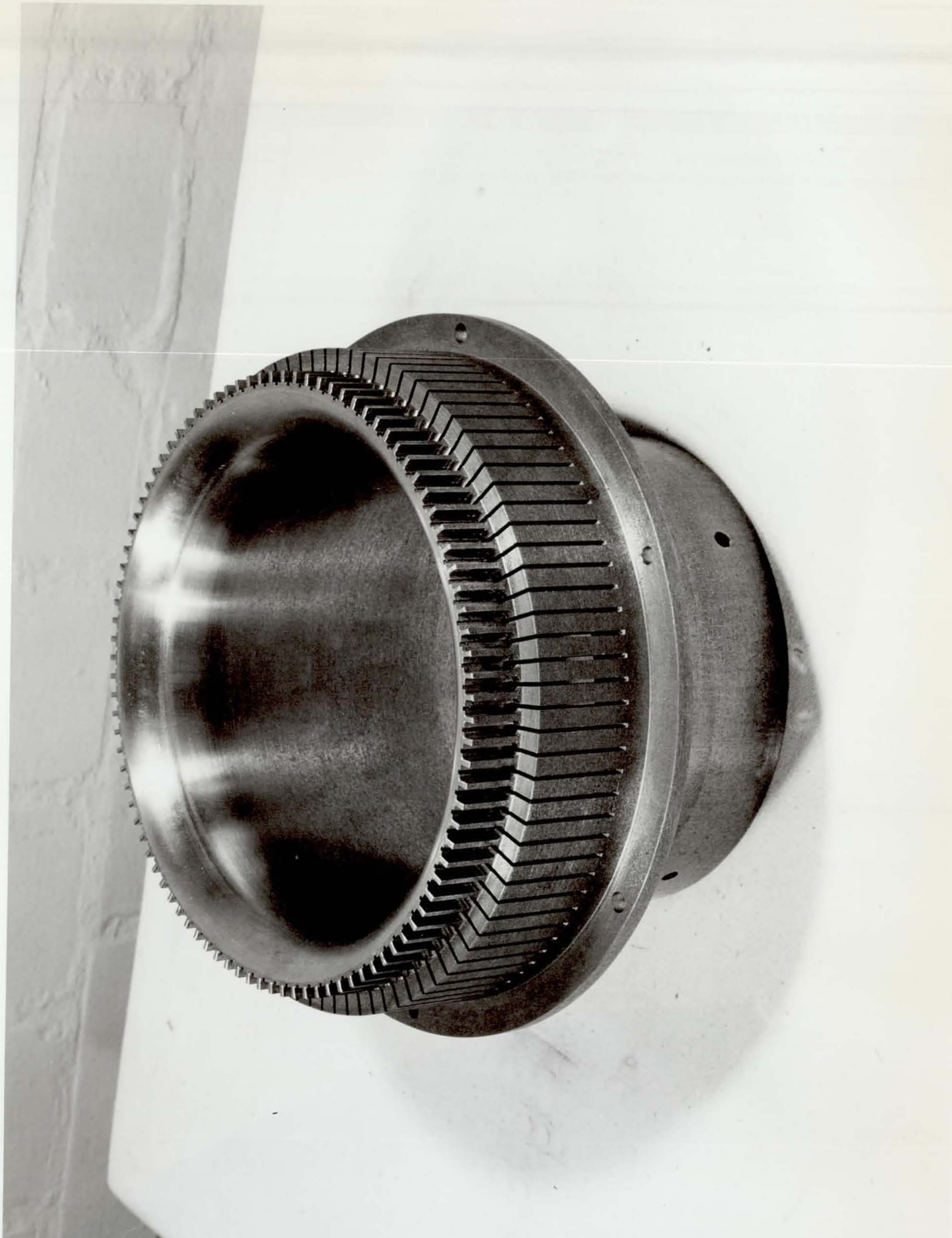


FIG. 31 KNITTING TRIX

FIG. 32 UNDERSIDE OF THE KNITTING TRIX

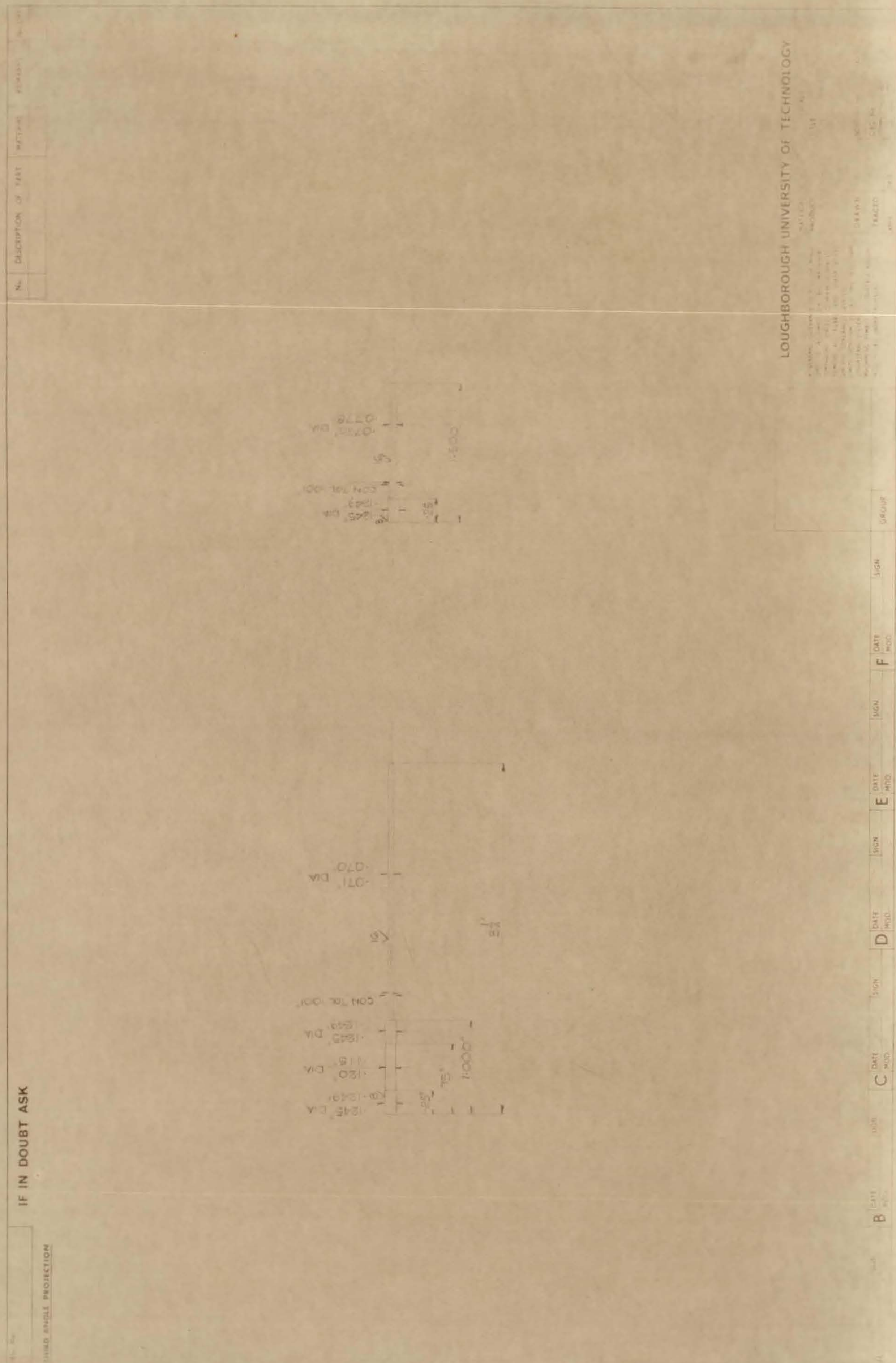

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY

FIG. 33

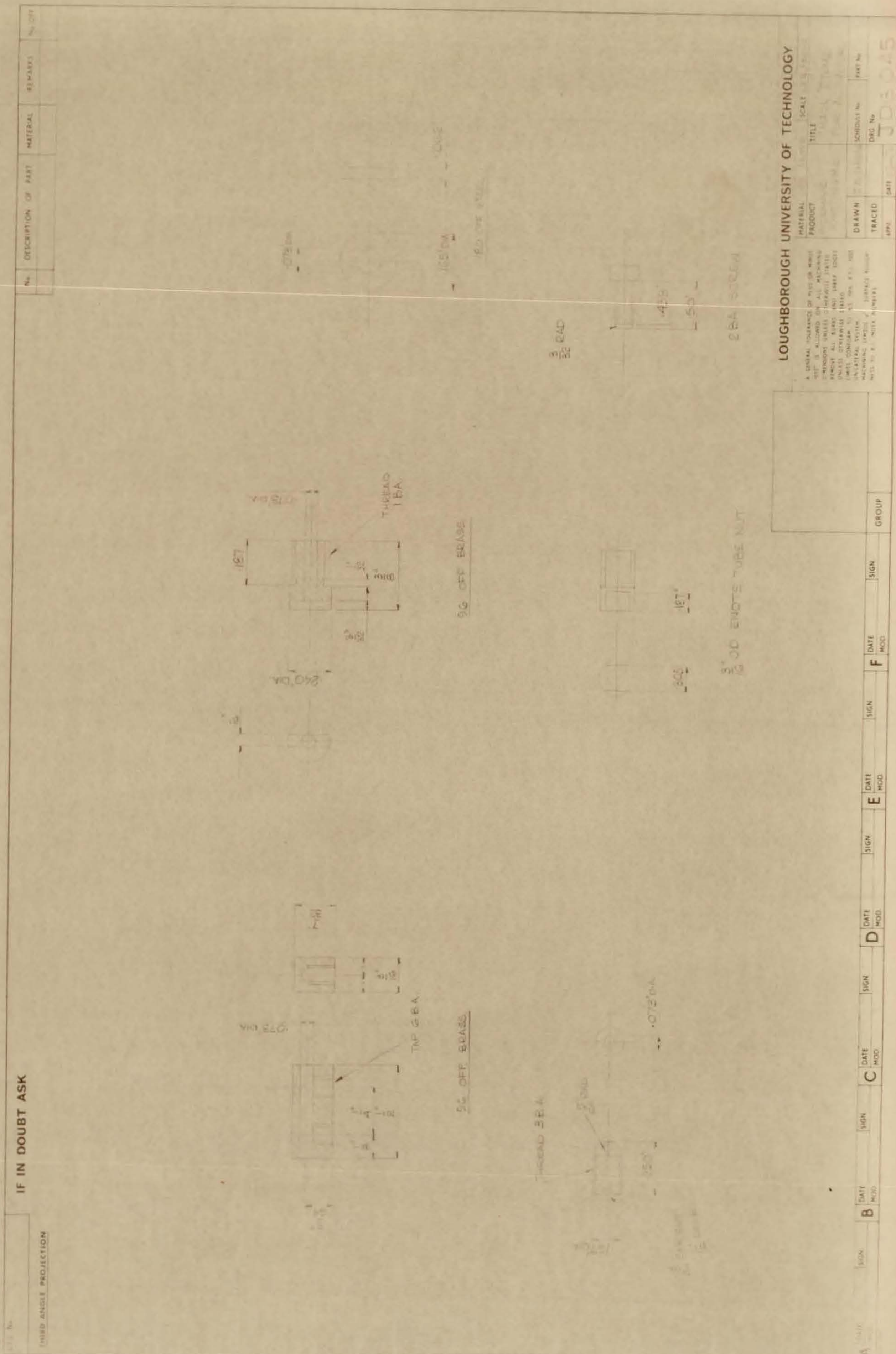

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
96 OFF BRASS
THREAD 1 BA
TAP 6 BA
DRAWN
TRACED
GROUP

FIG. 34

shows the knitting piston and tuck probe dimensions. These two components were manufactured in Kelock 795 outside the University. The only modification to the knitting piston was the centre section. This was relieved to help centralise the piston and reduce the hydraulic lock found on earlier pistons. The drawing J.D.G. 045 (figure 34) contains all the small components required for effecting the seal on the actuator and also the linking member for fixing the needles to the actuator.

3.2.7. The Yarn Carrier.

The components for the yarn carrier are shown in figure 35 on drawing J.D.G. 048. This particular yarn carrier was again based on the Griswold machine. The components were designed such that it could be adjusted vertically and horizontally to suit the knitting trix. Figure 36 drawing J.D.G. 047 shows the retai ning ring for the stepped pulley. In figure 37 the complete assembly of the timing belt pulley and yarn carrier can be seen mounted onto the actuator block. The stepped pulley was manufactured by purchasing a standard timing pulley and machining away the centre.

3.2.8. Detail Design for Auxilliary Items.

The remainder of the design work consisted of auxilliary items. The basic layout was to be similar to the mulit-actuator block therefore the actuator block would be positioned over the rotary valve. Using this configuration the knitted fabric would have to pass through the actuator block and then

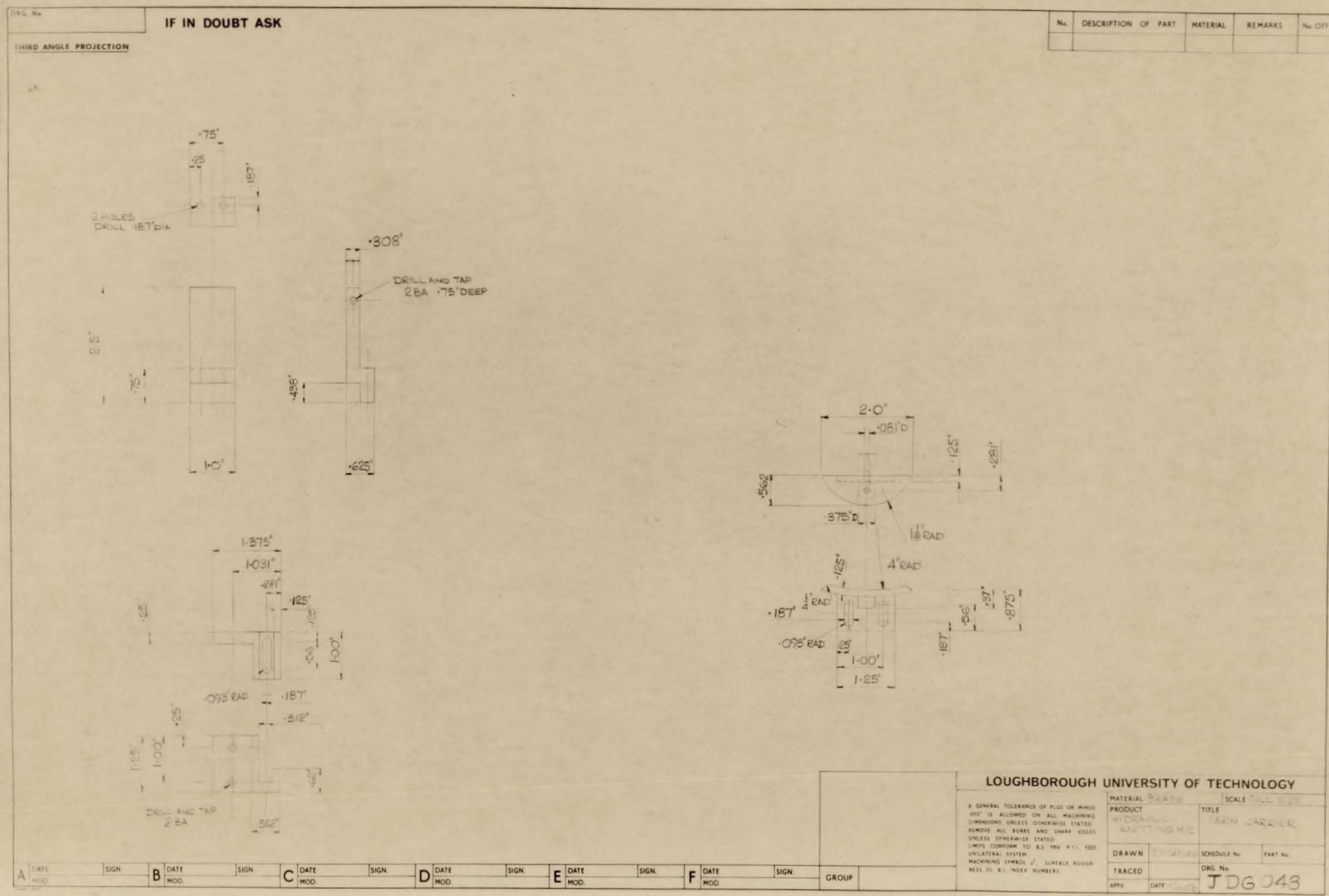
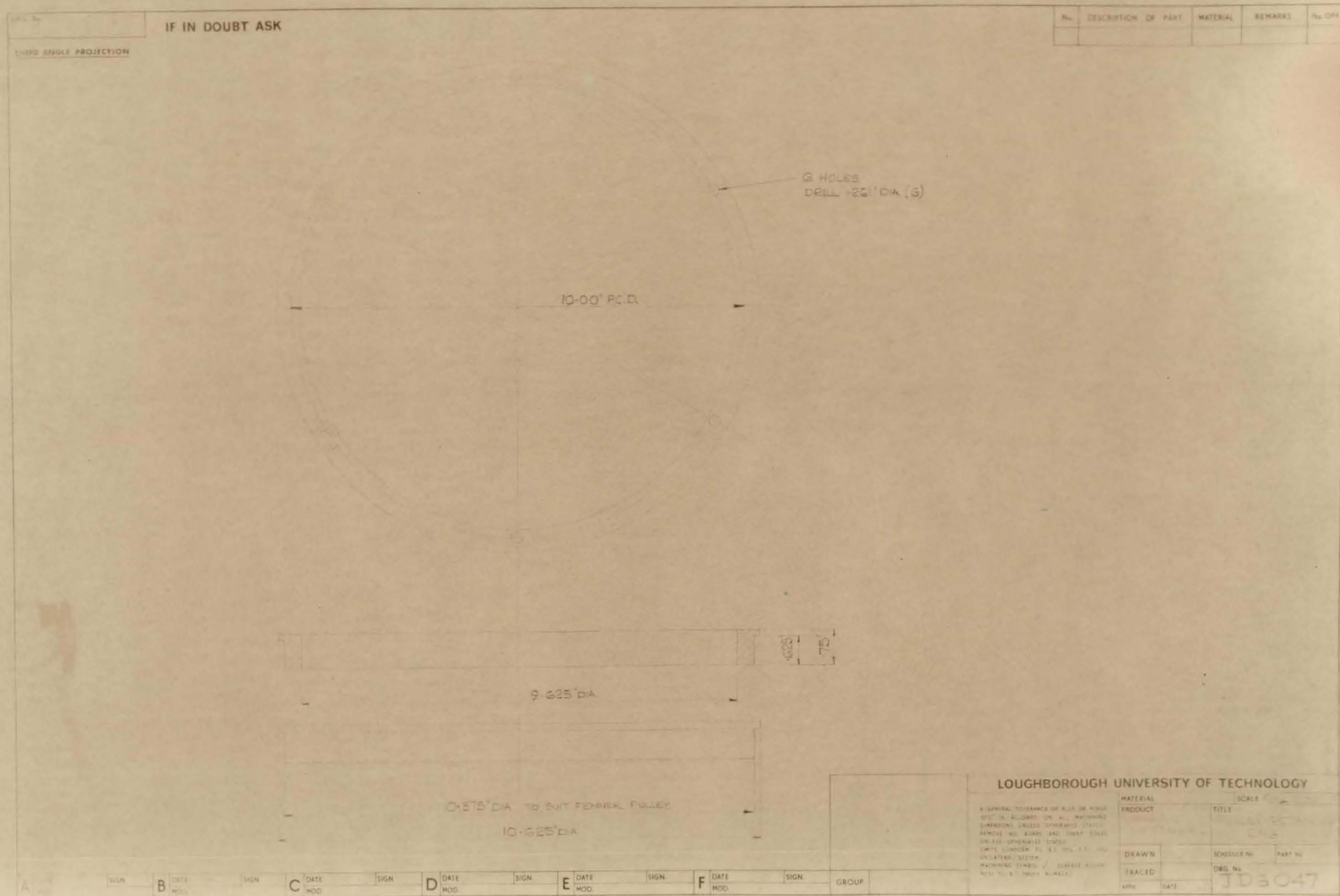

DRG. No
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. | DESCRIPTION OF PART | MATERIAL | REMARKS | No. OFF
2 HOLES DRILL ·187"DIA
DRILL AND TAP 2BA ·75"DEEP
DRILL AND TAP 2 BA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
SCHEDULE No.
PART No.
DRG. No
GROUP

FIG. 35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THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
No. OFF
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10·00" P.C.D.
9·625" DIA
0·875" DIA TO SUIT FENNER PULLEY
10·625" DIA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
SCHEDULE No.
PART No.
DRG. No.
GROUP
SIGN
DATE
MOD.

FIG. 36

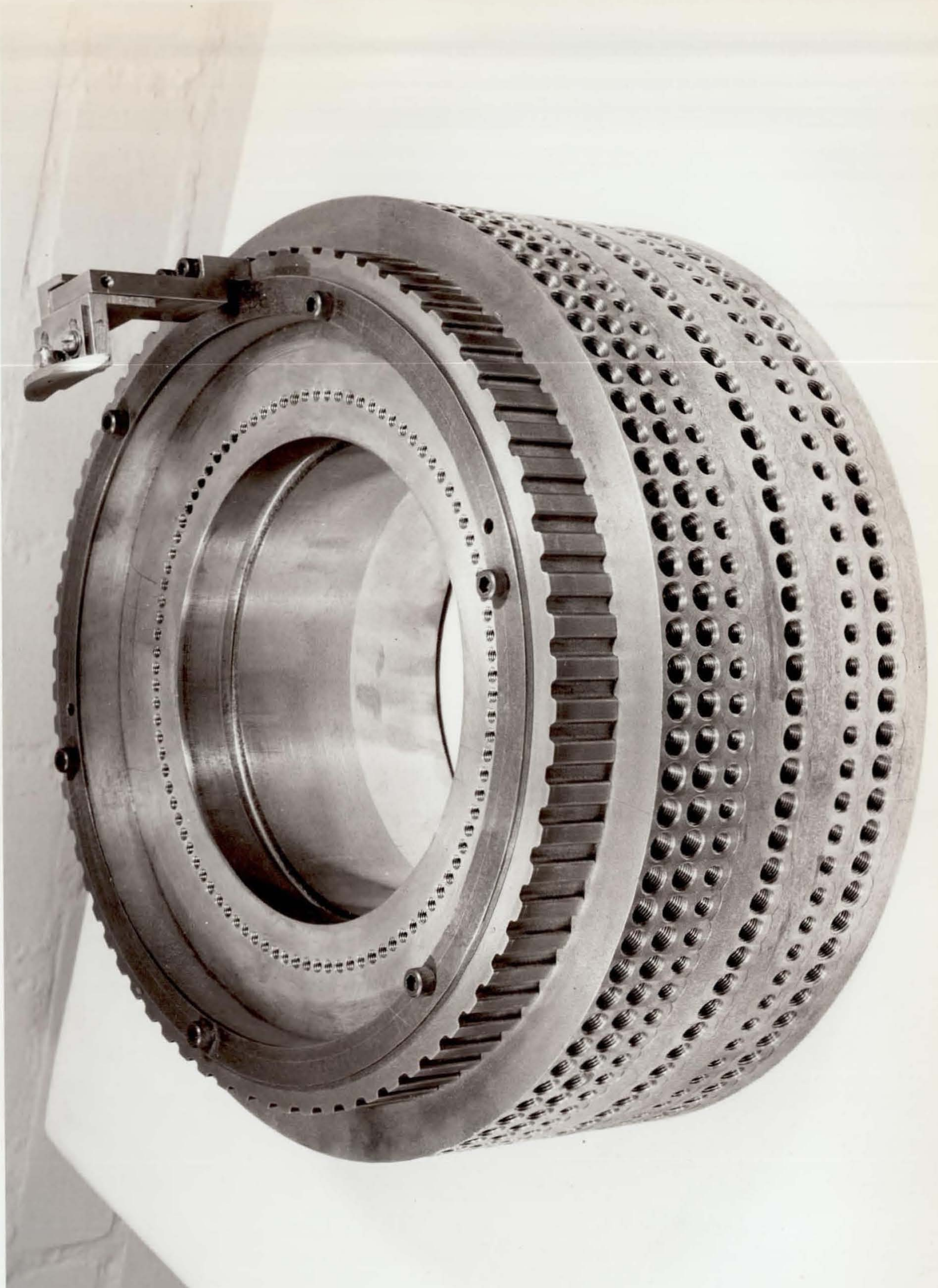


FIG. 37 YARN CARRIER ASSEMBLY

over a roller to emerge horizontally from the machine. Tests were performed on the Griswold machine and provided that the vertical fall of knitted fabric exceeded six inches, no knitting complications were introduced by changing the direction of the fabric. The tensioning force on the fabric was introduced by two rollers that were rotated in a set relationship to the speed of the rotary valve. Figures 38 to 42 (drawing nos. J.D.G. 049, J.D.G. 050, J.D.G. 051 J.D.G. 052, J.D.G. 053) give details and dimensions of the framework required to mount the actuator block to the existing chassis and frame, and rollers. The roller blocks were made adjustable and the fixing holes for the various components elongated, to enable timing belt length adjustment to be made.

To complete the design, the hydraulic circuits between the rotary valve and the actuators, together with the restrictors for the hydraulic stops had to be accommodated. This again involved using manifold blocks in the circuits between the rotary valve and the actuators. Inserting a manifold block enabled the pipes from the actuators to be made symmetrical, thus enhancing its appearance. It also allowed larger bore pipe to be used between the rotary valve and the manifold block, thus reducing the overall pressure drop in the pipes. The actuators each required two hydraulic stops to be returned to tank. This was simplified by using a common connecting block into which the pipes could be fed, with a single large bore pipe taken to tank. These designs can be seen on drawings J.D.G. 040 and J.D.G. 041 in figures 43 and 44 respectively.

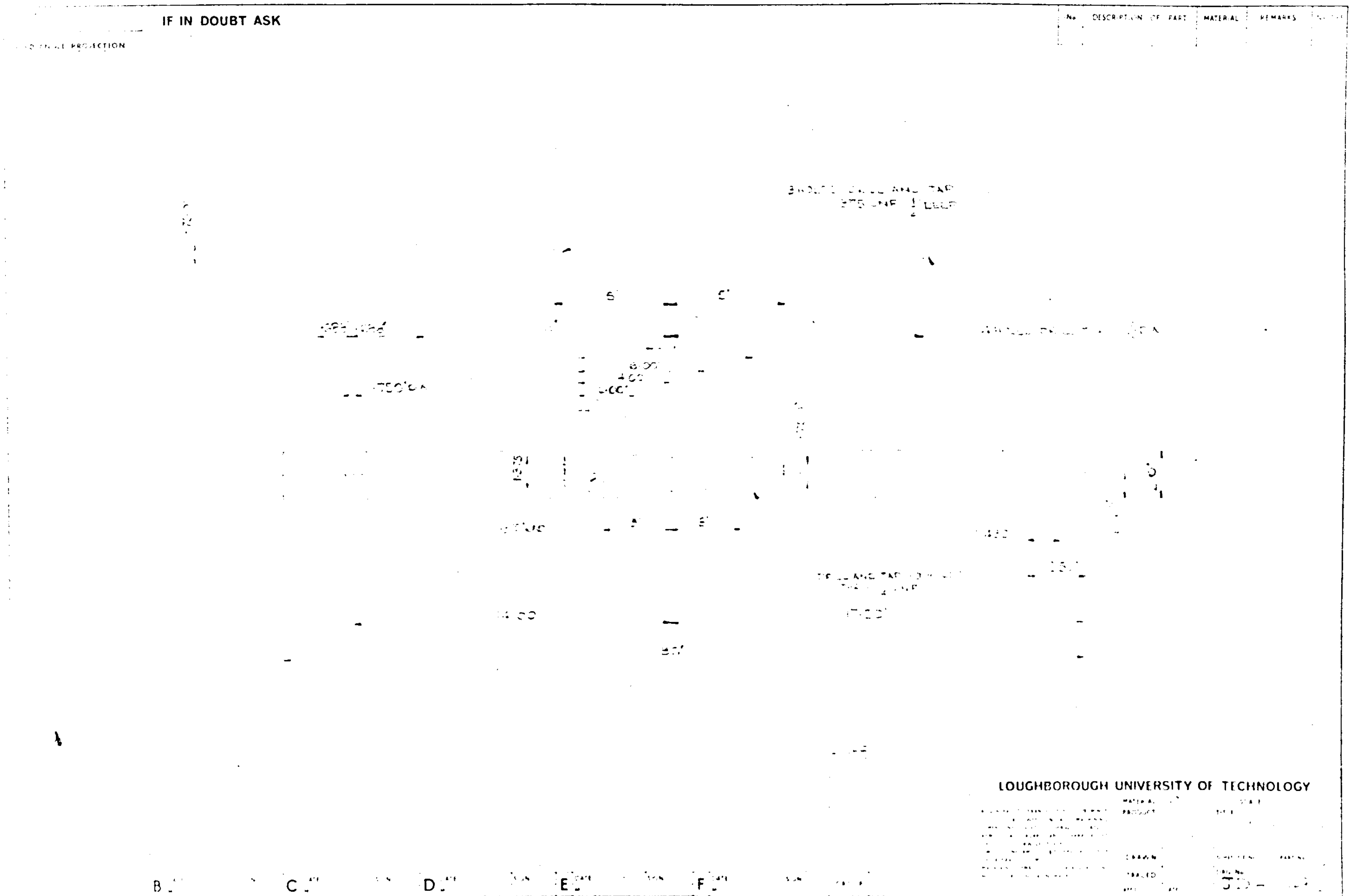
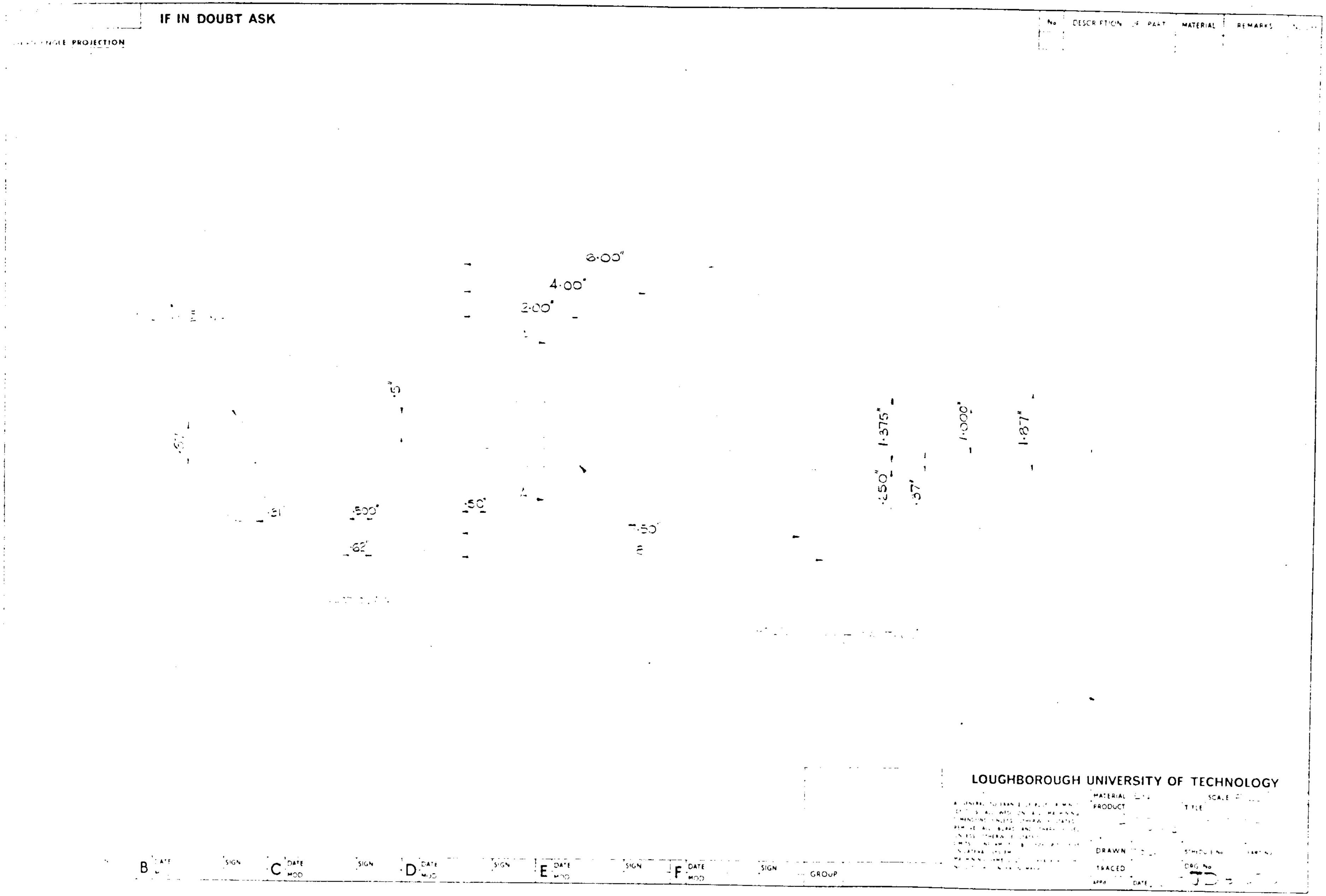


FIG. 38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PROJECTION
No
MATERIAL
REMARKS
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4·00"
2·00"
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1·000"
1·87"
·250"
·57"
·500"
·62
·50"
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
PRODUCT
SCALE
DRAWN
TRACED
DATE
DRG No
B
C
D
E
F
SIGN
GROUP

FIG. 39

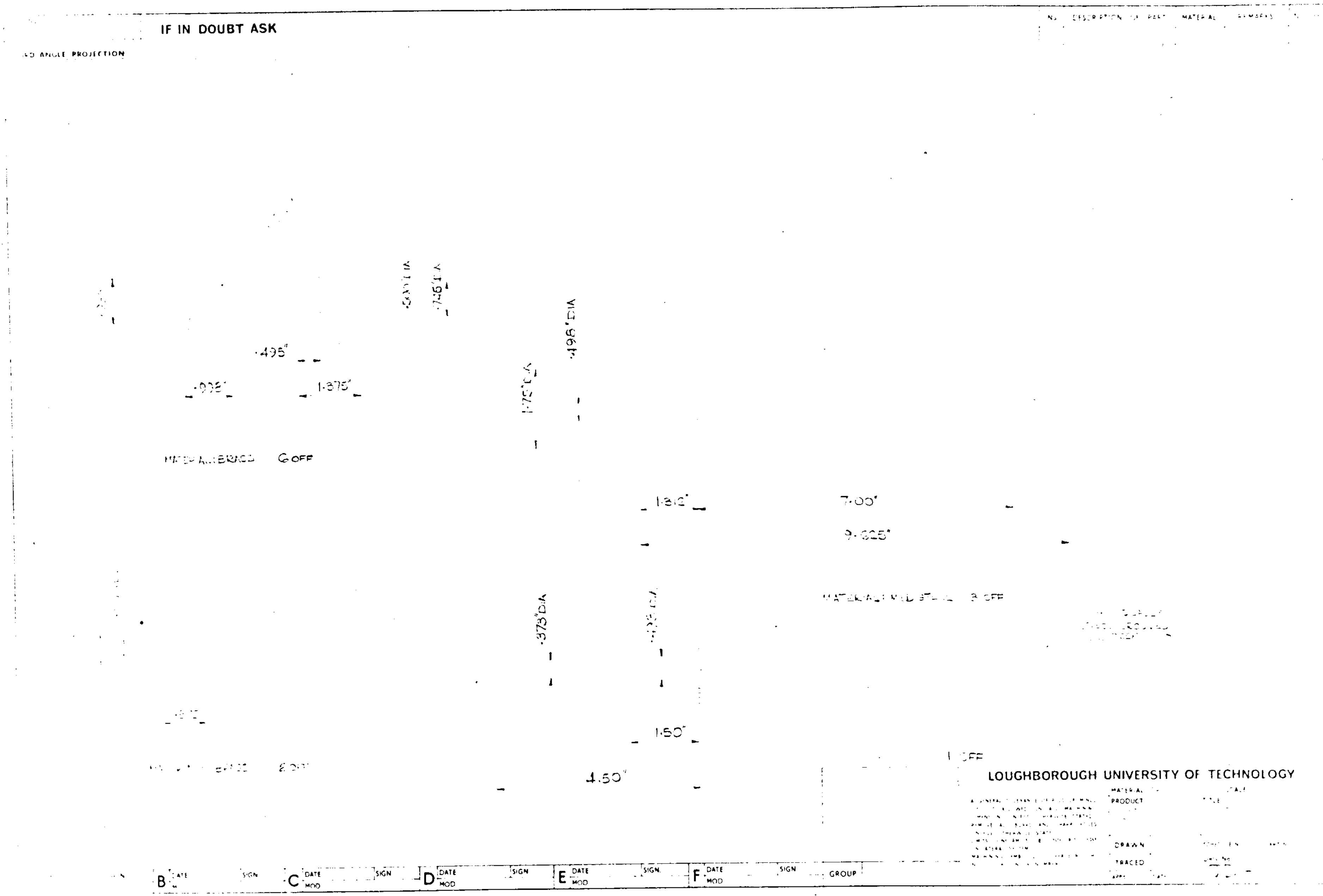
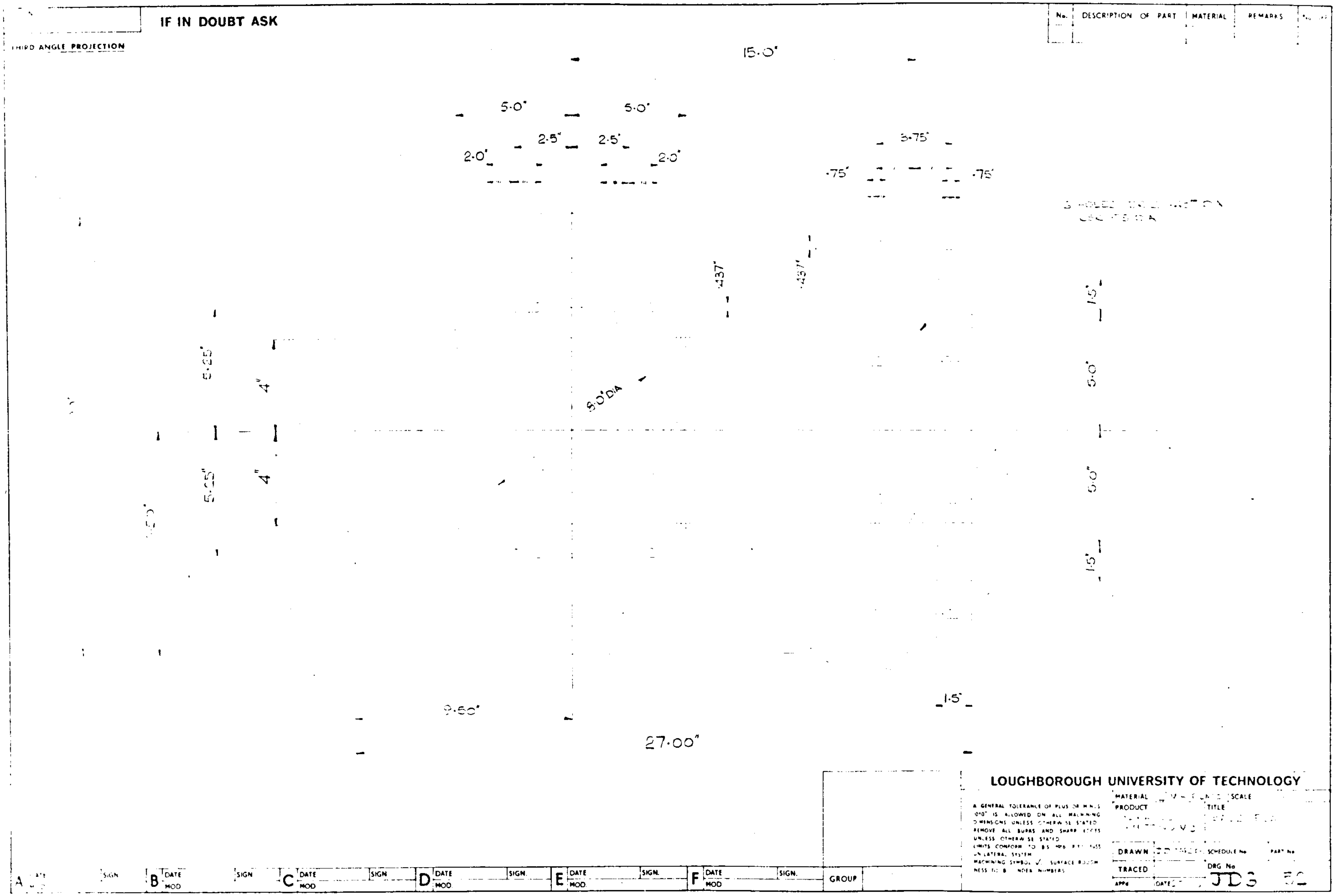

IF IN DOUBT ASK
ANGLE PROJECTION
·495"
1·375"
·496"DIA
1·75"DIA
GOFF
1·312"
7·00"
9·625"
·373"DIA
1·50"
4·50"
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
PRODUCT
DRAWN
TRACED
DATE
MOD
SIGN
GROUP

FIG. 40


IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
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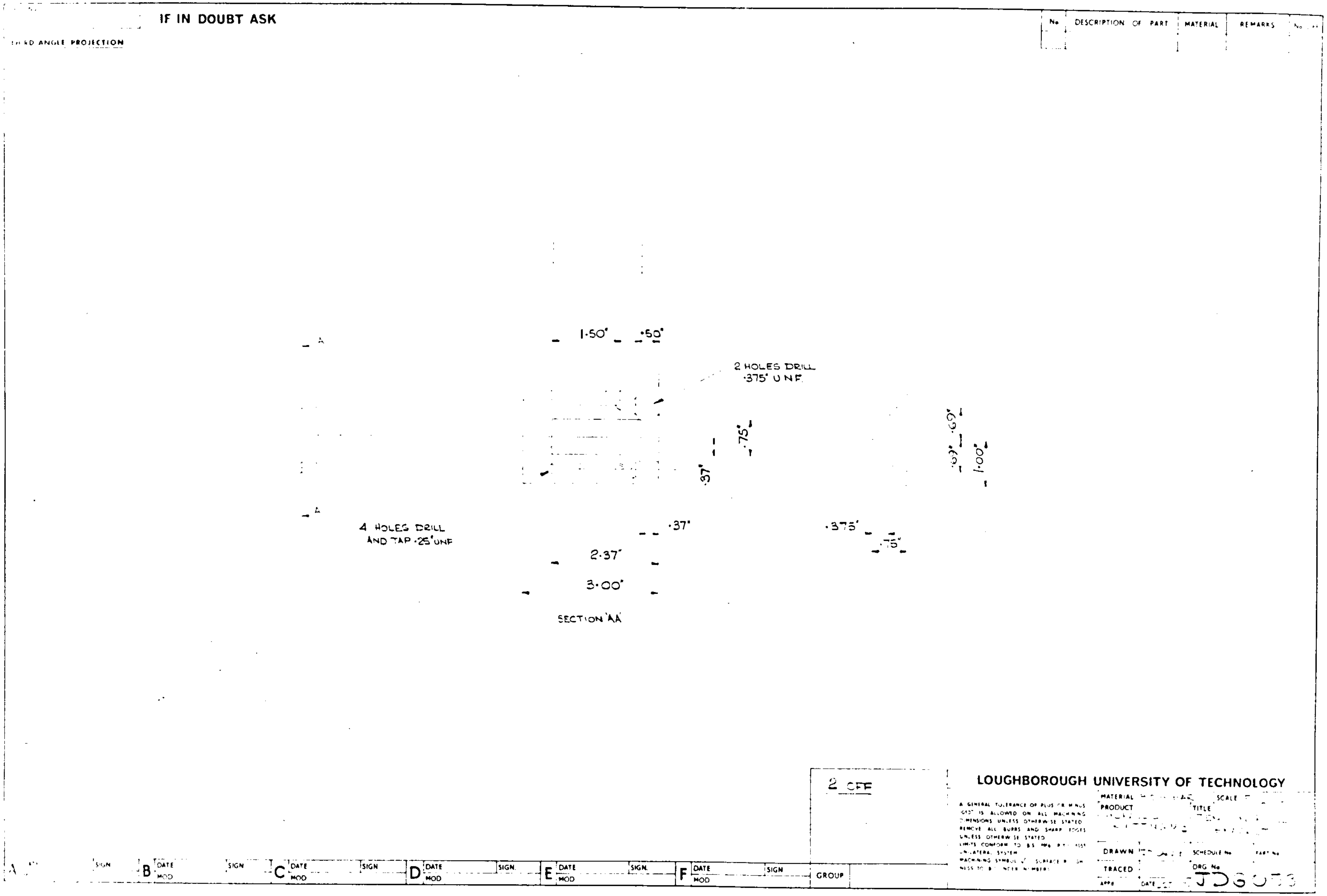
5·0"
5·0"
2·5"
2·5"
2·0"
2·0"
5·75"
·75"
·75"
·437"
·437"
1·5"
5·0"
5·0"
1·5"
5·25"
5·25"
4"
4"
9·0" DIA
9·50"
27·00"
1·5"
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
SCHEDULE No
PART No
DRG No
GROUP
JD3

FIG.41


IF IN DOUBT ASK
THIRD ANGLE PROJECTION
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·50"
2 HOLES DRILL ·375" U.N.F.
·75"
·37"
·69"
·69"
1·00"
A
A
4 HOLES DRILL AND TAP ·25" UNF
·37"
·375"
·75"
2·37"
3·00"
SECTION 'AA'
2 OFF
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
SCHEDULE No
PART No
DRG No
A
SIGN
B
DATE
MOD
C
D
E
F
GROUP

FIG. 42

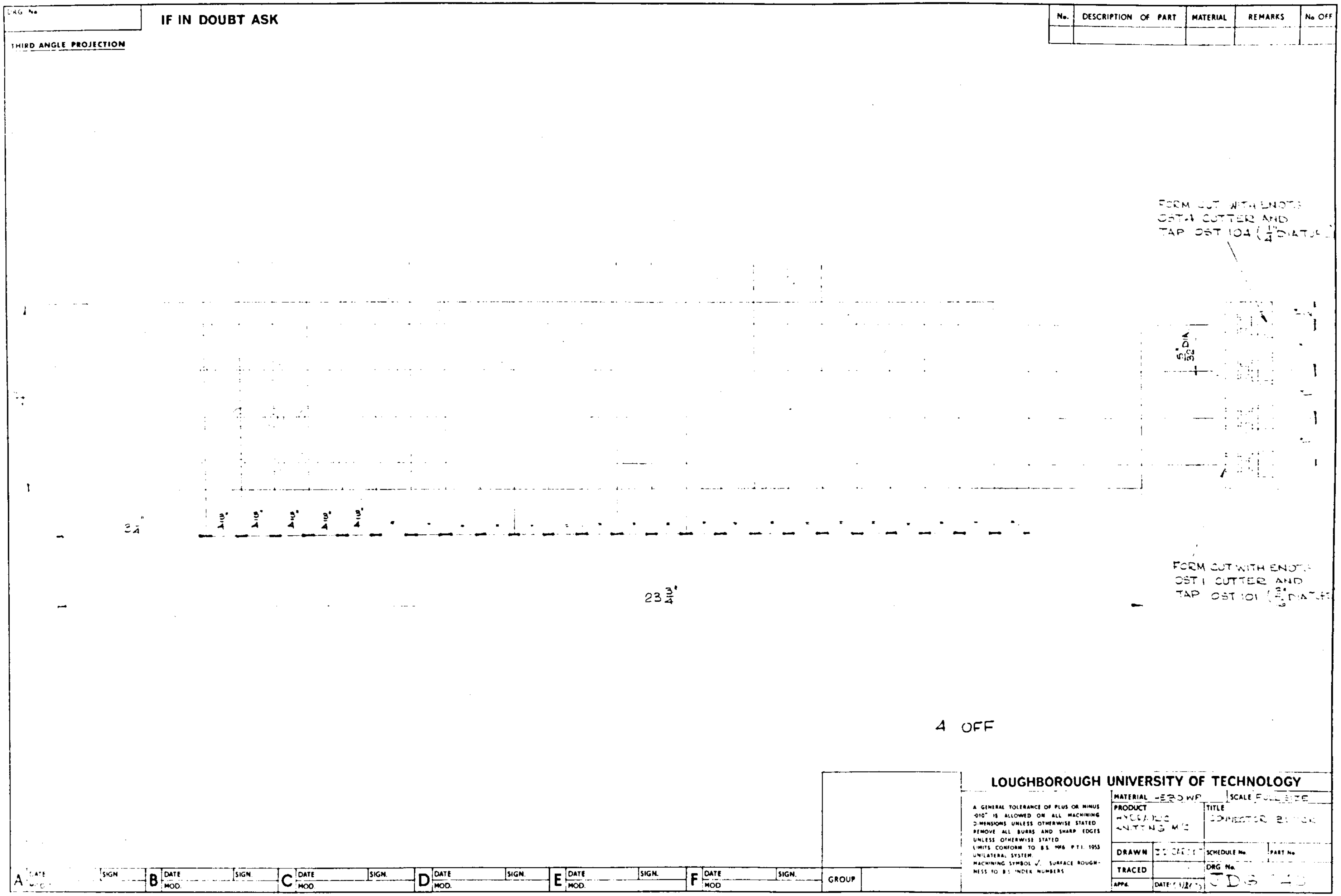
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. | DESCRIPTION OF PART | MATERIAL | REMARKS | No OFF
23 3/4"
4 OFF
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
SCHEDULE No.
PART No.
DRG. No.
GROUP

FIG. 43

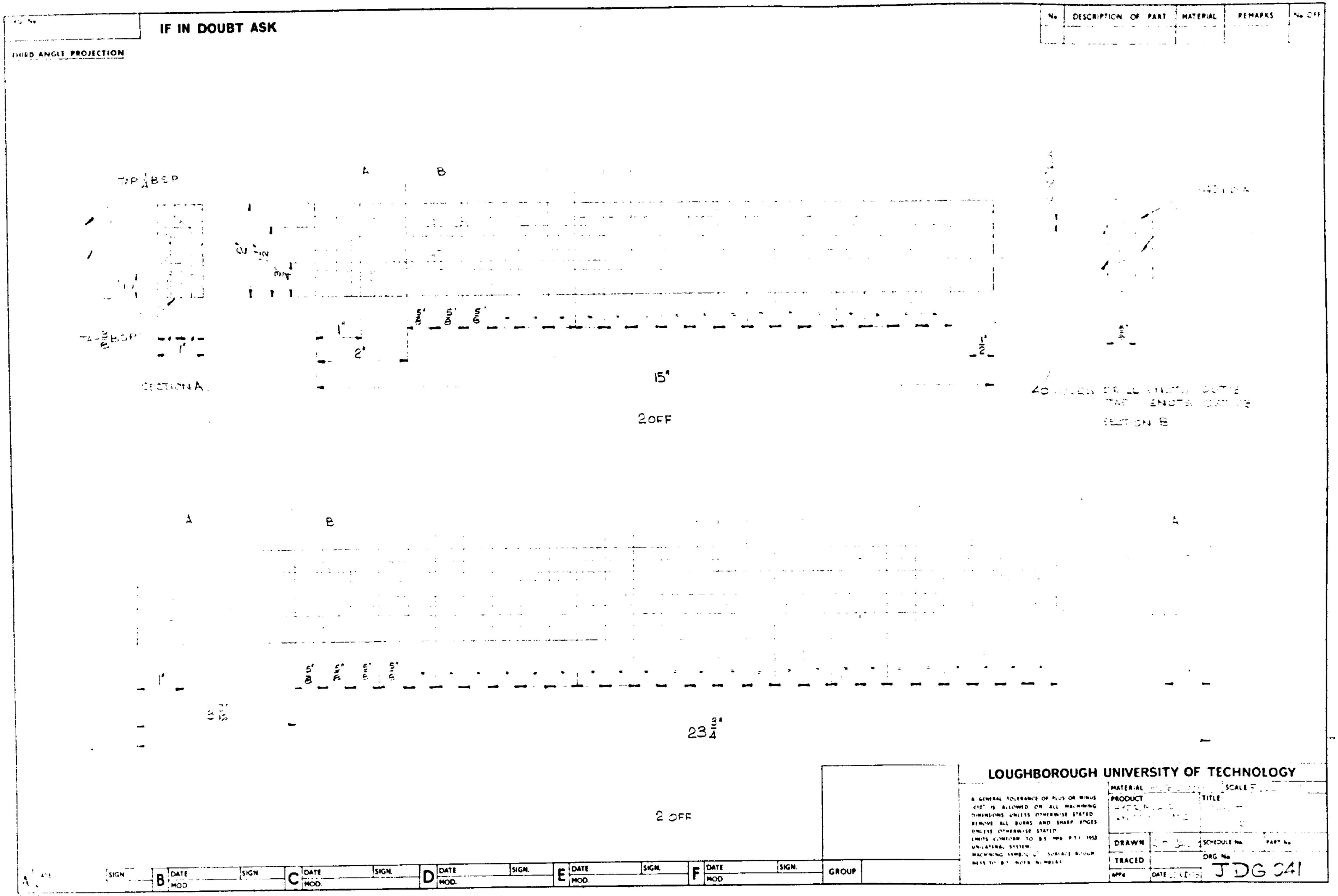
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No
DESCRIPTION OF PART
MATERIAL
REMARKS
No OFF
A
B
SECTION A
15"
2 OFF
SECTION B
23 3/4"
2 OFF
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
DRG No
GROUP

FIG. 44

3.2.9. Assembly of the Hydraulic Knitting Machine.

All the components were manufactured by the technical staff of the department ready for assembly. Each actuator was built individually; all the bores were lapped to ensure a smooth finish with no burrs where the supply holes entered the actuator. The tuck probe and the knitting piston were checked to ensure free movement before the sealing mechanisms were inserted, completing the actuator assembly. As each actuator was completed it was tested using a servo-valve. This ensured that all the supply ports had been drilled into the actuators and that it would operate in both the knit and tuck mode at cycling rates of 20 cycles per second using a supply pressure of 200 lbf/in^2. If this specification was not achieved or the 'O' seal allowed excessive oil to escape, then it was broken down and rectified. When the complete actuator block had been tested and assembled six actuators were found to have machining inaccuracies that were probably due to blow holes or hard particles in the original casting. However, after polishing and careful assembling, four were found to operate at a slightly higher pressure, leaving only two actuators not operating with 100% repeatability. These two actuators would operate at low speeds but when performing at high cycling rates, the amplitude of the tuck stroke was attenuated. At this stage the actuator block was accepted as complete.

The next stage in the building sequence was to fix the actuator block onto the base plate (see J.D.G. 052) and to assemble the knitting machine complete with manifold blocks, and tensioning rollers. Once the framework was complete, the yarn carrier drive was fitted. Care was taken to align the idler pulleys to the two main stepped pulleys mounted at right angles, before fitting the endless belt over the pulley system. The slack in the belt was taken up by adjusting the position of the actuator block on the base plate, before finally fixing all the components. Power was applied to the variable speed drive and the belt system operated with no tendency to run off either pulley wheel. The belt system to the small variable speed gearbox was installed on the outer edge of the machine.

3.2.10. Piping the Hydraulic Circuits

All that remained was to construct the hydraulic circuits between the rotary valve and the actuators. Consideration had been given to the rotary valve's rotational direction and to the yarn carrier mechanism, so that the actuators could be piped in sequence following the same direction of rotation. A set of pipes joined the rotary valve to the manifold block. (Position number one). The next set of ports in sequence from the rotary valve were connected to position number two in the manifold block. This procedure was repeated round the rotary valve and manifold block until all the ninety six supply ports were connected in sequence. The pipe connections between the valve and manifold block were made using .25 inches outside diameter flexible nylon tube with Enots fittings. Five colours were used to provide a visual check for the sequencing of the connections and assisted when tracing the various supply paths.

The connections between the manifold blocks and the actuators was a straightforward process of linking the correct ports in sequence. Figure 45 shows the actuators as piped to the manifold blocks.

3.2.11. Initial Tests on the Hydraulic Actuators.

When all the actuators were piped to the rotary valve, the rig was connected to a hydraulic power pack. After flushing the system for several minutes, the supply pressure was increased with the rotary valve stationary. On applying the power to turn the rotary valve, the actuators started operating in sequence forming a perfect cam profile, as defined in figure 1, circulating round the top of the actuator block like a moving wave. Each motion of the actuator was so precise that only one actuator performed a particular movement at any instance. A film of the circular actuator block was taken at this stage, to enable a direct comparison to be made with the first multi-actuator block. This sequence can be seen in the film "Development of a Hydraulic Knitting Machine" and it shows the actuators operating at various cycling rates, up to 40 hertz. It must be realised that the speed of the rotary valve must be multiplied by a factor of four to obtain the equivalent cycling rate between the two systems, because each actuator is only pressurised for 25% of a complete revolution of the rotary valve.

3.2.12. Introduction of the Knitting Mechanism.

After the system had been run faultlessly for ten hours, the decision to complete the machine was taken. It had been

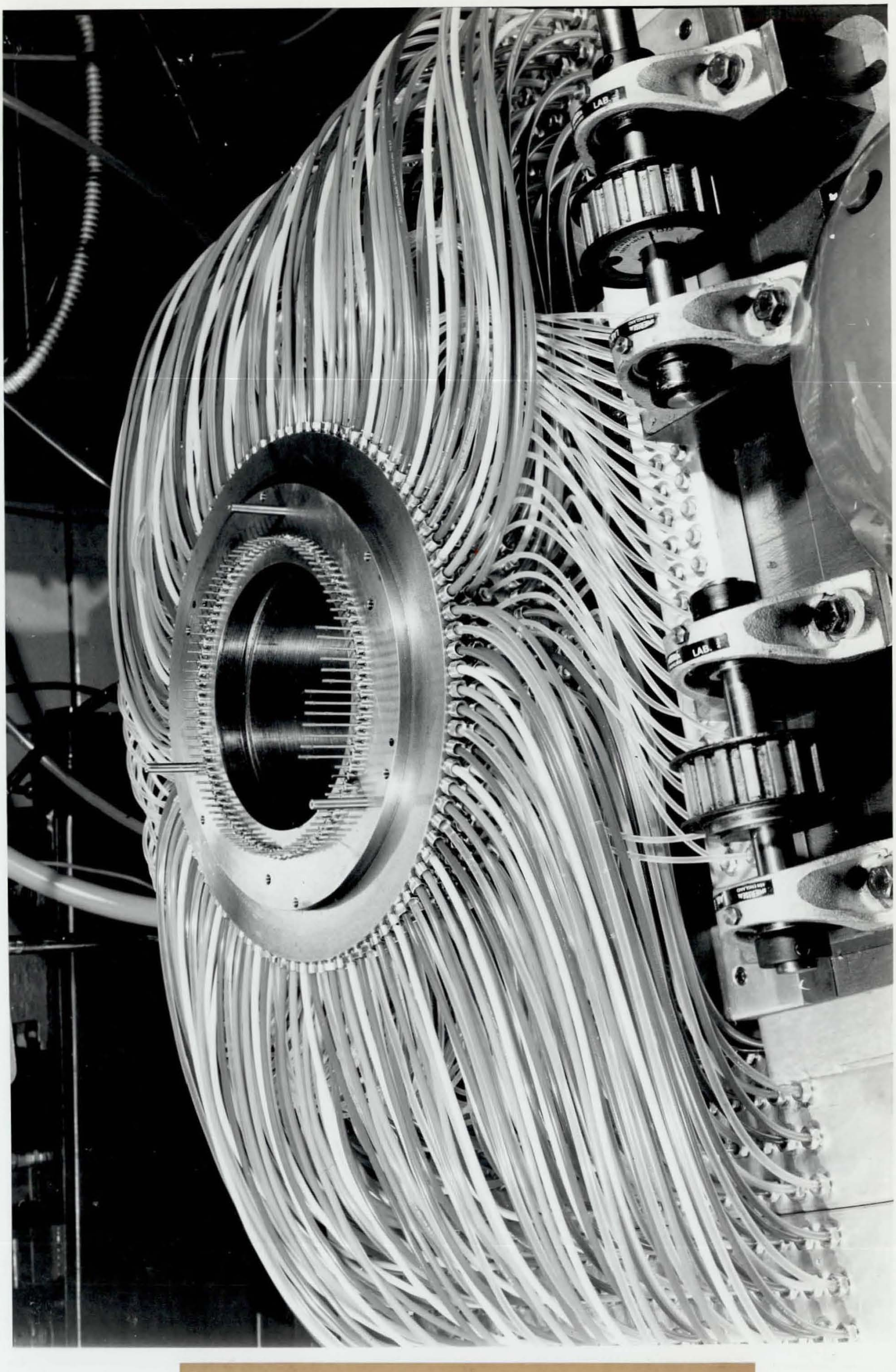

FIG. 45 ASSEMBLED ACTUATOR BLOCK PIPED INTO THE RIG

envisaged that the needles should be attached to the actuators by means of a collar with two grub screws providing the locking force. However, several linking methods were tested, and it soon became apparent that the lighter the linking piece, the more reliable it would be at high cycling rates when inertia forces are at a maximum. Several methods involving indentation of the actuator shaft were investigated but resulted in the indent shearing. The method which was finally adopted, and found to hold the needle onto the actuator at cycling rates of 75 hertz, was to solder the needle into a brass slug, and using a 9 B.A. thread on the end of the actuator piston rod, to screw the needle onto the actuator. This technique had two main advantages in that the height of the needle could be adjusted by screwing the piston rod into the needle; the linking member could be made using .152" hexagonal brass rod thus keeping the inertia low. A tension test was conducted on several of the methods investigated. Under pure tension, the screw thread required a breaking force of 200 lbf. to snap the rod across the thread. This technique was only intended for use on a prototype machine because of inherent limitations i.e. having to remove the trix to change a broken needle and having to cut a screw thread on the piston rod, thus creating a stress raiser. However, it would enable the question of "Will it knit?" to be answered. It was envisaged that a further area of research would be in techniques for attaching the needles to the actuators. Techniques of producing the actuator and needle as a single unit, or friction welding the needle to an actuator, or sweating the two components together, all require investigating. The most suitable solution would be a small clip that would allow broken needles to be removed from the actuator rod without disturbing the knitting.

A batch of needles were cut off at the butt and soft soldered into brass slugs that had been machined in the base with a 9 B.A. thread. The soldering was performed using a small jig to maintain an alignment between the needle and the screw thread. Each actuator rod was then screwed for .375 inches and a needle attached. Each needle was set to a predetermined height and locked into position using a locking nut coated in shellac. When all the needles had been correctly fitted, the knitting trix was offered up to the actuator block. It was suspended above the needles on the locating rods and each individual needle introduced into a guide slot before lowering the complete trix to the knitting position and clamping it with locking nuts. The machine was now complete and ready for commissioning.

3.3. Testing the Circular Weft Knitting Machine

3.3.1. Initial Attempts to Produce a Knitted Fabric.

All the needles in the trix were lubricated and the machine run with no yarn in the carrier to check that the needles would function correctly. It was realised that as knitting was about to commence the slowest speed of the variable speed drive would not provide sufficient flexibility when setting up the machine for knitting. This difficulty was overcome by fitting a mechanical handle to the main electric motor shaft.

The yarn carrier was fixed onto the stepped pulley and adjusted radially to be in the same sector as the needles, constituting the knitting profile. The height and position of the yarn carrier was also adjusted relative to the needles so that the yarn would be introduced under the hook of the needles. A length of loosely knitted fabric was threaded through the pulleys and hooked over the needles. A new yarn was introduced via the yarn carrier and with the hydraulic pressure set at 300 lbf/in^2 the machine was turned by hand. This produced a long "spider's web" with about 10% of the needles knitting at any particular time. The reasons for not knitting were twofold. Firstly, the knitted stitches did not appear to fall off the needles. This could be traced to the trix not being bevelled sufficiently to allow the knitted fabric to be pulled down off the needles. Secondly, the yarn was not introduced accurately into the hook of every needle as the carrier was rotated. The first problem was solved by re-shaping the curve on the inside of the trix. This was remachined to form a 20° bevel that would allow the knitted fabric to be pulled off the needles without making contact with the lower reaches of the trix. To solve the yarn carrier problem, a new yarn post was introduced capable of feeding the yarn to the needles from a higher position. After re-hooking the length of fabric onto the machine, it was again turned over by hand. A piece of fabric resembling knitting was then produced. The setting for the yarn carrier was readjusted until the machine could produce an acceptable fabric on power drive at low running speeds.

The quality of knitting appeared to be good, though when the knitting speed was increased, the yarn tended to vibrate and jump over the hook of the needles.

3.3.2. Modifications to the Yarn Carrier.

To overcome this yarn vibration, a tensioner was introduced making a marginal improvement to the performance. The results at this stage were encouraging. However, when the yarn carrier was moved closer to the needles, to a position where the latches appeared to be at rest 90% of the latches were broken in two subsequent revolutions of the machine. The machine was completely re-needled and a small circular brush was placed on the yarn carrier, so that as the needles travelled from the miss to the knit position, the latches were held open by the bristles. This brush had an advantageous effect in helping to pick up dropped stitches. It also allowed the yarn post to be positioned nearer the needles without damage being caused to the latches. At this stage, the machine was knitting relatively satisfactorily, producing a row of 96 stitches per second. On close examination it was seen that the yarn was caught by the needles as they travelled from the knit to the tuck position. Theyarn carrier was lowered to prevent this happening. However, the occasional latch bounced and closed before the new yarn was introduced in the hook, thus creating a ladder. A new yarn carrier was designed based on the latch guards fitted to production machines. This technique involved introducing a plate over the needles when the latches were open so preventing the latches from closing until the new yarn was introduced. With the new yarn carrier, the improvement in performance was very marked and speeds exceeding 180 revolutions per minute were easily achieved. In a paper by J. Knapton (17) titled "The Dynamics of Weft-Knitting : a Mathematical Analysis", it is quoted "At present, needle velocities, as they are moved through the cam system, are limited to the order of 165 cm/sec., equivalent to a $3\frac{3}{4}$ inch diameter hosiery machine running at 230 revolutions per minute" (Textile Research Journal August 1966). Thus, the hydraulic knitting machine, when operating at 115 revolutions per minute was running faster than the quoted $3\frac{3}{4}$ inch diameter hosiery machine.

3.3.3. Film of the Hydraulic Knitting Machine

The knitting machine operating at various speeds can be seen in the latter section of the film titled "Development of a Hydraulic Knitting Machine". This film shows the overall layout of the rig together with the knitting of fabric at various speeds from 60 to 180 rows per minute. At the slower speeds, the knitting is 100% but as the knitting speed greatly exceeds 165 cm/sec. two ladders appear in the fabric in positions corresponding to the faulty actuators. A close-up of the knitting profile filmed at the laddered section clearly indicates that the needle overshoots the tuck position and comes to rest below the other needles. This overshoot causes the yarn to miss the hook and results in a dropped stitch. The actuator block could have been modified by plugging the offending bores and re-machining the two actuators, but at this stage it was not deemed necessary, as the system could be made to operate with 98% accuracy at speeds approaching double the accepted knitting speed for cam-driven knitting machines. (Also, the reason for the faults had been established). In future machines, the actuators would have to be produced as exchangeable units in order to allow for any inconsistencies in machining.

3.3.4. Description of the Completed Machine

Figures 46 and 47 show general views of the complete machine together with the knitted fabric as it emerges through the tension rollers. Figure 47 shows the complete yarn feed mechanism, the bobbin holder, and the tensioning mechanism. The modified yarn carrier still utilises the brush to open the latches, but the triangular-shaped latch guard holds the latches open while travelling from the knit to tuck position. The fabric take-off and tensioning mechanism can be seen in figure 48. The variable ratio gear box allows the take-off rate to be adjusted to suit the type of fabric being produced. This mechanism was satisfactory for the prototype machine, but for future machines a well-established commercial take-down mechanism should be used. These have a compensating mechanism whereby the tension in the fabric is obtained by a combination of a static load and a dynamic pull exerted by the rollers. The timing belt drive for the yarn carrier can be seen in figure 49. This shows the technique used to alter the direction of the drive from the horizontal to the vertical plane by means of the idler pulleys. The idler pulley as can be seen in the figure, also provides the drive for the fabric tensioning gear box.

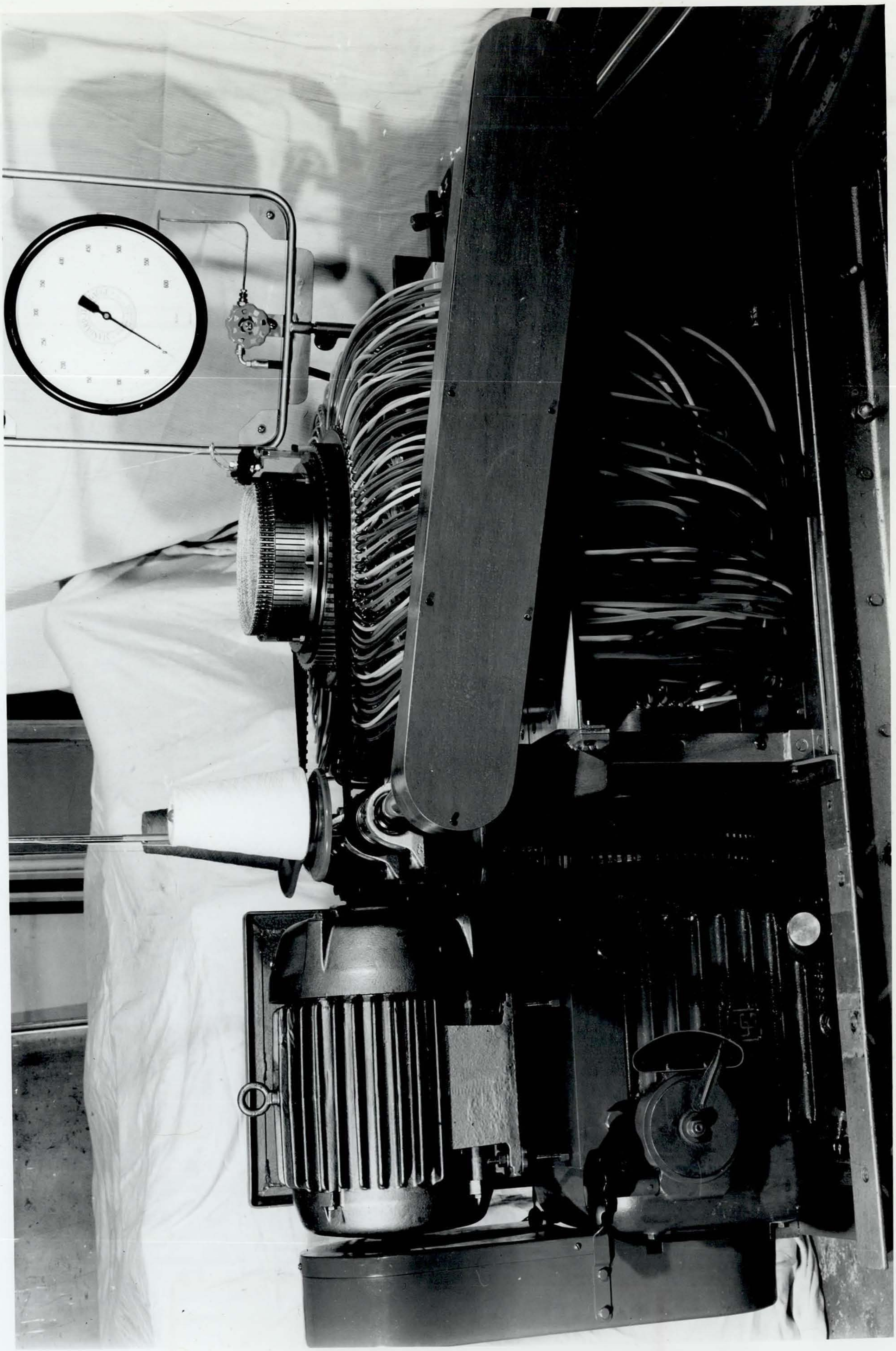

FIG. 46 SIDE VIEW OF THE HYDRAULIC KNITTING MACHINE

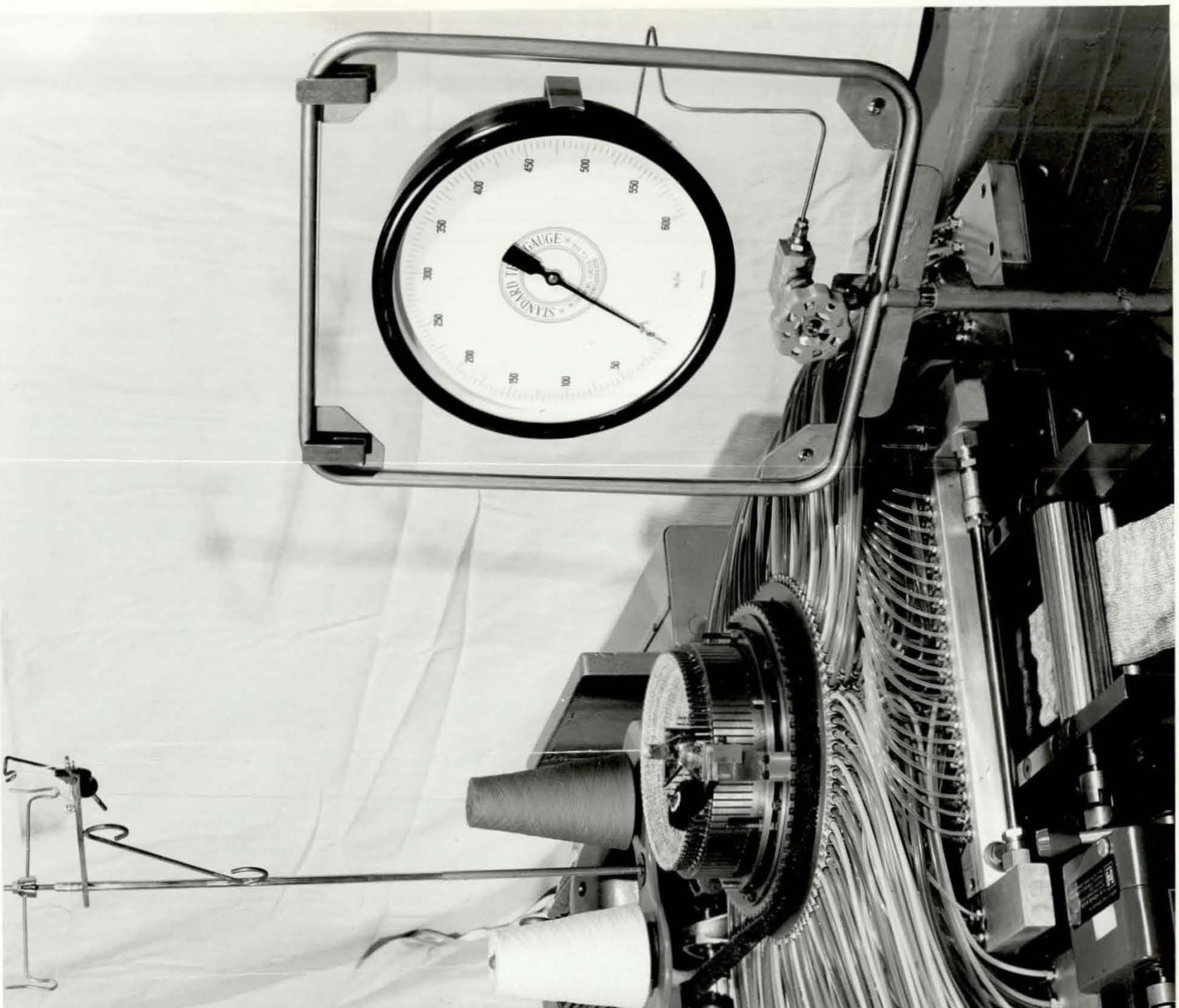

FIG. 47 GENERAL VIEW OF THE KNITTING TRIX

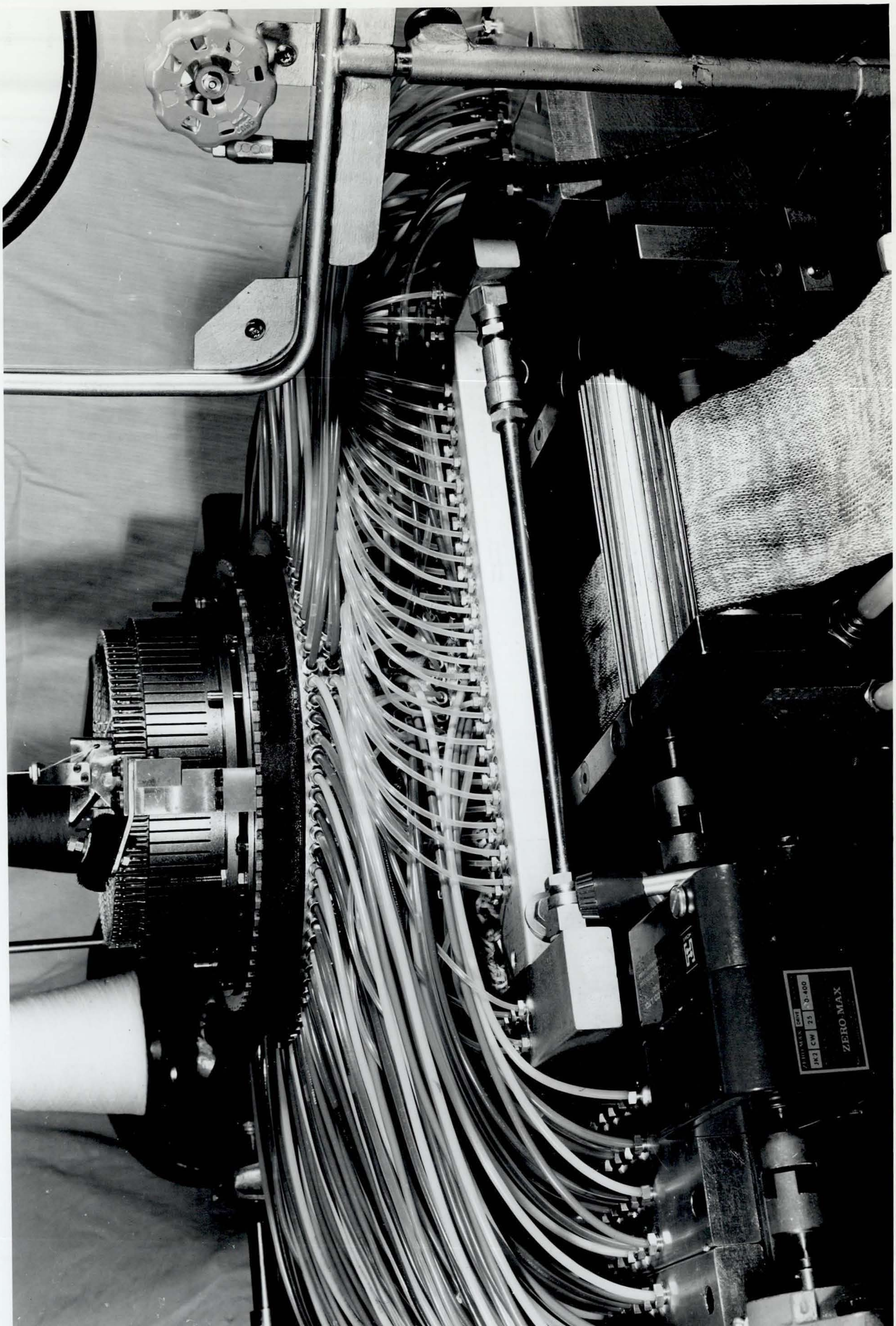

FIG. 48 FABRIC TAKE-OFF MECHANISM

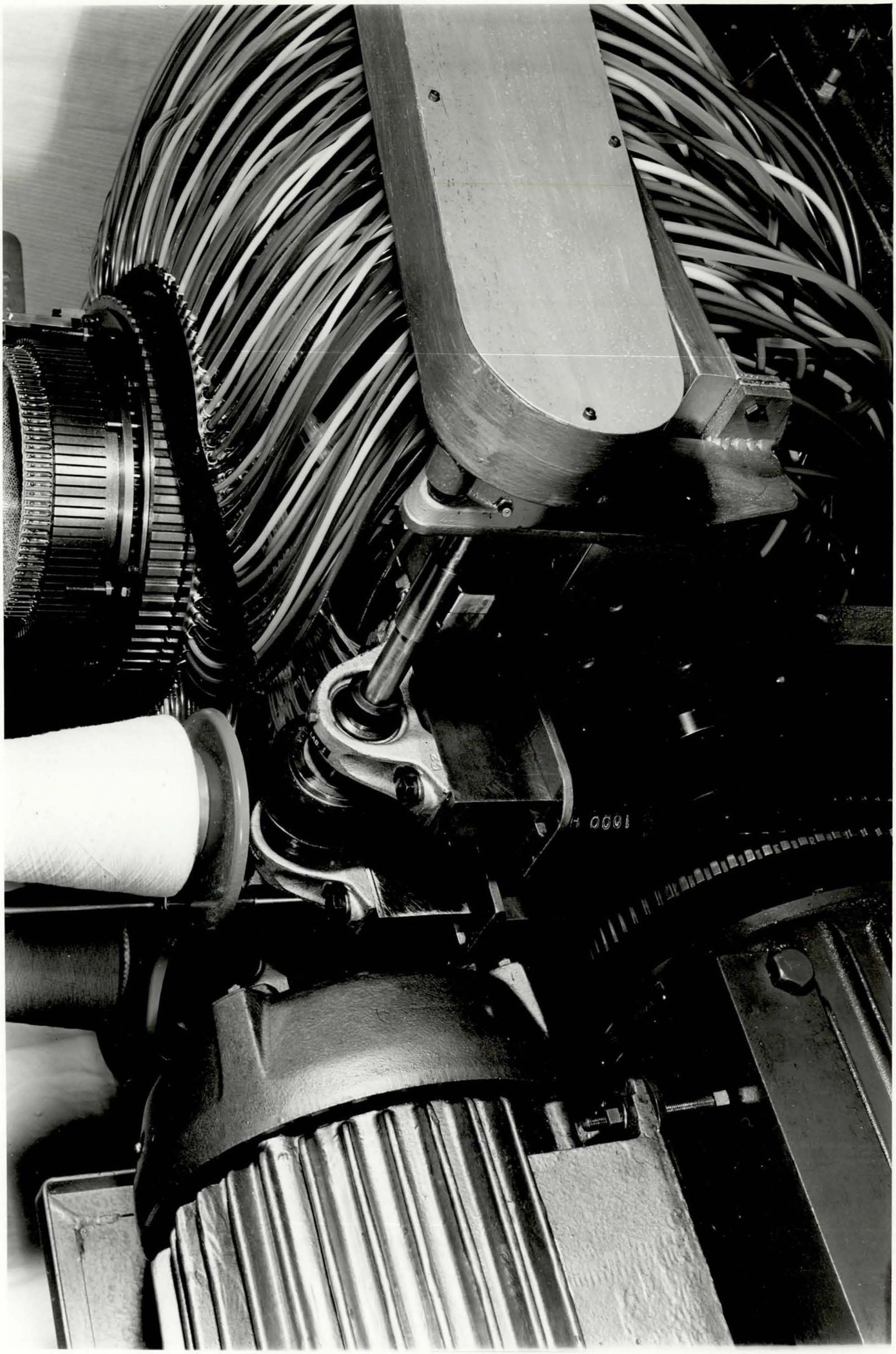

FIG. 49 YARN CARRIER DRIVE

Figure 50 shows the knitting as it is held on the needles. It will be noted that the cam profile is a block formation with six needles in the knit, and six in the tuck position, indicating that only one needle transposes position at any one time. This feature makes a hydraulic knitting machine superior to present day cam driven machines. On a cam driven knitting machine, when the loop for a stitch is being formed, (i.e. knitting) several consecutive needles are all at some stage of loop formation. Consequently, the yarn has to be pulled through the hook of several needles which introduces tensions in the yarn leading to needle and yarn breakage. However, on the hydraulic knitting machine only one needle is knitting at any particular instance, which enables all the yarn to be pulled directly from the yarn carrier without passing through the other hooks, thus relieving tension and strain. This unique advantage is manifest in the machines' ability to knit cotton yarn at high knitting speeds (as can be seen in the film). The quality of the knitted fabric is of an acceptable standard, because the characteristic described above (known as pull-back) has been eliminated by knitting with a single needle.

3.3.5. Comments on Machine Performance.

This prototype hydraulic knitting machine has been run over considerable periods at various speeds, and on no occasion has either a needle or the yarn broken, in the process of knitting. The only fault in the design has been the breakage of actuator rods across the screw thread. The screw thread on the piston rod had introduced a stress raiser, and the slight

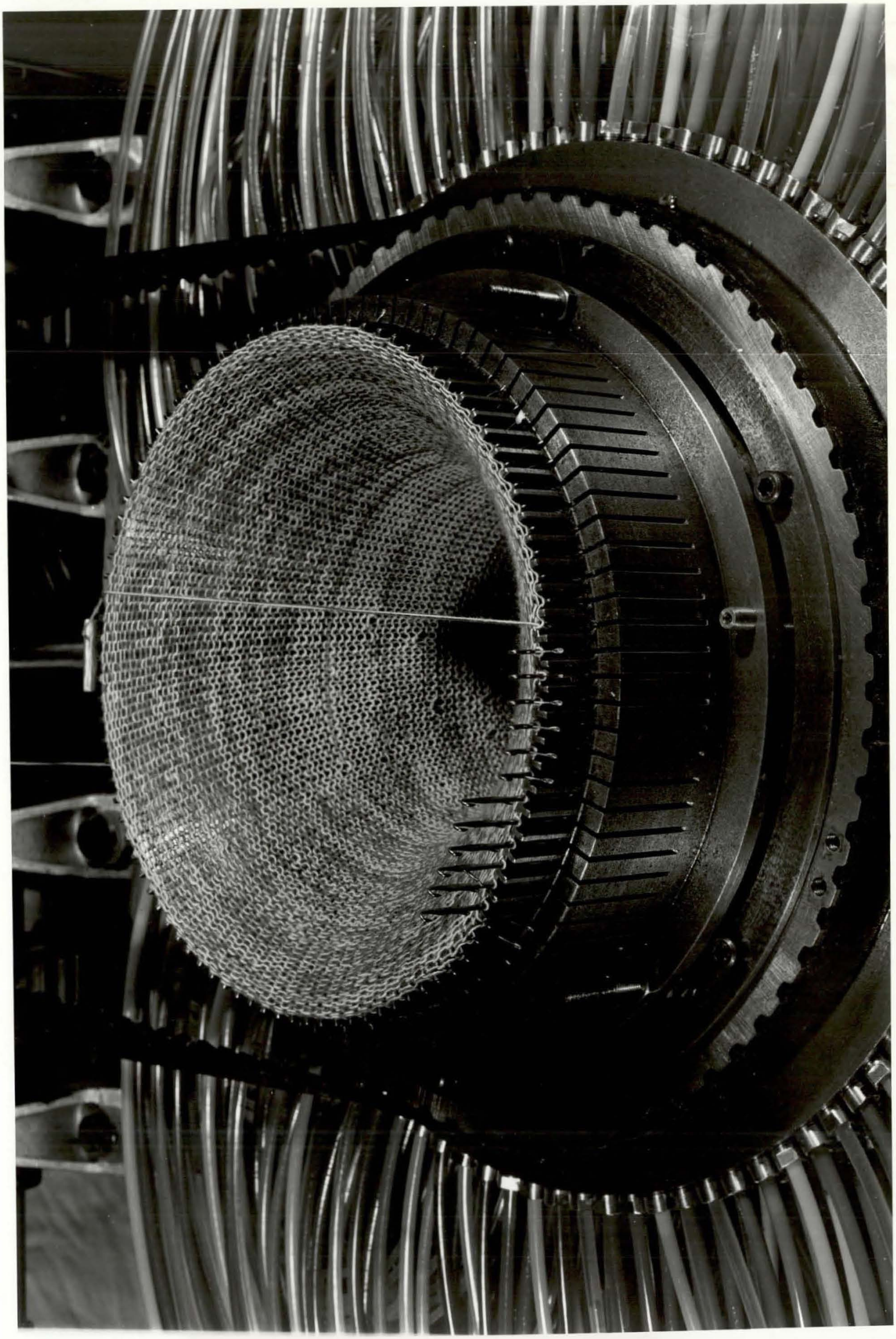

FIG. 50 KNITTING ON THE NEEDLES IN THE TRIX

deflection imposed on the rod due to misalignment of the piston rod and needle, eventually caused a fracture. This fault occurred on six occasions but never in the same actuator position.

The overall performance of the first prototype hydraulic knitting machine has been most encouraging, the quality and speed of knitting providing stimulus for future work. The concept of hydraulic knitting has been proved and the project has shown beyond any doubt that knitting can be produced by using hydraulic actuators.

3.4. The Hydraulic Power Pack.

The power pack used to supply the hydraulic oil was a Vickers V230 11 W Vane pump, driven by a 5 H.P. three phase motor at 1480 revolutions per minute. This system can be seen in figure 51 and it delivered 10 gallons of oil per minute at 400 lbf/in^2. It was custom-built in the department to suit the prototype hydraulic knitting machine. However, this present system is uneconomical from a power aspect due to the large number of needles being supplied with oil at the knitting station and by redesigning the rotary valve bobbin the same quantity of knitting could be produced using a 1.5 H.P. motor.

3.5. Findings and Recommendations for the Application of Hydraulic Actuation Techniques to Knitting Machines

3.5.1. Future Areas of Research

Now that a hydraulic knitting machine is a reality, with clearly defined advantages, the areas for future development

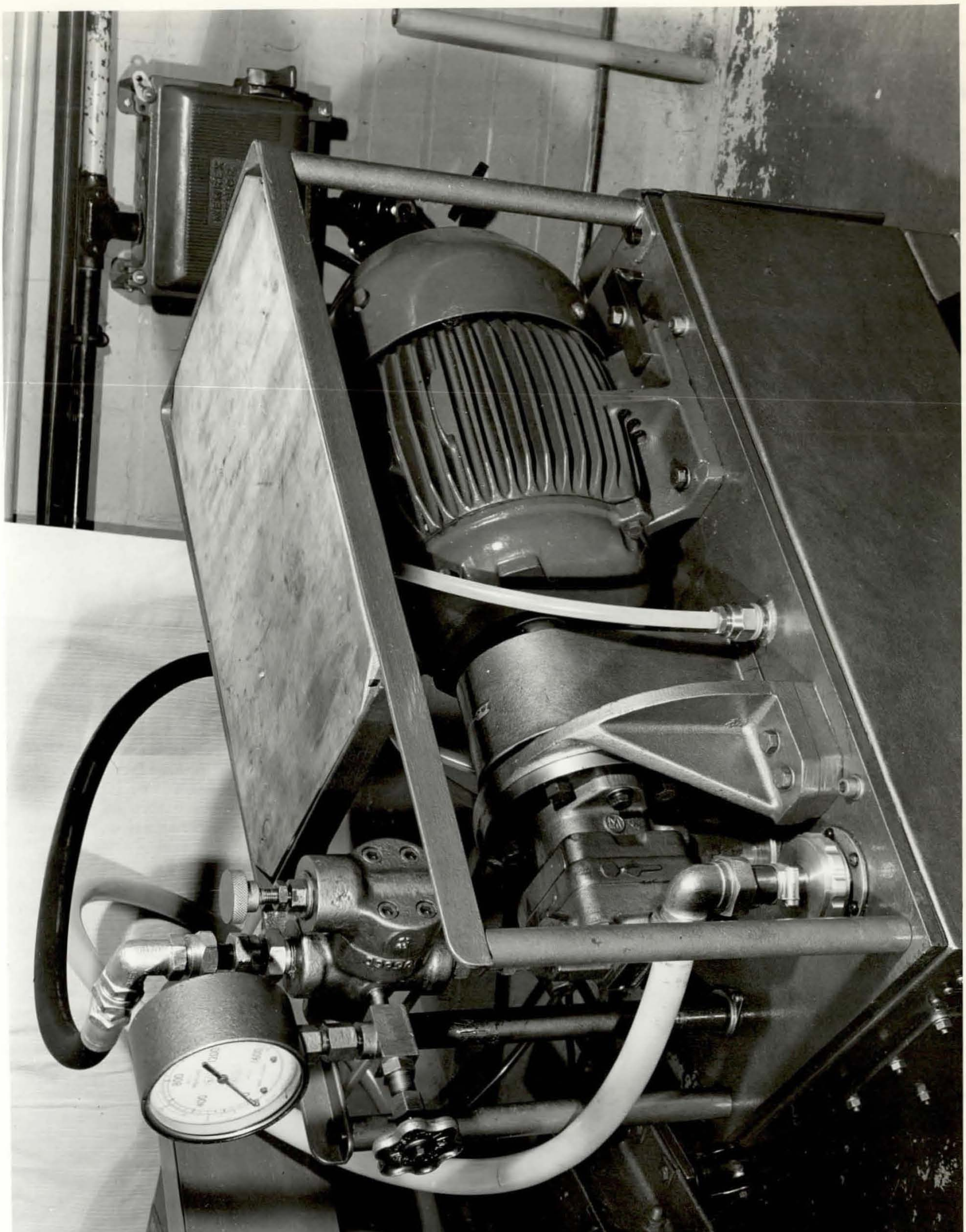

FIG. 51 HYDRAULIC POWER PACK

are as follows:-

(i) To examine the rotary valve's overall design in order to produce a design procedure for predicting the optimum valve dimensions, and hence calculate the performance characteristics by using mathematical models.

(ii) To study the orientation of the rotary valve and actuator block so that an integrated actuator-valve block could be produced, so eliminating the flexible piping.

(iii) To investigate the number of needles per feeder required for maximum fabric production.

(iv) To study more deeply the technique of knitting by hydraulic actuators, taking into account the present state of knitting technology as applied to cam driven machines.

(v) To decide in the light of technical experience gained, the most suitable commercial applications for hydraulic knitting machines.

(vi) To examine the feasibility of producing patterned fabrics using the various hydraulic control mechanisms.

3.5.2. Further Practical Usage of Hydraulic Actuation Techniques as Applied to Knitting Machines

The areas for further research mentioned above must take into account the greater degree of flexibility offered by a hydraulic knitting machine over a conventional cam-driven machine, and consideration must be given to the following concepts:-

(vii) The number of needles per knitting station is not governed by the length of cam track required to move the needles. Using hydraulic needle actuators, a knitting station with only two needles is feasible, although eight needles may well be regarded as the optimum number in the light of present technical knowledge. Eight needles would allow for two needles in the knit position, two needles in the tuck position, and four in the miss state, in readiness for the next cycle. From this, it can be seen that it would be possible to increase fabric production without increasing yarn velocities, by using a larger number of feeders. Taken to its ultimate conclusion, a circular warp knitting machine with a feeder per needle is feasible.

(viii) The bed configuration would no longer be limited to a circular or flat formation by the necessity of a cam drive. The flexibility of the actuators would enable any desired shape to be adopted. This could have applications when knitting a complete garment, e.g. when producing a pullover, three separate tubes (body, arms) could be knitted up to the arm holes after which the needles on the extreme perimeter of the tubes could be selected to knit an elipse (forming the neck section), hence a seamless garment.

(ix) The needle orientation of the machine could be varied to suit the yarn. It might well be advantageous to have some needles operating in the vertical plane progressively going to needles in the horizontal plane; using hydraulic actuators this could be accommodated.

(x) The gauge of the needles could be varied around the machine periphery, consequently when producing a body tube the section t o fit the front and back of the body could be knitted on a different gauge to the sides, thus increasing fabric flexibility and producing a better fitting garment.

(xi) To a limited degree, the needles could be made to move radially on a circular machine which would enable the gauge of the machine to be adjusted for shaping garments.

(xii) Complete control over each individual actuator would enable infinite patterns to be designed for exclusive garments.

(xiii) Very large gauge machines could be used for knitting carpets, exploiting the advantages of infinite pattern selection.

The hydraulic knitting machine developed in this project was never intended to have a direct commercial application but merely to demonstrate that knitting by using hydraulic actuators was possible. The research has clearly shown that the techniques outlined in this work have great potential provided that its application is considered carefully.

PART 2

THE CONSTRUCTION AND TESTING OF A HYDRAULIC LOCKSTITCH SEWING MACHINE

4. HYDRAULIC SEWING MACHINE

4.1. CONCEPT OF A HYDRAULIC SEWING MACHINE

4.2. SELECTING A MACHINE

4.3. THE MECHANICAL SEWING MACHINE

4.4. DESIGNING THE HYDRAULIC SEWING HEAD

4.4.1. The Sewing Head

4.4.2. The Rotary Valve

4.4.3. Mounting the Sewing Head and Rotary Valve

4.5. TESTING THE HYDRAULIC ACTUATORS

4.5.1. Initial tests and modifications

4.5.2. Instrumentation and U.V. Recordings

4.5.3. Comments on the U.V. Recordings

4.5.4. Comparison of the Hydraulic and Mechanical systems

4.6. TESTING THE ROTARY VALVE

4.6.1. Instrumentation

4.6.2. Modifications to Rotary Valve design

4.6.3. Readings for Rotary Valve performance

4.7. COMMISSIONING THE HYDRAULIC SEWING MACHINE

4.7.1. Timing the Sequence of the various Mechanisms

4.7.2. Testing the Sewing Action

4.8. FINDINGS AND RECOMMENDATIONS FOR THE APPLICATION OF HYDRAULIC ACTUATION TECHNIQUES TO SEWING MACHINES

4.8.1. Comments on the Hydraulic Sewing Machine

4.8.2. Further practical usage of Hydraulic Actuation Techniques applied to Sewing Machines.

PART 2

THE CONSTRUCTION AND TESTING OF A HYDRAULIC LOCKSTITCH SEWING MACHINE

4. HYDRAULIC SEWING MACHINE

4.1. Concept of a Hydraulic Sewing Machine

A second application using miniature hydraulic actuators and rotary valves in the field of textile engineering was for the needle and thread take-up mechanism of a sewing machine. A mechanical lockstitch sewing machine basically consists of a rotating bobbin and two linear motions of the needle and the thread take-up, thus it appeared to be ideally suited for the application of miniature hydraulic actuators and rotary valves.

The basic techniques for designing the miniature hydraulic actuators and rotary valve had been established in earlier work, consequently the project involved applying the expertise gained in the field of miniature hydraulic actuation techniques to an industrial sewing machine.

The main purpose in building a hydraulic sewing machine was to investigate new techniques in stitch formation for automatic machines. It was envisaged that a hydraulic sewing head would offer more flexibility than existing mechanical sewing actions in the following areas:-

(i) A smaller more compact sewing unit could be designed for automated machines.

(ii) A decoupling of the rigid driving link between the hook and needle drive shafts would be possible.

(iii) It would enable the mechanics of right and left hand machines to be simplified.

The decoupling of the two main driving shafts would enable simpler control techniques to be used on automated machines where the machine is to be moved on co-ordinate axes The use of hydraulic actuators would reduce (to four flexible pipes) the linkage between the top and bottom sections of the sewing machine, provided a register could be maintained between the needle and the hook. Greater manoeuverability could be obtained by considering the needle as a rotatable axis for the base of the machine, providing that the eye of the needle could present a loop of thread that could be picked up by the hook. This would simplify the control system required for contour seaming.

Some types of automatic sewing stations sew two sides of a garment at a single pass. This process utilises two machines, that is, a right hand and a left hand machine. While the mechanics for the right hand machine are well established, the reversal of particular components to make a left hand machine is relatively difficult. It is envisaged that using hydraulic techniques this reversal procedure will be simplified.

Consequently, in collaboration with Courtauld's Engineers, a decision to build a lockstitch sewing machine was made. The lockstitch machine was chosen because its basic mode of operation was fundamental to all types of sewing machine. This prototype machine was to show that a lockstitch could be produced by using hydraulic devices and also provide a basis upon which to assess future developments and digressions.

4.2. Selecting a Machine.

A survey of various types of industrial lockstitch machines was made by visiting the major manufacturer's sales offices situated in the Leicestershire area. The machines viewed were all basically similar in operation, but a final selection had to take into account the following features:-

(i) The needle and thread take-up mechanism must be contained in the arm of the sewing machine.

(ii) The shuttle and fabric feed mechanism had to be completely housed in the base of the machine.

(iii) The casing of the machine must be in two separable castings.

(iv) The stitch length adjustment had to be accessible and provide a fabric feed reversal facility.

(v) The machine had to be in current production with spare parts readily available.

The machine that fitted this specification was the Singer 660 Lockstitch machine. A complete sewing machine with stand and motor was purchased to provide the starting point for the project.

4.3. The Mechanical Sewing Machine

Prior to any modification to the sewing machine, a period of time was spent becoming familiar with the basic sewing machine mechanics. This involved operating the machine and investigating the effects of bobbin and needle tensions, and also becoming familiar with the setting up and timing procedures. In order to obtain detailed information concerning the needle motion and thread take-up mechanism, a series of scale drawings showing the crank at 30° intervals were constructed. From these, vector diagrams of velocity and acceleration were drawn. These vector diagrams can be seen in Figures 52 - 64, and a summary of the results is given in table 4. These results assume a crank speed of one radian per second which can be scaled to suit any desired crank input speed.

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	No. OFF

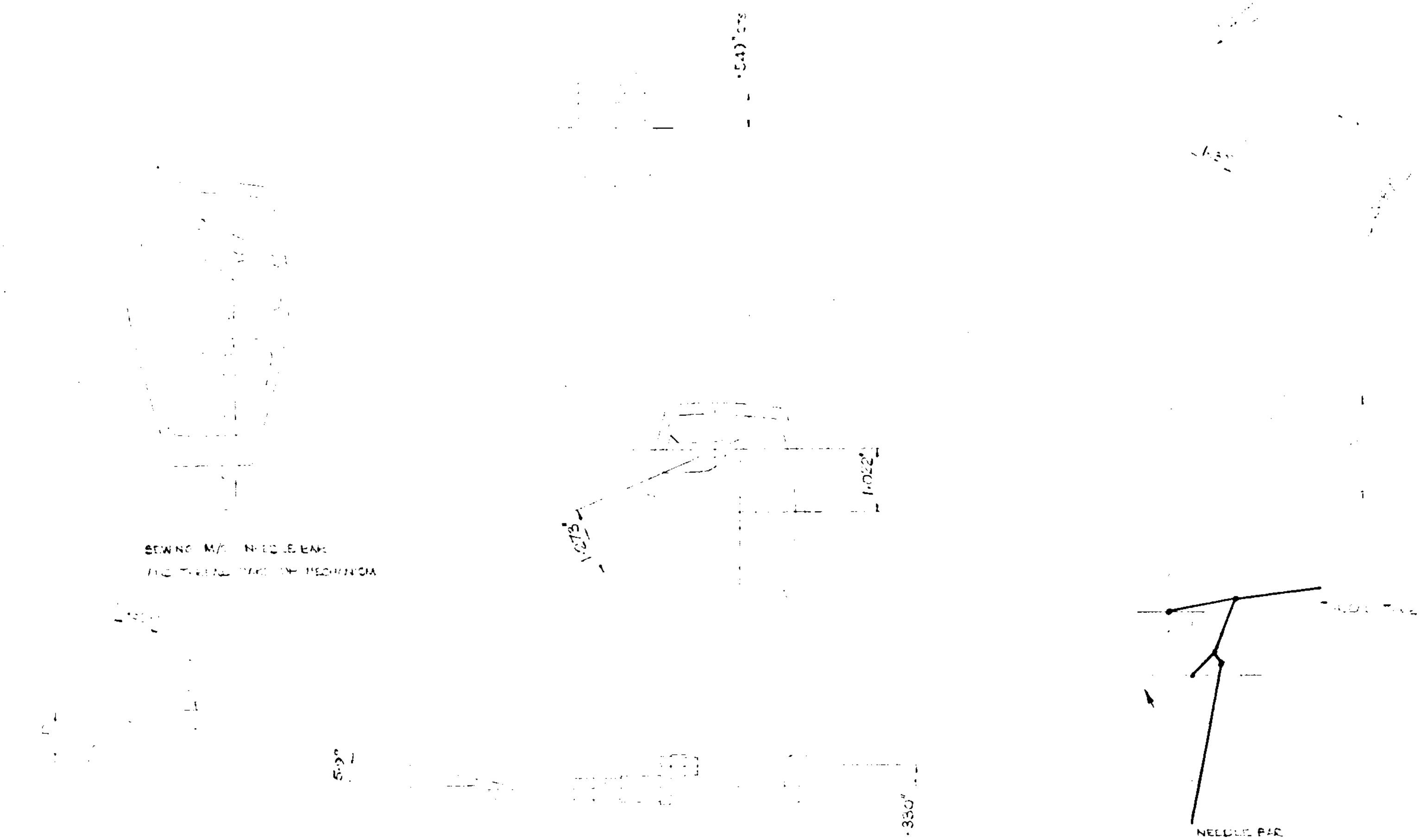

FIG. 52

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A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED. REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED. LIMITS CONFORM TO B.S. 1916 PT. I. 1953 UNILATERAL SYSTEM. MACHINING SYMBOL ✓ SURFACE ROUGHNESS TO BE INDEX NUMBERS

MATERIAL | SCALE

PRODUCT | TITLE

DRAWN | SCHEDULE No | PART No

TRACED | DRG No

APPD | DATE

SIGN | B DATE MOD | SIGN | C DATE MOD | SIGN | D DATE MOD | SIGN | E DATE MOD | SIGN | F DATE MOD | SIGN | GROUP

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

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= 1·180"

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[illegible] ω. OC = ·549 IN/SEC PERP TO OC

SCALE 1 CM ≡ ·1 IN/SEC.

[illegible] = ·265 IN/SEC ALONG ON

[illegible] = ·550 IN/SEC PERP TO BN

[illegible] = ·678 IN/SEC PERP TO CS

[illegible] = ·250 IN/SEC PERP TO PS

[illegible] = ·040 IN/SEC PERP TO CT

[illegible] ·540 IN/SEC PERP TO ST

[illegible] IN/SEC AT [illegible] TO ON

[illegible] IN/SEC [illegible] TO ON

CENT. ACC^n OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·211 IN/SEC^2 TOWARDS O

CENT. ACC^n OF C RELATIVE TO O = $\frac{(oc)^2}{OC}$ = ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT. ACC^n OF N RELATIVE TO B = $\frac{(bn)^2}{BN}$ = ·102 IN/SEC^2 TOWARDS B

CENT. ACC^n OF S RELATIVE TO C = $\frac{(sc)^2}{SC}$ = ·471 IN/SEC^2 TOWARDS C

CENT. ACC^n OF S RELATIVE TO P = $\frac{(sp)^2}{SP}$ = ·046 IN/SEC^2 TOWARDS P

CENT. ACC^n OF T RELATIVE TO C = $\frac{(tc)^2}{TC}$ = 1·084 IN/SEC^2 TOWARDS C

CENT. ACC^n OF T RELATIVE TO S = $\frac{(ts)^2}{TS}$ = ·782 IN/SEC^2 TOWARDS S

ACC^n OF T RELATIVE TO O = 2·65 IN/SEC^2 ALONG [illegible]

ACC^n OF N RELATIVE TO O = ·456 IN/SEC^2 ALONG ON

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MATERIAL | SCALE | PRODUCT | TITLE

A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED. REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED. LIMITS CONFORM TO B.S. 1916 PT. 1 1953 UNILATERAL SYSTEM. MACHINING SYMBOL ✓ SURFACE ROUGHNESS IN BY INDEX NUMBERS

DRAWN | SCHEDULE No. | PART No. | TRACED | DRG. No. | APPD. | DATE

B DATE MOD SIGN | C DATE MOD SIGN | D DATE MOD SIGN | E DATE MOD SIGN | F DATE MOD SIGN | GROUP

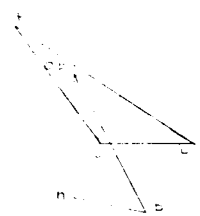
FIG. 53

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	No OFF

VELOCITY DIAGRAM



LENGTH OF LINKS

CENT ACC^N OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·611 IN/SEC² TOWARDS O

CENT ACC^N OF C RELATIVE TO O = $\frac{(oc)^2}{OC}$ = ·549 IN/SEC² TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC²

CENT ACC^N OF N RELATIVE TO B = $\frac{(bn)^2}{BN}$ = ·031 IN/SEC² TOWARDS B

CENT ACC^N OF S RELATIVE TO C = $\frac{(sc)^2}{SC}$ = ·144 IN/SEC² TOWARDS C

CENT ACC^N OF S RELATIVE TO P = $\frac{(sp)^2}{SP}$ = ·071 IN/SEC² TOWARDS P

CENT ACC^N OF T RELATIVE TO C = $\frac{(tc)^2}{TC}$ IN/SEC² TOWARDS C

CENT ACC^N OF T RELATIVE TO S = $\frac{(ts)^2}{TS}$ IN/SEC² TOWARDS S

... = ·611 IN/SEC PERP TO OB

... = ·549 IN/SEC PERP TO OC

1 CM ≡ ·1 IN/SEC

·48 IN/SEC ALONG ON

·290 IN/SEC PERP TO BN

·375 IN/SEC PERP TO CS

IN/SEC PERP TO PS

·570 IN/SEC PERP TO ST

IN/SEC PERP TO CT

IN/SEC AT ... TO ON

IN/SEC PAR TO ON

ACC^N OF T RELATIVE TO O = IN/SEC² AT ... TO ON

ACC^N OF T RELATIVE TO O = IN/SEC² PAR TO ON

ACC^N OF N RELATIVE TO O = IN/SEC² ALONG ON

ACCELERATION DIAGRAM

LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY

MATERIAL SCALE

PRODUCT TITLE

DRAWN

TRACED

DRG No J ... 03

B DATE MOD SIGN C DATE MOD SIGN D DATE MOD SIGN E DATE MOD SIGN F DATE MOD SIGN GROUP

123

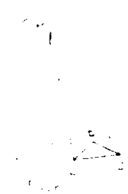
FIG. 54

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	No. OFF

VELOCITY DIAGRAM



ACCELERATION DIAGRAM

VELOCITY OF POINTS

VEL. OF B REL. TO O = ·211 IN/SEC PERP TO OB

VEL. OF C REL. TO O = ·540 IN/SEC PERP TO OC

SCALE 1 CM ≡ ·1 IN/SEC

·212 IN/SEC ALONG ON

·022 IN/SEC PERP TO BN

·55 IN/SEC PERP TO CS

·500 IN/SEC PERP TO PS

·105 IN/SEC PERP TO ST

·120 IN/SEC PERP TO CT

·430 IN/SEC AT 22° TO ON

·410 IN/SEC PAR TO ON

CENT ACC^N OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·31 IN/SEC^2 TOWARDS O

CENT ACC^N OF C RELATIVE TO O = $\frac{v^2}{r} = \frac{(oc)^2}{OC}$ = ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT ACC^N OF N RELATIVE TO B = $\frac{(bn)^2}{BN}$ = ·0003 IN/SEC^2 TOWARDS B

CENT ACC^N OF S RELATIVE TO C = $\frac{(sc)^2}{SC}$ = ·055 IN/SEC^2 TOWARDS C

CENT ACC^N OF S RELATIVE TO P = $\frac{(sp)^2}{SP}$ = ·217 IN/SEC^2 TOWARDS P

CENT ACC^N OF T RELATIVE TO C = $\frac{(tc)^2}{TC}$ = ·006 IN/SEC^2 TOWARDS C

CENT ACC^N OF T RELATIVE TO S = $\frac{(ts)^2}{TS}$ = ·006 IN/SEC^2 TOWARDS S

ACC^N OF T RELATIVE TO O = ·571 IN/SEC^2 AT 7° TO ON

ACC^N OF T RELATIVE TO O = ·525 IN/SEC^2 PAR TO ON

ACC^N OF N RELATIVE TO O = ·102 IN/SEC^2 ALONG ON

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A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED. REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED. LIMITS CONFORM TO B.S. 1916 PT 1 1953 UNILATERAL SYSTEM. MACHINING SYMBOLS ... SURFACE ROUGHNESS ... NUMBERS

MATERIAL | SCALE | PRODUCT | TITLE | DRAWN | TRACED | SCHEDULE No | PART No | DRG No

JDG 204

A DATE SIGN | B DATE MOD SIGN | C DATE MOD SIGN | D DATE MOD SIGN | E DATE MOD SIGN | F DATE MOD SIGN | GROUP

FIG. 55

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No	DESCRIPTION OF PART	MATERIAL	REMARKS	No.

VELOCITY DIAGRAM

ACCELERATION DIAGRAM

CENT ACC^N OF B RELATIVE TO O $= \frac{v^2}{r} = \frac{(ob)^2}{OB} = \cdot 311$ IN/SEC^2 TOWARDS O

CENT ACC^N OF C RELATIVE TO O $= \frac{(oc)^2}{OC} = \cdot 549$ IN/SEC^2 TOWARDS O

SCALE 1 CM $\equiv \cdot 1$ IN/SEC^2

CENT ACC^N OF N RELATIVE TO B $= \frac{(bn)^2}{BN} = \cdot 038$ IN/SEC^2 TOWARDS B

CENT ACC^N OF S RELATIVE TO C $= \frac{(sc)^2}{SC} = \cdot 002$ IN/SEC^2 TOWARDS C

CENT ACC^N OF S RELATIVE TO P $= \frac{(sp)^2}{SP} = \cdot 275$ IN/SEC^2 TOWARDS P

CENT ACC^N OF T RELATIVE TO C $= \frac{(tc)^2}{TC} = \cdot 005$ IN/SEC^2 TOWARDS C

CENT ACC^N OF T RELATIVE TO S $= \frac{(ts)^2}{TS} = \cdot 005$ IN/SEC^2 TOWARDS S

ACC^N OF T RELATIVE TO O $= \cdot 247$ IN/SEC^2 AT 11° TO ON

ACC^N OF T RELATIVE TO O $= \cdot 241$ IN/SEC^2 PAR TO ON

ACC^N OF N RELATIVE TO O $= \cdot 268$ IN/SEC^2 ALONG ON

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A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED. REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED. LIMITS CONFORM TO B.S. 1916 PT. 1 1953 UNILATERAL SYSTEM. MACHINING SYMBOL ∇ SURFACE ROUGHNESS TO B.S. INDEX NUMBERS

MATERIAL SCALE

PRODUCT TITLE

DRAWN SCHEDULE No PART No

TRACED DRG No JDS

APPD DATE

A DATE SIGN B DATE MOD SIGN C DATE MOD SIGN D DATE MOD SIGN E DATE MOD SIGN F DATE MOD SIGN GROUP

125

FIG. 56

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	No OFF

VELOCITY DIAGRAM

ACCELERATION DIAGRAM

CENT ACC^N OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·611 IN/SEC^2 TOWARDS O

CENT ACC^N OF C RELATIVE TO O = $\frac{(oc)^2}{OC}$ = ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT ACC^N OF N RELATIVE TO B = $\frac{(bn)^2}{BN}$ = ·107 IN/SEC^2 TOWARDS B

CENT ACC^N OF S RELATIVE TO C = $\frac{(sc)^2}{SC}$ = ·020 IN/SEC^2 TOWARDS C

CENT ACC^N OF S RELATIVE TO P = $\frac{(sp)^2}{SP}$ = ·195 IN/SEC^2 TOWARDS P

CENT ACC^N OF T RELATIVE TO C = $\frac{(tc)^2}{TC}$ = ·047 IN/SEC^2 TOWARDS C

CENT ACC^N OF T RELATIVE TO S = $\frac{(ts)^2}{TS}$ = ·086 IN/SEC^2 TOWARDS S

ACC^N OF T RELATIVE TO O = ·245 IN/SEC^2 AT 70° TO ON

ACC^N OF T RELATIVE TO O = ·092 IN/SEC^2 PAR TO ON

ACC^N OF N RELATIVE TO O = ·614 IN/SEC^2 ALONG ON

·21 IN/SEC^2 PERP TO EB

·549 IN/SEC^2 PERP TO OC

·241 IN/SEC^2 ALONG ON

·543 IN/SEC^2 PERP TO BN

·123 IN/SEC^2 PERP TO SC

·472 IN/SEC^2 PERP TO PS

IN/SEC^2 PERP TO ST

IN/SEC^2 PERP TO CT

·307 IN/SEC^2 ALONG ON

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A GENERAL TOLERANCE OF PLUS OR MINUS ·010" IS ALLOWED ON ALL MACHINING DIMENSIONS UNLESS OTHERWISE STATED. REMOVE ALL BURRS AND SHARP EDGES UNLESS OTHERWISE STATED. LIMITS CONFORM TO B.S. 1916 PT. 1 1953 UNILATERAL SYSTEM. MACHINING SYMBOL √ SURFACE ROUGHNESS TO B.S. INDEX NUMBERS

MATERIAL | SCALE | PRODUCT | TITLE | DRAWN | TRACED | SCHEDULE No | PART No | DRG No JDG 208

A DATE SIGN | B DATE MOD SIGN | C DATE MOD SIGN | D DATE MOD SIGN | E DATE MOD SIGN | F DATE MOD SIGN | GROUP

126

FIG. 57

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No	DESCRIPTION OF PART	MATERIAL	REMARKS

VELOCITY DIAGRAM

ACCELERATION DIAGRAM

CENT ACC [illegible]

SCALE [illegible]

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MATERIAL SCALE

PRODUCT TITLE

DRAWN

TRACED

B DATE MOD SIGN C DATE MOD SIGN D DATE MOD SIGN E DATE MOD SIGN F DATE MOD SIGN GROUP

FIG 58

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	

VELOCITY DIAGRAM

ACCELERATION DIAGRAM

CENT. ACCN OF B RELATIVE TO O = [illegible] N/SEC² TOWARDS O

CENT. ACCN OF C RELATIVE TO O = [illegible] = ·549 N/SEC² TOWARDS O

SCALE 1 CM ≡ [illegible] N/SEC²

CENT. ACCN OF N RELATIVE TO B = [illegible] N/SEC² TOWARDS B

CENT. ACCN OF S RELATIVE TO C = [illegible] = ·257 N/SEC² TOWARDS C

CENT. ACCN OF S RELATIVE TO P = [illegible] N/SEC² TOWARDS P

CENT. ACCN OF T RELATIVE TO C = [illegible] = ·594 N/SEC² TOWARDS C

CENT. ACCN OF T RELATIVE TO S = [illegible] = ·40[illegible] N/SEC² TOWARDS S

ACCN OF T RELATIVE TO O = ·200 N/SEC² AT [illegible]° TO ON

ACCN OF [illegible] RELATIVE TO O = [illegible] N/SEC² PAR. TO ON

ACCN OF N RELATIVE TO O = ·542 N/SEC² ALONG ON

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MATERIAL SCALE

PRODUCT TITLE

DRAWN SCHEDULE No. PART No.

TRACED DRG. No. JDS [illegible]

APPD DATE

B DATE MOD SIGN C DATE MOD SIGN D DATE MOD SIGN E DATE MOD SIGN F DATE MOD SIGN GROUP

FIG. 59

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS

VELOCITY DIAGRAM

ACCELERATION DIAGRAM

RAD/SEC

... OB = ·211 IN/SEC PERP TO OB

... OC = ·549 IN/SEC PERP TO OC

SCALE 1 CM ≡ ·1 IN/SEC

·300 IN/SEC ALONG ON

·034 IN/SEC PERP TO BN

·648 IN/SEC PERP TO CC

·017 IN/SEC PERP TO PS

·644 IN/SEC PERP TO ST

·620 IN/SEC PERP TO CT

·661 IN/SEC AT ... TO ON

·625 IN/SEC ... TO ON

CENT ACC^N OF B RELATIVE TO O $= \frac{v^2}{r} = \frac{(ob)^2}{OB} =$ ·61 IN/SEC^2 TOWARDS O

CENT ACC^N OF C RELATIVE TO O $= \frac{(oc)^2}{OC} =$ ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT ACC^N OF N RELATIVE TO B $= \frac{(bn)^2}{BN} =$ ·031 IN/SEC^2 TOWARDS B

CENT ACC^N OF S RELATIVE TO C $= \frac{(sc)^2}{SC} =$ ·802 IN/SEC^2 TOWARDS C

CENT ACC^N OF S RELATIVE TO P $= \frac{(sp)^2}{SP} =$ ·000 IN/SEC^2 TOWARDS P

CENT ACC^N OF T RELATIVE TO C $= \frac{(tc)^2}{TC} =$ ·691 IN/SEC^2 TOWARDS C

CENT ACC^N OF T RELATIVE TO S $= \frac{(ts)^2}{TS} =$ ·492 IN/SEC^2 TOWARDS S

ACC^N OF T RELATIVE TO O = ·375 IN/SEC^2 AT 64° TO ON

ACC^N OF T RELATIVE TO O = ·165 IN/SEC^2 PAR TO ON

ACC^N OF N RELATIVE TO O = ·203 IN/SEC^2 ALONG ON

FIG. 60

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MATERIAL | SCALE | PRODUCT | TITLE | DRAWN | TRACED | SCHEDULE No. | PART No. | DRG No. | APPD | DATE

A | DATE MOD | SIGN | B | DATE MOD | SIGN | C | DATE MOD | SIGN | D | DATE MOD | SIGN | E | DATE MOD | SIGN | F | DATE MOD | SIGN | GROUP

130

FIG. 61

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

No.	DESCRIPTION OF PART	MATERIAL	REMARKS	No OFF

VELOCITY DIAGRAM

[illegible] OF LINKS

[illegible] = ·2
[illegible] = ·541"
BN = 2·750
CT = 2·8"
CS = ·474"
ST = ·477"
PS = ·415"

[illegible] 1 RAD/SEC.

[illegible] CB = ·611 IN/SEC PERP TO CB
[illegible] OC = ·549 IN/SEC PERP TO OC

[illegible] CM ≡ ·1 IN/SEC

[illegible] = ·605 IN/SEC ALONG ON
[illegible] = ·082 IN/SEC PERP TO BN
[illegible] = ·420 IN/SEC PERP TO CC
[illegible] = [illegible] IN/SEC PERP TO PS
[illegible] = [illegible] IN/SEC PERP TO ST
[illegible] = 1·080 IN/SEC PERP TO CT
[illegible] = ·000 IN/SEC AT 3° TO ON
[illegible] = ·802 IN/SEC PAR TO ON

CENT ACC^N OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·611 IN/SEC^2 TOWARDS O

CENT ACC^N OF C RELATIVE TO O = $\frac{(oc)^2}{OC}$ = ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT ACC^N OF N RELATIVE TO B = $\frac{(bn)^2}{BN}$ = ·000 IN/SEC^2 TOWARDS B

CENT ACC^N OF S RELATIVE TO C = $\frac{(sc)^2}{SC}$ = ·246 IN/SEC^2 TOWARDS C

CENT ACC^N OF S RELATIVE TO P = $\frac{(sp)^2}{SP}$ = ·022 IN/SEC^2 TOWARDS P

CENT ACC^N OF T RELATIVE TO C = $\frac{(tc)^2}{TC}$ = ·514 IN/SEC^2 TOWARDS C

CENT ACC^N OF T RELATIVE TO S = $\frac{(ts)^2}{TS}$ = ·342 IN/SEC^2 TOWARDS S

ACC^N OF T RELATIVE TO O = ·700 IN/SEC^2 AT 13° TO ON
ACC^N OF T RELATIVE TO O = ·692 IN/SEC^2 PAR TO ON
ACC^N OF N RELATIVE TO O = ·165 IN/SEC^2 ALONG ON

ACCELERATION DIAGRAM

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MATERIAL | SCALE | PRODUCT | TITLE | DRAWN | TRACED | SCHEDULE No | PART No | DRG No JDG 210

A DATE SIGN | B DATE MOD SIGN | C DATE MOD SIGN | D DATE MOD SIGN | E DATE MOD SIGN | F DATE MOD SIGN | GROUP

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

CENT. ACC^{n} OF B RELATIVE TO O = $\frac{v^2}{r} = \frac{(ob)^2}{OB}$ = ·6.1 IN/SEC^2 TOWARDS O

CENT. ACC^{n} OF C RELATIVE TO O = $\frac{(oc)^2}{OC}$ = ·549 IN/SEC^2 TOWARDS O

SCALE 1 CM ≡ ·1 IN/SEC^2

CENT. ACC^{n} OF N RELATIVE TO B = $\left(\frac{bn}{BN}\right)^2$ = ·028 IN/SEC^2 TOWARDS B

CENT. ACC^{n} OF S RELATIVE TO C = $\left(\frac{sc}{SC}\right)^2$ = [illegible] IN/SEC^2 TOWARDS C

CENT. ACC^{n} OF S RELATIVE TO P = $\left(\frac{sp}{SP}\right)^2$ = [illegible] IN/SEC^2 TOWARDS P

CENT. ACC^{n} OF T RELATIVE TO C = $\left(\frac{tc}{TC}\right)^2$ = [illegible] IN/SEC^2 TOWARDS C

CENT. ACC^{n} OF T RELATIVE TO S = $\left(\frac{ts}{TS}\right)^2$ = [illegible] IN/SEC^2 TOWARDS S

ACC^{n} OF T RELATIVE TO O = 1·472 IN/SEC^2 AT [illegible] TO ON

ACC^{n} OF T RELATIVE TO O = 1·421 IN/SEC^2 PAR. TO ON

ACC^{n} OF N RELATIVE TO O = [illegible] IN/SEC^2 ALONG ON

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MATERIAL SCALE

PRODUCT TITLE

DRAWN

TRACED

DRG No. TD3

A B C D E F GROUP

FIG. 62

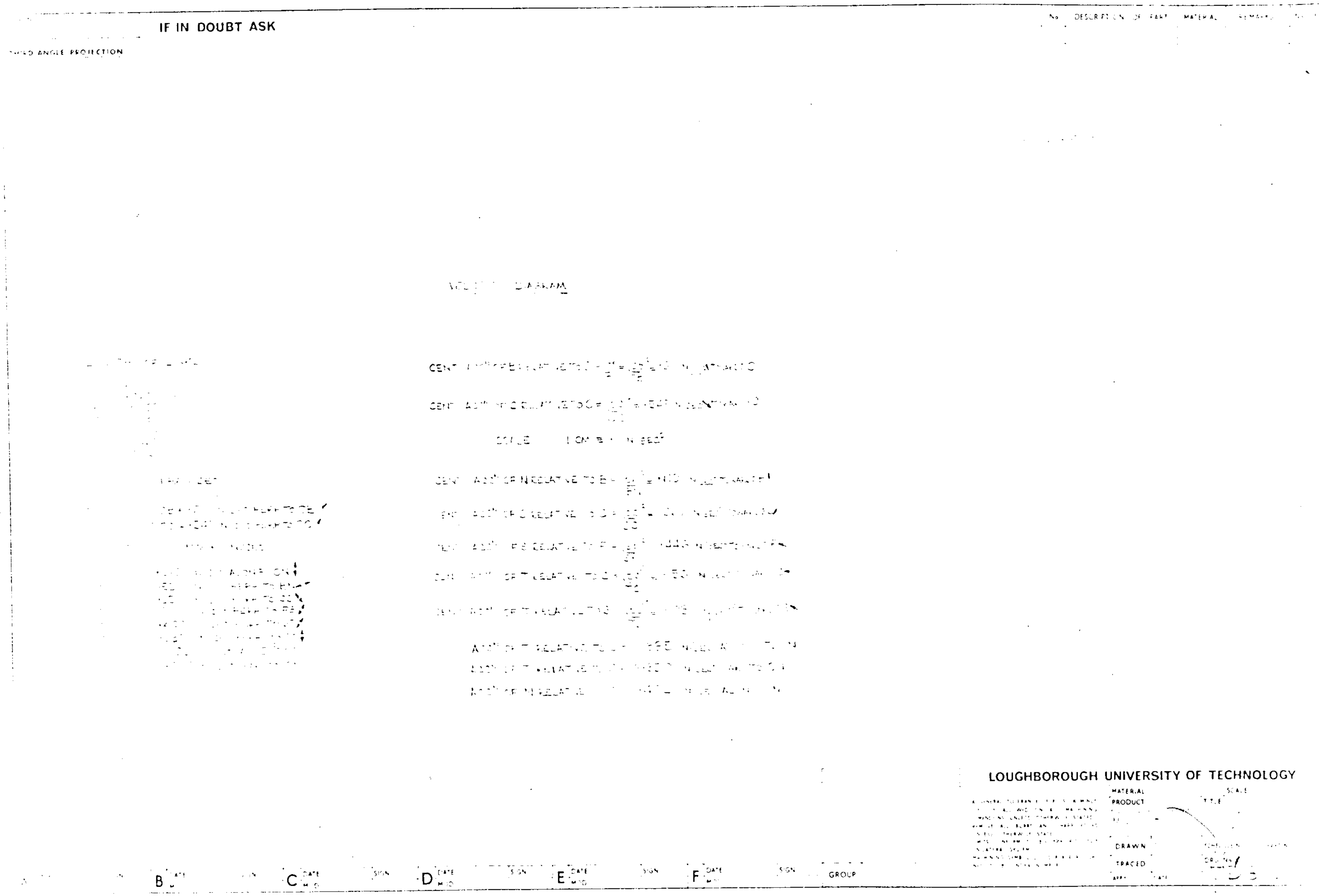
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
SCALE
PRODUCT
DRAWN
TRACED
GROUP
B
C
D
E
F

FIG. 63

IF IN DOUBT ASK

THIRD ANGLE PROJECTION

CENT ACCN OF B RELATIVE TO C = [illegible] N/SEC² TOWARDS C

CENT ACCN OF C RELATIVE TO [illegible] = ·549 N/SEC² TOWARDS C

SCALE [illegible]

CENT ACCN OF N RELATIVE TO [illegible] = ·[illegible] N/SEC² TOWARDS [illegible]

CENT ACCN OF S RELATIVE TO [illegible] = ·[illegible] N/SEC² TOWARDS [illegible]

CENT ACCN OF S RELATIVE TO P = [illegible] N/SEC² TOWARDS P

CENT ACCN OF T RELATIVE TO [illegible] = [illegible] N/SEC² TOWARDS [illegible]

CENT ACCN OF T RELATIVE TO [illegible] = [illegible] N/SEC² TOWARDS [illegible]

ACCN OF T RELATIVE TO [illegible] = 1·[illegible] N/SEC² [illegible]

ACCN OF T RELATIVE TO [illegible]

ACCN OF N RELATIVE TO [illegible] ·[illegible] N/SEC² [illegible]

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MATERIAL | SCALE

PRODUCT | TITLE

DRAWN | SCHEDULE No | PART No

TRACED | DRG No

B DATE SIGN | C DATE MOD SIGN | D DATE MOD SIGN | E DATE MOD SIGN | F DATE MOD SIGN | GROUP

FIG. 64

TABLE 4

SUMMARY OF THE RESULTS OBTAINED FROM THE ANALYSIS OF THREAD TAKE-UP AND NEEDLE BAR MECHANISMS.

Drawing Number	Position of Crank	Needle Bar			Thread Take Up Mechanism				
		Distance of N from O in	Velocity of N with respect to O in/sec	Acceleration of N relative to O in/sec^2	Vertical Displacement of T from O (VT) in	Absolute Velocity of T with respect to O in/sec	Vertical Component of the vel. of T with respect to O in/sec	Absolute Accn. of T relative to O in/sec^2	Vertical component of T rel. to O in/sec^2
J.D.G.202	0°	-2.187	-.265	-.455	.910	+1.290	+1.200	-2.35	-2.35
J.D.G.203	30°	-2.412	-.481	-.360	2.288	+ .300	+ .191	-1.31	-1.27
J.D.G.204	60°	-2.710	-.612	-.108	2.200	- .460	- .420	- .571	-.565
J.D.G.205	90°	-3.020	-.580	+.268	2.000	- .515	- .507	- .247	-.241
J.D.G.206	120°	-3.2509	-.342	+.614	1.700	- .607	- .607	- .245	-.086
J.D.G.207	150°	-3.375	+.030	+.745	1.375	- .700	- .675	- .387	-.218
J.D.G.208	180°	-3.240	+.381	+.542	1.000	- .840	- .800	- .209	-.175
J.D.G.209	210°	-2.960	+.600	+.208	.530	- .831	- .825	+ .378	+.165
J.D.G.210	240°	-2.660	+.605	-.165	.250	-.555	- .553	+ .700	+.682
J.D.G.211	270°	-2.410	+.492	-.362	0	-.088	- .088	+1.472	1.421
J.D.G.212	300°	-2.200	+.225	-.462	.230	+1.108	+1.066	+2.385	+2.350
J.D.G.213	330°	-2.125	-.026	-.478	1.000	+1.980	+1.978	+1.382	+.188

4.4. Designing the Hydraulic Sewing Head.

Once the two motions had been analysed and the timing established, the hydraulic sewing head and rotary valve could then be designed.

4.4.1. The Sewing Head.

The design for the sewing head containing the two actuators can be seen in figure 65. The head contains two miniature hydraulic devices, both simple actuators using a hydraulic stop in the mid-position to reduce the impact at the end of the traverse. The sewing unit layout conforms with the mechanical sewing head and houses the presser foot for the fabric feed together with a thread tensioning mechanism. In order to keep the machining requirements to a minimum, all auxilliary fittings that could be utilised on the hydraulic sewing head were transferred directly from the original machine, leaving only the components shown in figures 66 and 67 to be manufactured. These components included the hydraulic pistons, cover plates, needle holder, and thread take-up arm.

4.4.2. The Rotary Valve.

The rotary valve design which can be seen in figures 68 69 and 70 accommodates the refinements established with the knitting machine rotor. Such features as an 'O' ring groove and central exhaust bore were introduced at the design stage. The valve porting and slot lengths were designed using the results obtained from the analysis of the mechanical motions. At this stage it became evident that the timing could not be based directly on the mechanical system and the switching of the hydraulic actuators had to be tailored to suit the hook geometry.

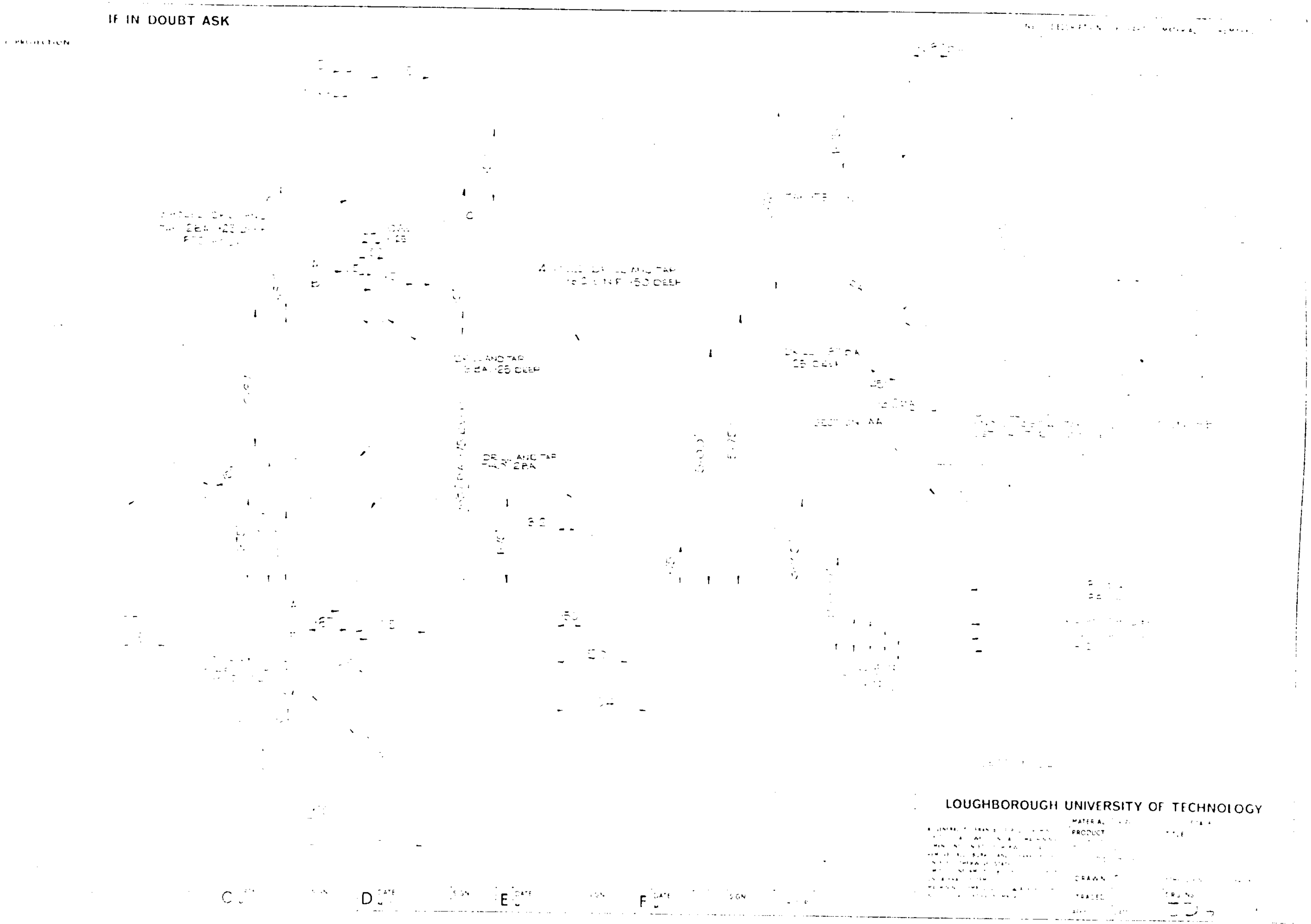
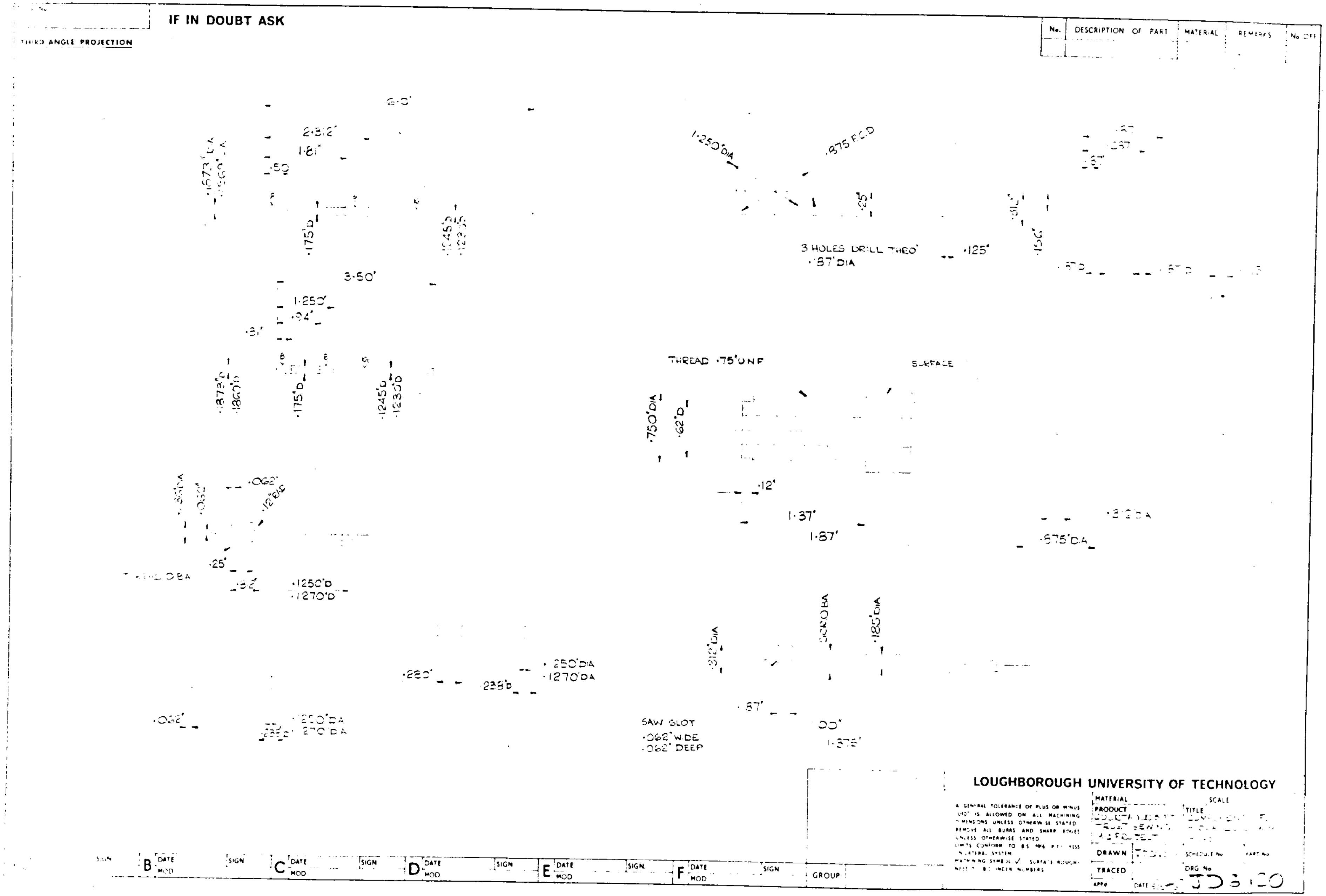


FIG. 65


IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
No. OFF
3·50'
1·250'
·94'
1·250' DIA
·875 P.C.D
3 HOLES DRILL THRO'
·187' DIA
·125'
THREAD ·75' UNF
SURFACE
·750' DIA
·62' D
·12'
1·37'
1·87'
·375' DIA
·062'
·25'
·1250' D
·1270' D
·250' DIA
·1270' DIA
SAW SLOT
·062' WIDE
·062' DEEP
·185' DIA
1·375'
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MATERIAL
SCALE
PRODUCT
TITLE
DRAWN
TRACED
DRG. No
GROUP
SIGN
DATE
MOD

FIG. 66

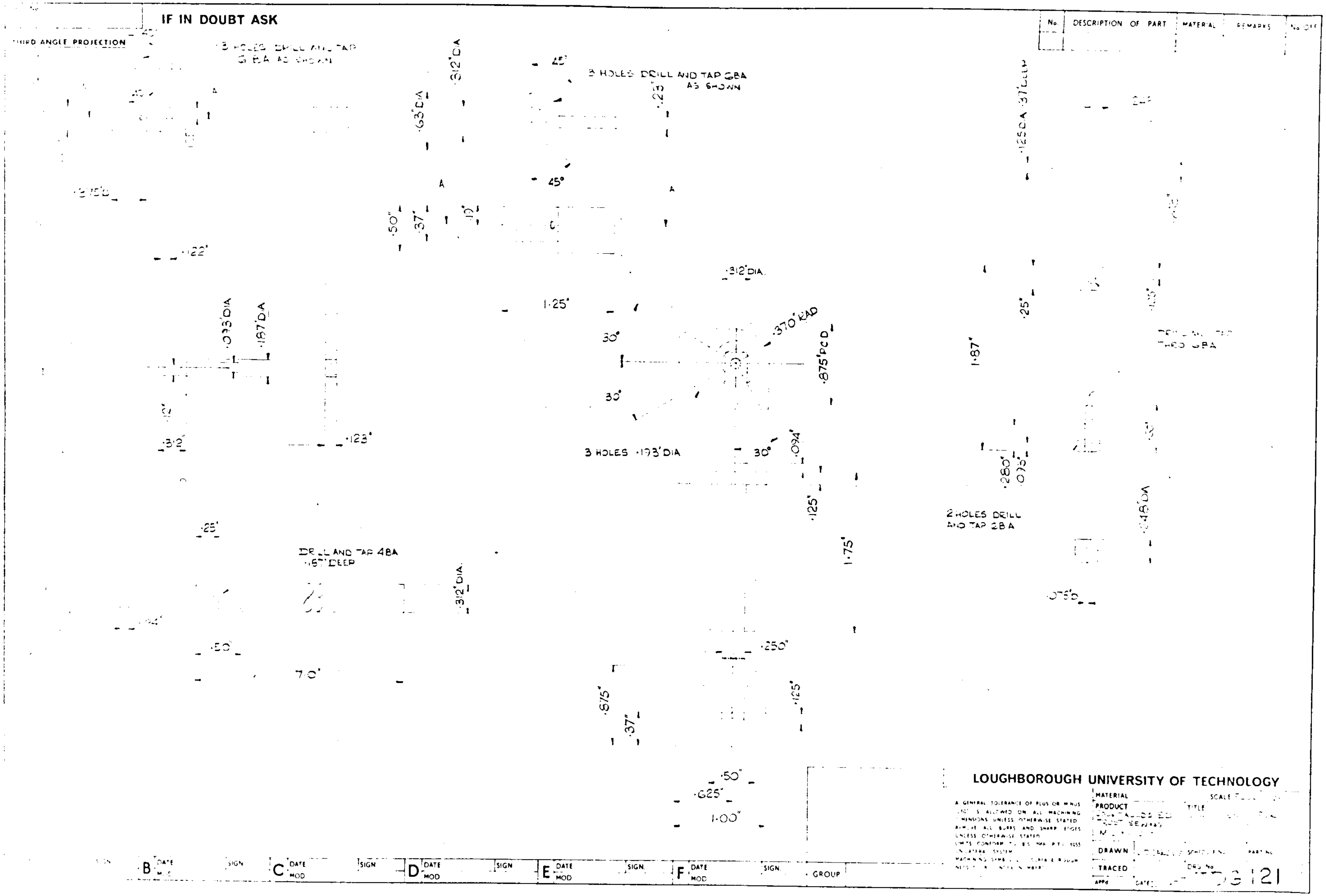
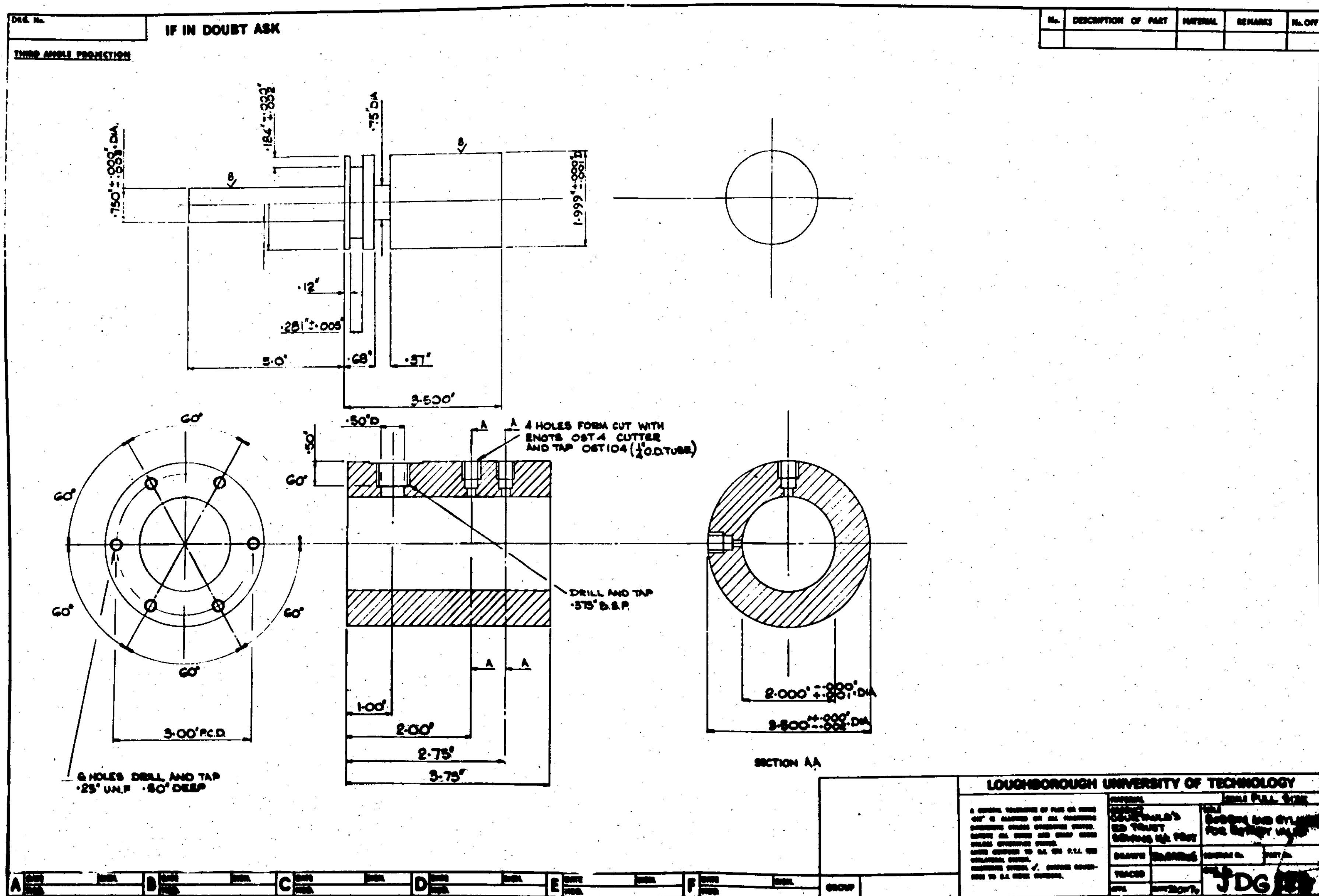

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
3 HOLES DRILL AND TAP 6BA AS SHOWN
·312" DIA
·625" DIA
·50"
·37"
45°
·122"
·093" DIA
·187" DIA
·312"
·123"
·25"
DRILL AND TAP 4BA ·187" DEEP
·312" DIA.
·50"
7·0"
1·25"
·312" DIA.
·370" RAD
·875" P.C.D.
30°
3 HOLES ·173" DIA
·094"
·125"
1·75"
·250"
·875"
·37"
·125"
·50"
·625"
1·00"
·125" DIA
·37"
·25"
1·87"
·280"
·075"
2 HOLES DRILL AND TAP 2BA
·248" DIA
·075" D
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL
PRODUCT
SCALE
TITLE
DRAWN
TRACED
DRG. No.
121
GROUP
SIGN
DATE
MOD

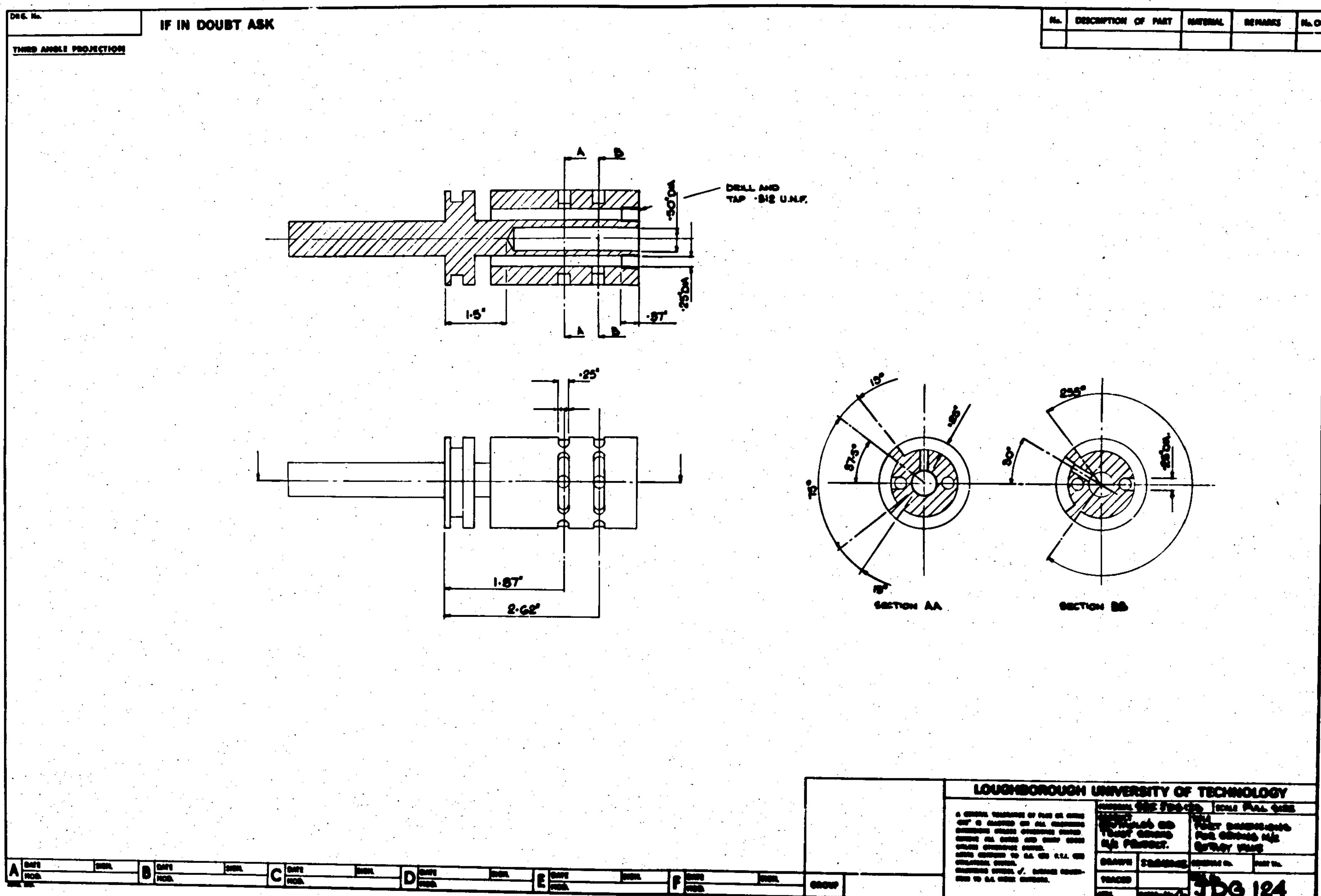
FIG. 67

139



DRG. No.
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
No. OFF
·750"+·000"/-·003 DIA.
·184"+·333/·332
·75"DIA
1·999"+·000/-·001
·12"
·281"±·005"
5·0"
·68"
·57"
3·500"
·50"D
·50"
60°
4 HOLES FORM CUT WITH ENOTS OST.4 CUTTER AND TAP OST104 (1/4"O.D.TUBE)
DRILL AND TAP ·375" B.S.P.
3·00" P.C.D.
6 HOLES DRILL AND TAP ·25" UN.F ·50" DEEP
1·00"
2·00"
2·75"
3·75"
2·000"+·000/-·001 DIA
3·500"+·000/-·002 DIA
SECTION AA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
JDG
GROUP

FIG 68



IF IN DOUBT ASK
THIRD ANGLE PROJECTION
DRILL AND TAP ·312 U.N.F.
·50" DIA.
·25" DIA.
1·5"
·87"
A
B
·25"
1·87"
2·62"
15°
87·5°
30°
255°
SECTION AA
SECTION BB
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
JDG 124

140

FIG. 69

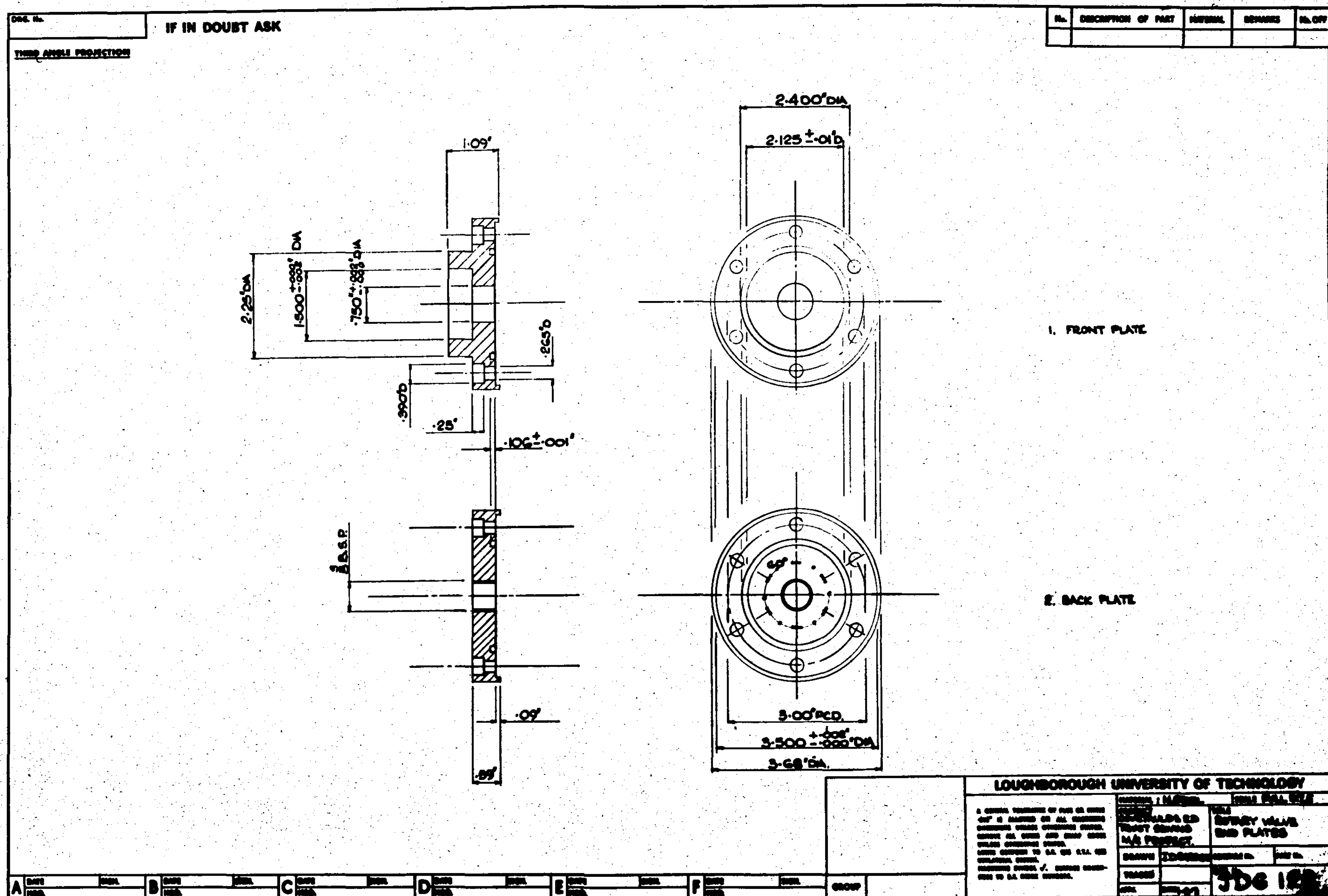

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
2·400"DIA
2·125 ±·01"D
1·09"
2·25"DIA
1·500 +·002 -·002 DIA
·750 +·002 -·003 DIA
·265"D
·390"D
·25"
·106±·001"
1/8 B.S.P.
60°
·09"
·89"
3·00"PCD.
3·500 +·002 -·000 "DIA
3·68"DIA.
1. FRONT PLATE
2. BACK PLATE
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
ROTARY VALVE
END PLATES

FIG. 7D

The shuttle rotation is geared to be twice the speed of the input shaft and so enables the basic motion of the machine to be considered as four segments each of 90° duration.

(i) $0 - 90^{\circ}$ of rotation of the main shaft; the needle must enter the fabric to enable the hook to pick up the thread passing through the eye of the needle. The thread take up mechanism must also move to create slack in the thread.

(ii) $90 - 180^{\circ}$ of rotation of the input shaft the needle must withdraw from the fabric to prevent the needle being damaged by the shuttle, the thread take-up mechanism remaining stationary enabling the yarn to pass round the shuttle.

(iii) $180 - 270^{\circ}$ of shaft rotation the actuators do not move.

(iv) $270 - 360^{\circ}$ of shaft rotation the thread take-up mechanism is powered to pull the thread off the shuttle and lock the thread into the fabric.

These motions were selected as the basis for the valve bobbin design and since the two motions were similar in duration with a phase difference of 90°, then both actuators could be driven from the same pair of slots with the required phase difference being obtained by the relative position of the take off ports.

The materials used for the sewing head actuator block was brass with Kelock 795 pistons. The rotary valve was manufactured with a Meehanite Cast Iron cylinder and a mild steel bobbin.

4.4.3. Mounting the Sewing Head and Rotary Valve.

The sole remaining task before the sewing head could be tested was to devise a method of mounting the sewing head over the shuttle and to position the rotary valve so that it could be directly coupled to the main drive shaft of the sewing machine. Since only the base of the existing sewing machine was going to be used, the obvious place to mount the rotary valve was on the end of the hook drive shaft. The drive for the machine was then transferred from the top needle drive shaft to the hook driving shaft. This was accomplished by designing a new starting wheel and pulley that could be fixed to the main machine driving shaft.

The arm for holding the hydraulic sewing head was manufactured from two pieces of rectangular steel section welded together to form an overhang very similar to that of the existing mechanical sewing machine. The general view and layout of the hydraulic sewing machine can be seen in figures 71, 72 and 73.

Figure 71 shows the machine, stand, hydraulic power pack and main driving motor. Figures 72 and 73 show details of the sewing head and rotary valve respectively.

4.5. <u>Testing the Hydraulic Actuators.</u>

4.5.1. Initial Tests and Modifications.

Prior to installing the hydraulic system onto the sewing machine, a series of bench tests were constructed to assess the performance of the hydraulic devices. At this juncture it became evident that the actuators similar inform to those used

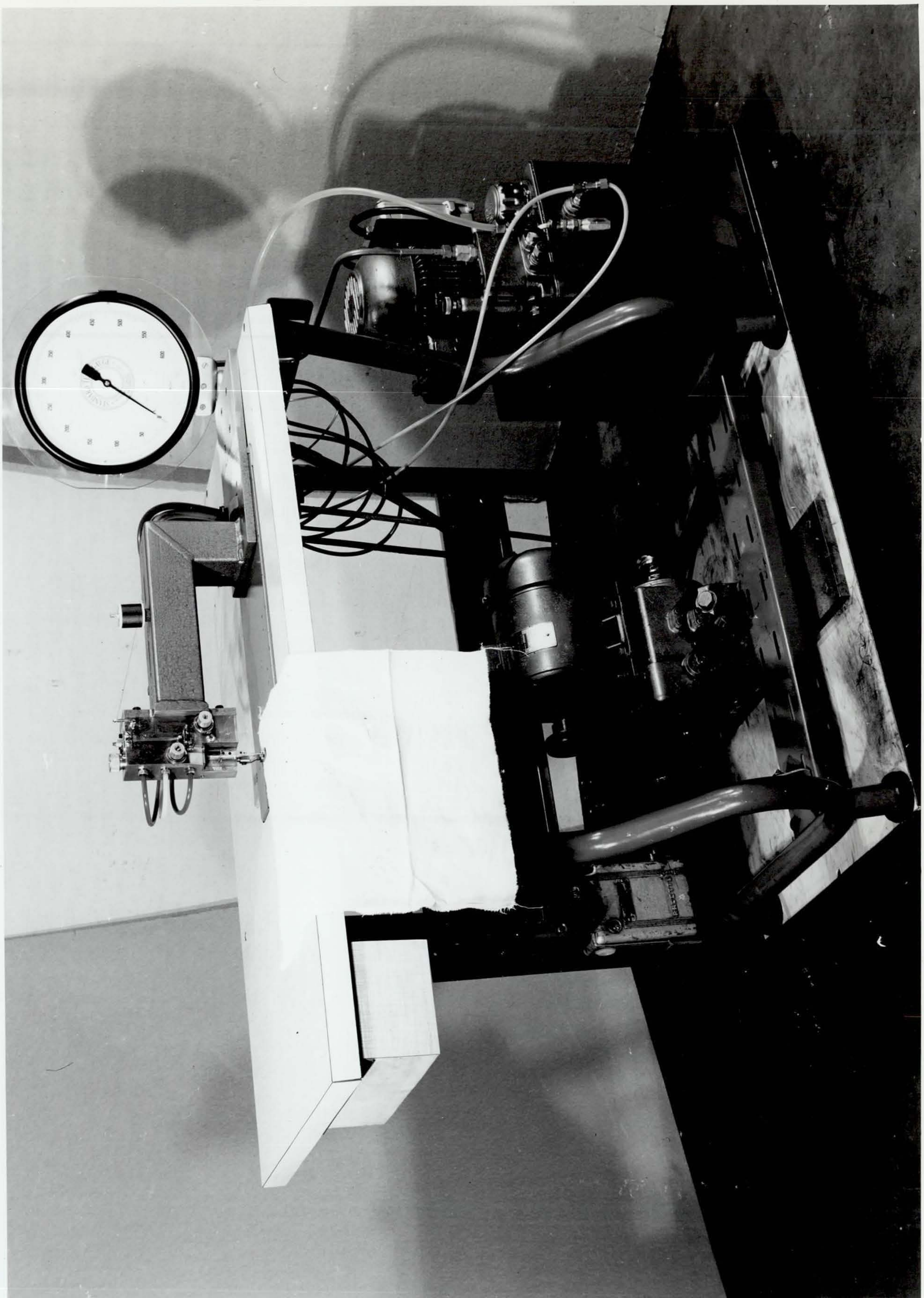


FIG. 71 GENERAL VIEW OF THE HYDRAULIC LOCKSTITCH SEWING MACHINE

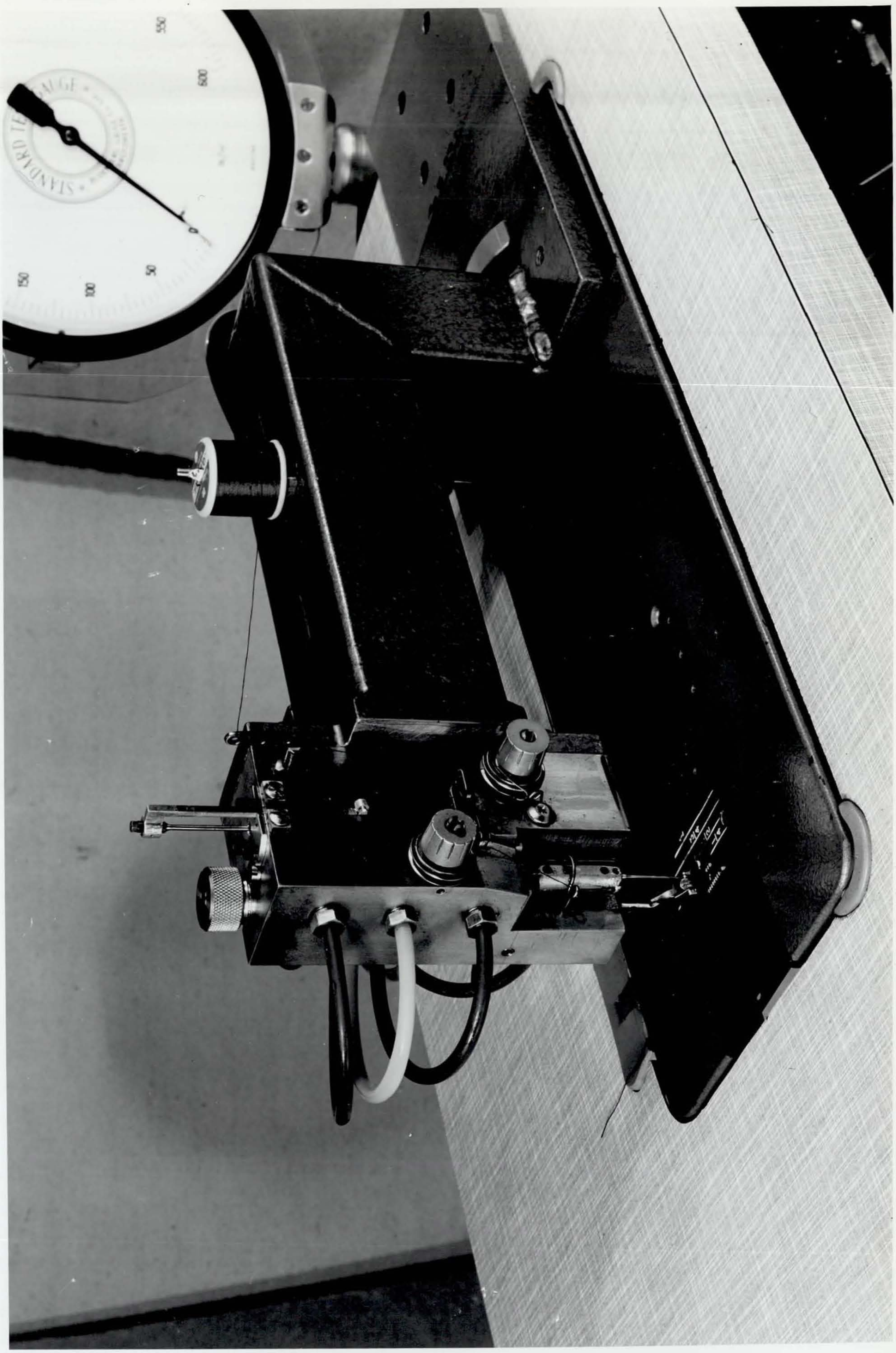


FIG. 72 HYDRAULIC SEWING HEAD

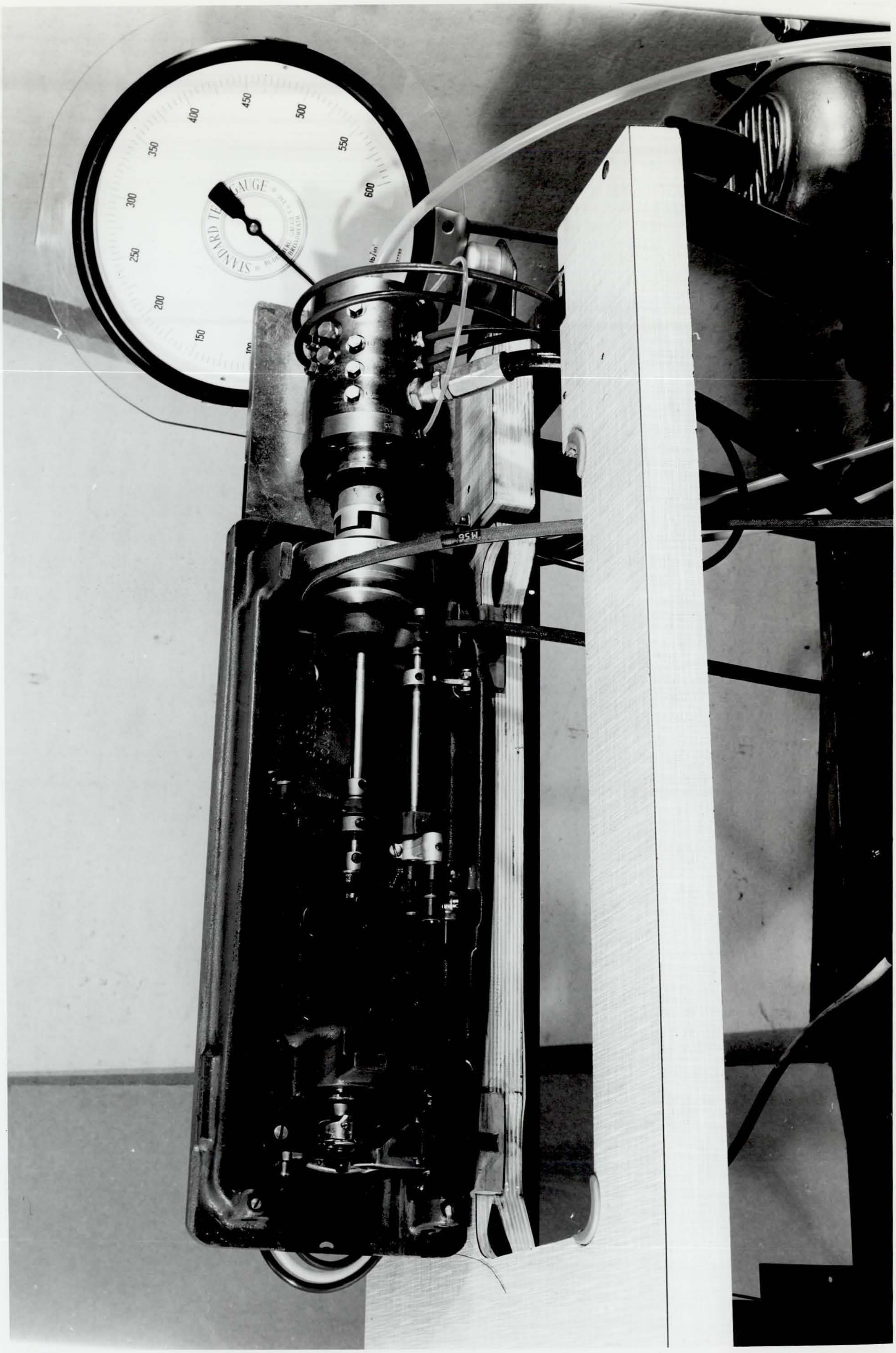


FIG. 73 ROTARY VALVE IN POSITION ON THE SEWING MACHINE

successfully on previous occasions were too large for this particular application. Increasing the size of the piston from .125" diameter to .187" diameter had introduced high impact forces at the end of the actuator movement due to the inertia of the piston rod. This problem could only be solved by reducing the mass of the moving parts, which could be achieved in three ways.

(i) Manufacture the pistons from a light alloy such as titanium. This piston could suffer from deformation on the lands.

(ii) Reduce the diameter of the piston.

(iii) Re-design the actuator to have two programmed hydraulic exhausts in place of the central hydraulic stop. This would enable the length of the piston to be reduced.

To investigate the most suitable modification a series of test actuators were manufactured. These tests showed that the piston length could be reduced by using an actuator with two hydraulic stops. These stops could not be connected directly to exhaust but required separate complementary exhausts. For testing this type of actuator the system was piped with two one-way valves linking the hydraulic stops to the main supply lines. This technique proved that the idea was feasible but for a full assessment, a new rotary valve with a set of grooves cut in the rotor was required. These new grooves would be positioned to provide a hydraulic stop only when the actuator was travelling in a particular direction. The second modification to the actuator consisted of reducing its overall diameter to

.125 inches. This proved equally successful and under the circumstances seemed the most practical solution to the problem. To change the overall diameter of the actuator entailed bushing the .187 inch diameter bores and re-machining them to .125 inches diameter as against re-designing the complete sewing head and rotary valve. Once the hydraulic sewing machine has been shown to be viable then further development of the rotary valve and actuator could well be justified, however, at this juncture it was felt that the sewing action must be verified.

4.5.2. Instrumentation and U. V. Recordings.

In order to ascertain the performance of the hydraulic devices, the system was instrumentated using pressure transducers on the oil supply lines together with a linear displacement transducer to monitor the piston rod action. The results were recorded on an ultra violet recorder using the techniques outlined previously in section 2.8. The calibration of the pressure transducers was again set to correspond to the values given in Table 2. A sample of the ultra violet recording traces obtained for the needle actuator can be seen in figure 74. Here the pressure pulses generated by the rotary valve are monitored and are shown in relation to actual piston movement. The amplitude of the needle movement was 1.25 inches and the calibration of the linear transducer showed that it provided a linear output that was acceptable for the accuracy required. The traces show the actuator operating at 11 hertz at a pressure of 200 lbf/in^2, this produces a cyclic motion of the actuator,

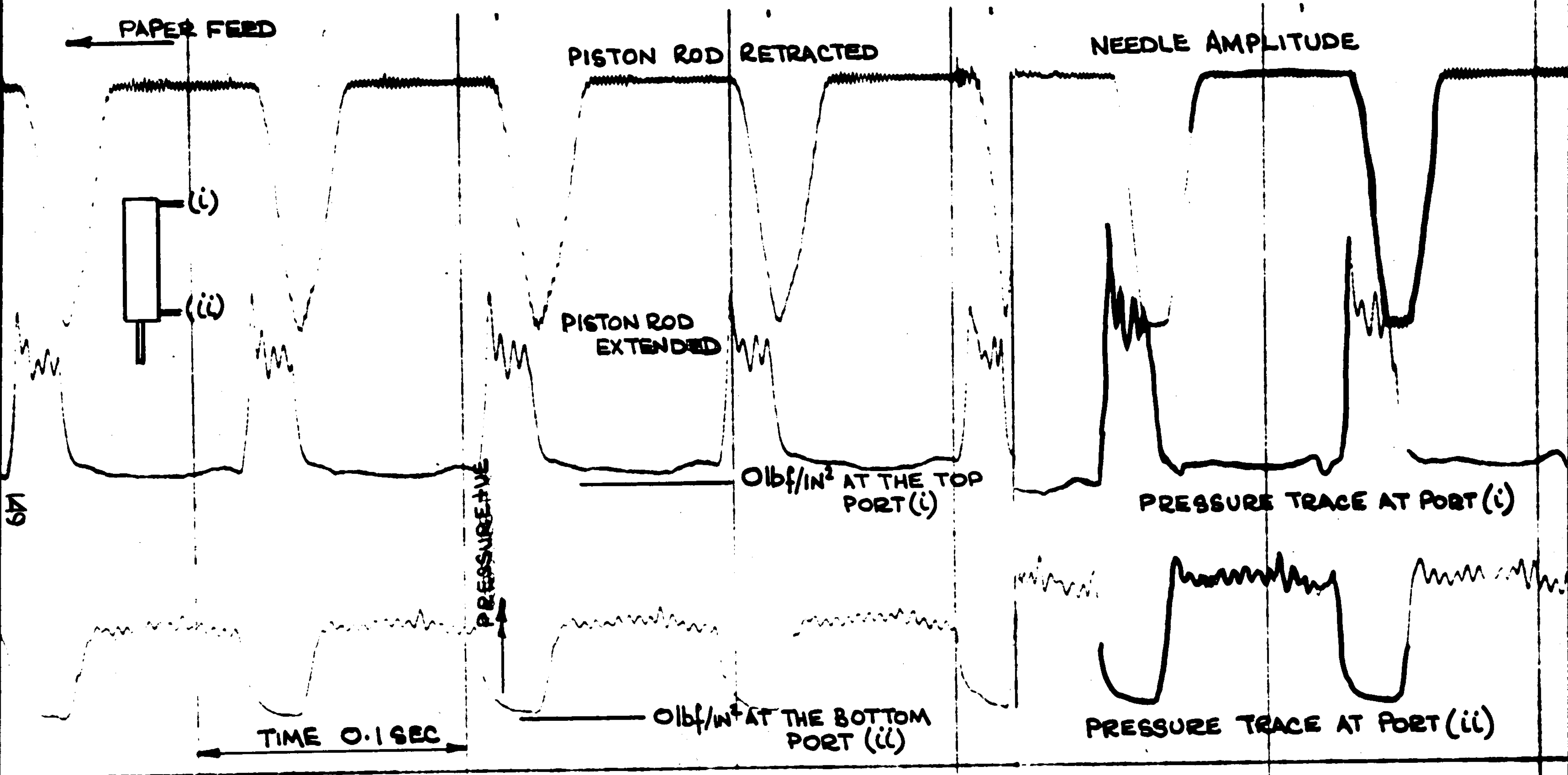


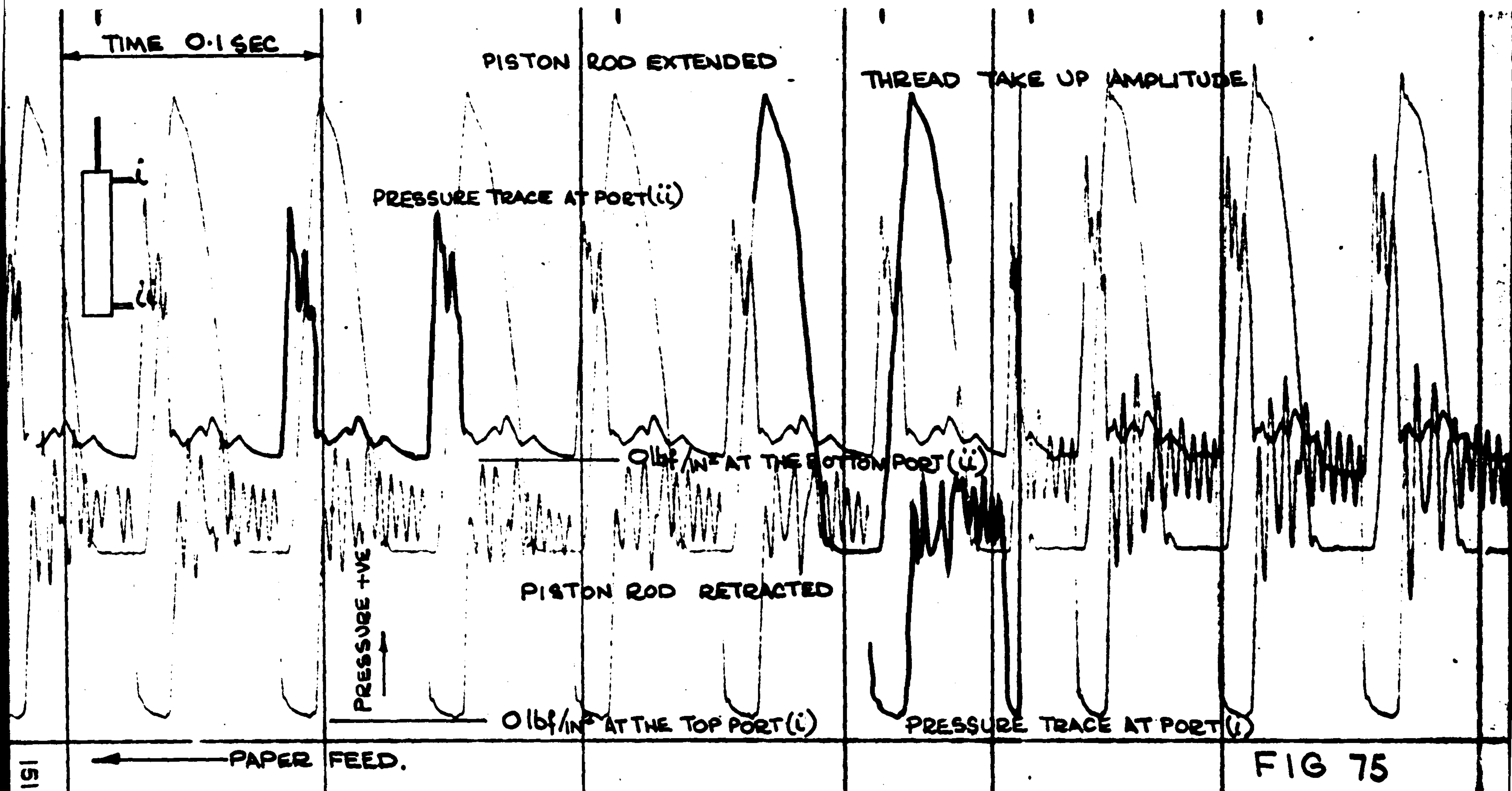
FIG. 74

U.V. RECORDING OF THE NEEDLE HYDRAULIC ACTUATOR OPERATING AT 11 C/SEC. THE TRACES SHOW (i) THE PRESSURE AT THE TOP PORT (ii) THE PRESSURE AT THE BOTTOM PORT (iii) PISTON ROD DISPLACEMENT.

but the dwell in the extended position is only a few milliseconds. However, it can be seen that if the pressure is increased to 400 lbf/in^2, the speed of the actuator is increased and consequently the dwell time which is required to enable the hook to pick up the thread, is extended. Figure 75 shows the ultra-violet recordings obtained for the thread take-up hydraulic actuator, here the amplitude of the actuator is greater, 2.312 inches, consequently an increased force is required to produce a cyclic displacement in the same period as the needle actuator. Thus the overall performance of the machine will be governed by the functioning of the thread take-up actuator. On examining the pressure traces, it can be seen that the pulses are sharp on the leading edge and when switched the port is exhausted with no back pressure being generated, due to a restriction in the flow back to tank. The leading spike on the pressure pulses could well be overshoot on the recording instrument while the cyclic pressure vibrations could be generated by the hydraulic pump in the power pack. The main purpose for taking the ultra-violet recordings was not to give analytical results but to verify that the systems were operating through the desired amplitude with no freak pressure variations.

4.5.3. Comments on the U. V. Recordings.

The traces shown in figure 76 gives the phase relationship between the needle actuator and the thread take-up actuator. Here it can be seen that the needle and thread take-up actuators are powered simultaneously. The needle passes down through the fabric into a position where the hook can pick up the yarn. At the same moment, the thread take-up actuator introduces



151

FIG 75

U.V. RECORDING OF THE THREAD TAKE-UP HYDRAULIC ACTUATOR OPERATING AT 18 C/SEC. THE TRACES SHOW (i) THE PRESSURE AT THE TOP PORT (ii) THE PRESSURE AT THE BOTTOM PORT (iii) PISTON ROD DISPLACEMENT.

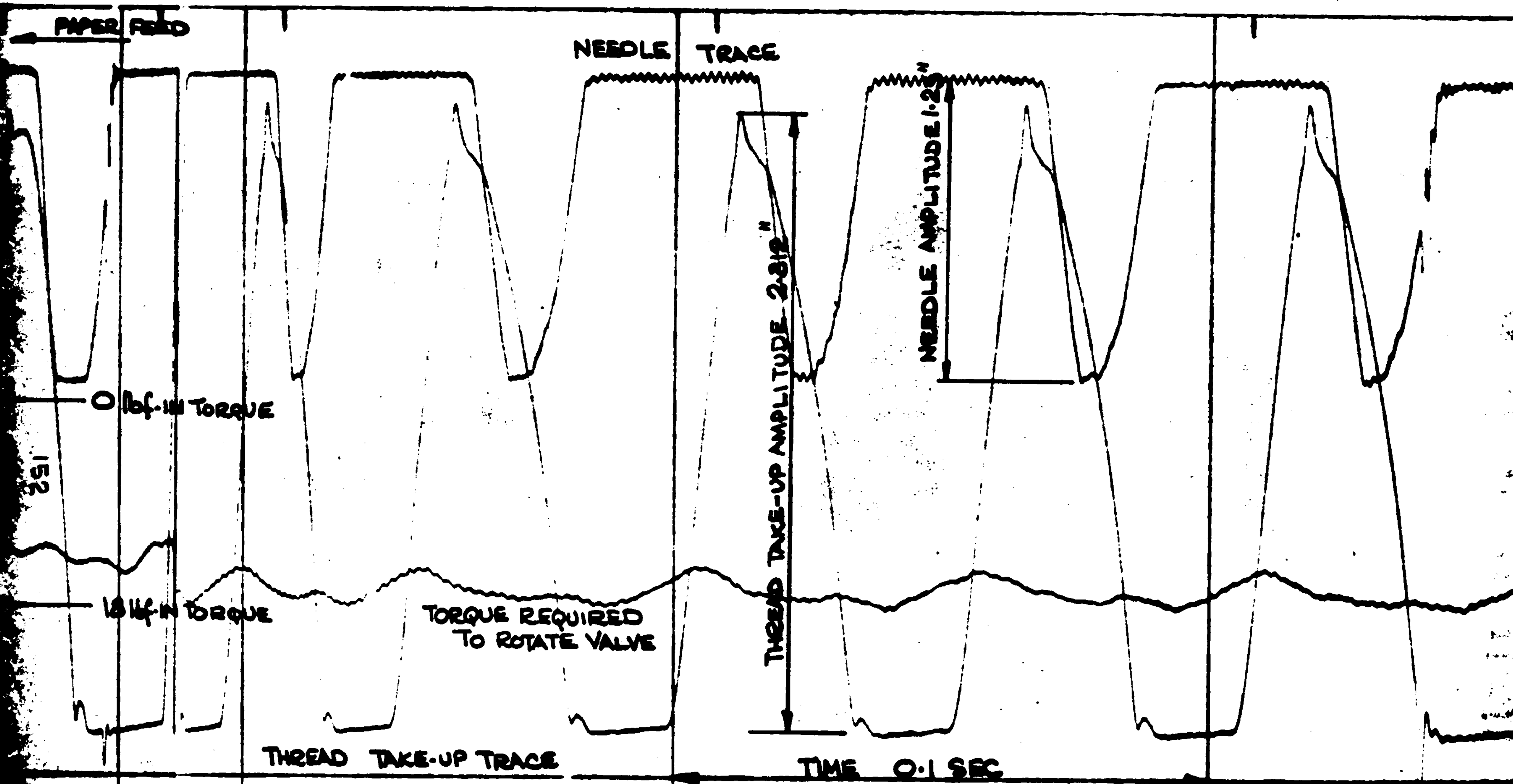


FIG. 76

U.V. RECORDING OF THE NEEDLE HYDRAULIC ACTUATOR AND THE THREAD TAKE-UP HYDRAULIC ACTUATOR TO SHOW THE PHASE RELATIONSHIP WHEN OPERATING AT 18 c/sec.

sufficient slack into the yarn to enable it to pass round the shuttle. Once the yarn has been picked up by the hook, the needle is withdrawn allowing the hook to take the yarn round the shuttle, then the thread take-up actuator is moved to pull the stitch to its pre-set tension. A useful exercise at this juncture was to compare the motions of the mechanical and hydraulic systems, so that conclusions could be drawn as to their respective merits. The relative positions of the needle actuator and the thread take-up actuator were scaled directly from the ultra-violet traces shown in figure 76. A complete cycle was divided into 24 equal strips and measurements taken at each co-ordinate to give the results in Tables 5 and 6. To enable further comparisons between the two types of drive, the average velocity and acceleration between each two adjacent points were calculated for the complete cycle.

4.5.4. Comparison of the Hydraulic and Mechanical Systems.

In order to compare the two systems, the results in Table 4 have to be scaled to the equivalent angular velocity for the drive shaft. The frequency of the hydraulic system is 18.82 cycles per second thus the values in Table 4 scaled to the same shaft speed can be seen in Table 7.

While no rigorous comparisons can be drawn, it is interesting to compare the overall features. Figures 77 and 78 show the relative displacement of the two systems. In figure 77 which shows the relative motions for the needle movement, it can be seen that the mechanical system produces a smooth cyclic motion per revolution of the input shaft while the hydraulic system moves faster and the cycle is completed in half

TABLE 5

RESULTS FOR THE NEEDLE ACTUATOR AS TAKEN FROM THE U.V. RECORDING IN FIGURE 76

EQUIVALENT CRANK ANGLE°	U.V. RECORDING VALUES IN	ACTUAL ACTUATOR POSITION IN	AVERAGE VELOCITY OF ACTUATOR IN/SEC	AVERAGE ACCELERATION OF ACTUATOR IN/SEC²
0	0	0	0	0
15	0	0	0	0
30	0	0	0	0
45	0	0	0	0
60	0	0	0	0
75	0	0	-64.70	-29276
90	.25	-.143	-258.37	-87633
105	1.25	-.714	-145.70	+50981
120	1.812	-1.036	-96.83	+22113
135	2.187	-1.250	0	+43814
150	2.187	-1.250	+32.57	+14738
165	2.062	-1.178	+96.83	+20976
180	1.687	-.964	+81.00	-7162
195	1.375	-.785	+161.08	+36235
210	.75	-.429	+178.28	+7782
225	.062	-.035	+15.82	-73506
240	0	0	0	-7162
255	0	0	0	0
270	0	0	0	0
285	0	0	0	0
300	0	0	0	0
315	0	0	0	0
330	0	0	0	0
345	0	0	0	0
360	0	0	0	0

T A B L E 6

RESULTS FOR THE THIRD TAKE-UP ACTUATOR AS TAKEN FROM THE U.V. RECORDINGS IN FIGURE 76

EQUIVALENT CRANK ANGLE	U.V. RECORDING VALVES IN	ACTUATOR DISPLACEMENT IN	AVERAGE VELOCITY OF ACTUATOR IN/SEC	AVERAGE ACCELERATION OF ACTUATOR IN/SEC^2
0	2.375	1.204	214.93	-12104
15	3.312	1.679	143.43	-32352
30	3.937	1.995	142.98	-203
45	4.562	2.312	0	-64696
60	4.562	2.312	-57.01	-25796
75	4.312	2.185	-29.86	+12285
90	4.182	2.119	-73.75	-19859
105	3.862	1.957	-68.32	+2457
120	3.562	1.805	-71.49	-1434
135	3.230	1.636	-114.93	-19656
150	2.750	1.393	-114.93	+208
165	2.250	1.140	-157.91	-19656
180	1.562	.791	-143.43	-6552
195	.937	.474	-171.94	-12900
210	.187	.094	-49.77	+55280
225	.062	.031	-14.47	+15972
240	0	0	0	-6547
255	0	0	0	0
270	0	0	0	0
285	0	0	0	0
300	0	0	0	0
315	0	0	0	+39104
330	.375	.190	86.42	57941
345	1.312	.664	214.47	13312
360	2.375	1.204	243.89	-13104

TABLE 7

RESULTS FOR THE MECHANICAL SEWING MACHINE WITH A MAIN SHAFT SPEED OF 18.82 REVOLUTIONS PER SECOND

POSITION OF CRANK	NEEDLE BAR			THREAD TAKE-UP		
	DISTANCE FROM HOOK IN	VELOCITY OF N WITH RESP.TO O IN/SEC^2	ACCN.OF N REL. TO O IN/SEC^2	VERTICAL DISP.OF. T FROM O IN.	ABSOL.VEL. OF T WITH RESP.TO O IN/SEC	ABSOL.ACCN. OF T REL. TO O IN/SEC^2
0	-.062	-31.33	-6362	1.910	+152.54	-32862
30	-.287	-56.87	-5032	2.288	+35.47	-18318
60	-.585	-72.37	-1510	2.200	-54.39	-7984
90	-.895	-68.58	3747	2.000	-60.90	-3454
120	-1.125	-40.44	8586	1.700	-71.78	-3426
150	-1.250	3.50	10418	1.375	-82.77	-5411
180	-1.115	45.05	7579	1.000	-99.33	-2922
210	-.835	70.95	2908	.530	-98.26	+5285
240	-.535	71.54	-2307	.250	-65.63	+9788
270	-.285	58.53	-5062	0	-10.40	20584
300	-.075	26.60	-6460	.230	+131.02	33351
330	0	3.07	-6684	1.000	+234.14	19325

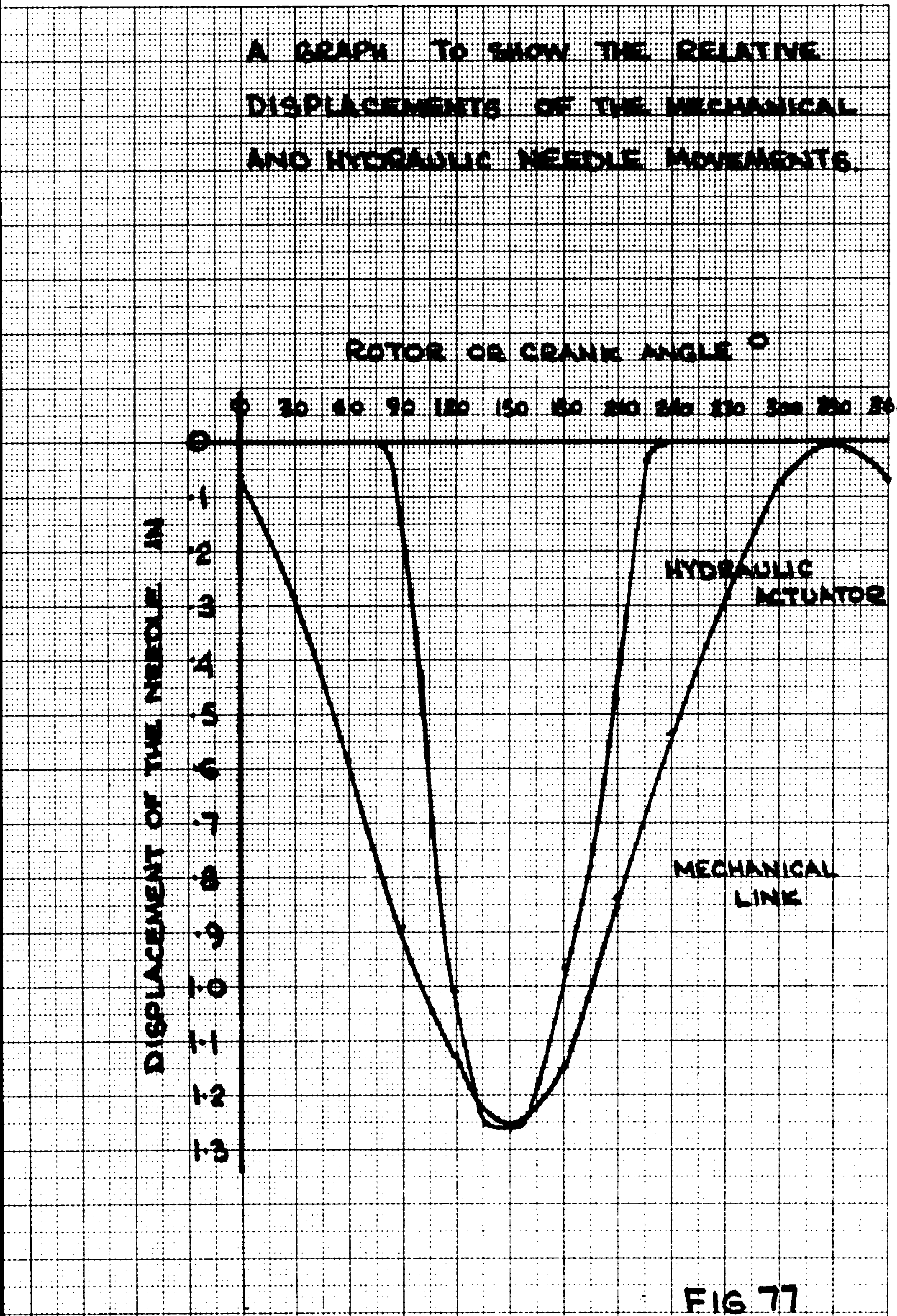

A GRAPH TO SHOW THE RELATIVE DISPLACEMENTS OF THE MECHANICAL AND HYDRAULIC NEEDLE MOVEMENTS.
ROTOR OR CRANK ANGLE °
0 30 60 90 120 150 180 210 240 270 300 330 360
DISPLACEMENT OF THE NEEDLE IN
0 ·1 ·2 ·3 ·4 ·5 ·6 ·7 ·8 ·9 1·0 1·1 1·2 1·3
HYDRAULIC ACTUATOR
MECHANICAL LINK

FIG 77

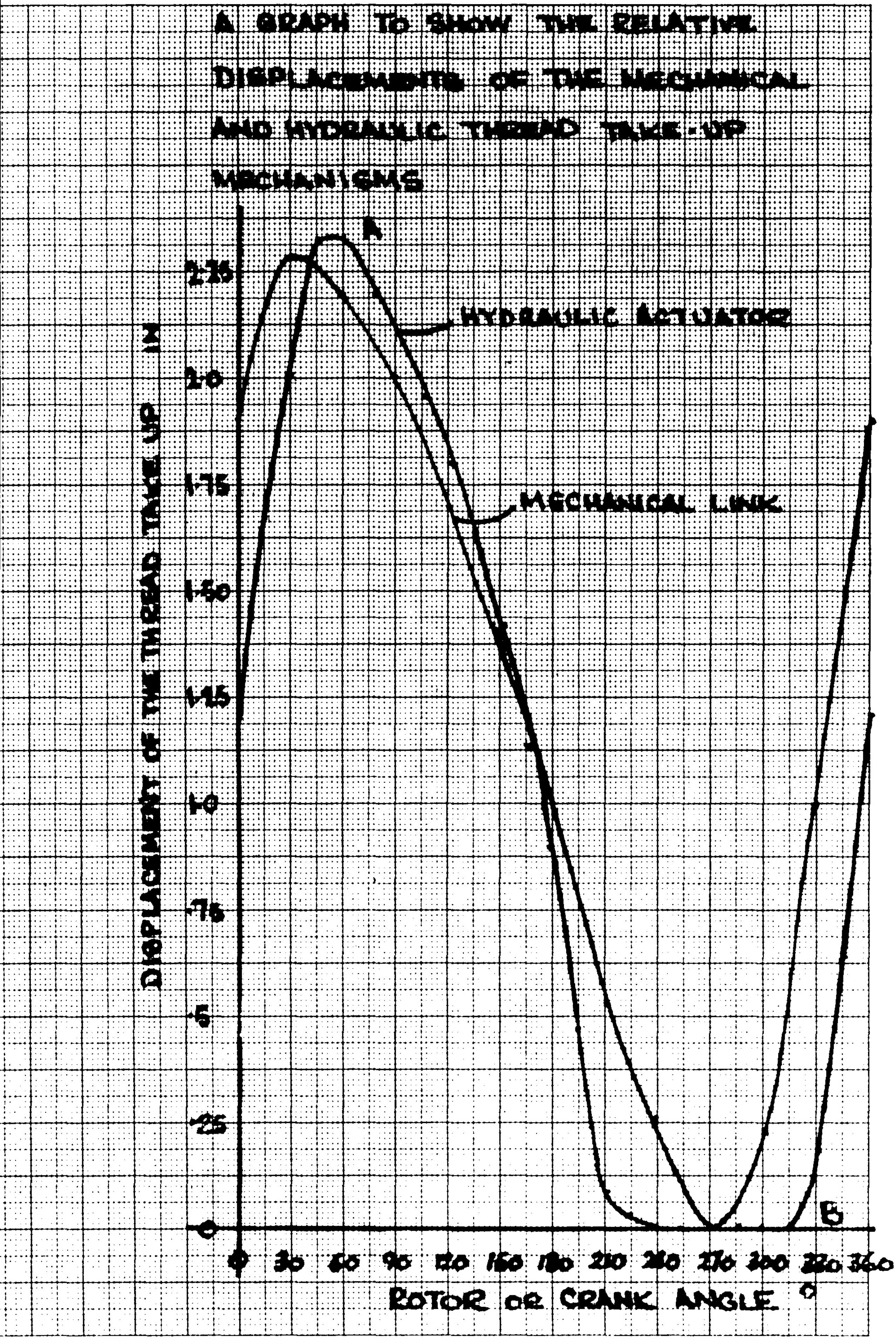

A GRAPH TO SHOW THE RELATIVE DISPLACEMENTS OF THE MECHANICAL AND HYDRAULIC THREAD TAKE-UP MECHANISMS
A
HYDRAULIC ACTUATOR
MECHANICAL LINK
B
DISPLACEMENT OF THE THREAD TAKE UP IN
2·25
2·0
1·75
1·50
1·25
1·0
·75
·5
·25
0
0
30
60
90
120
150
180
210
240
270
300
330
360
ROTOR OR CRANK ANGLE °

FIG 7B

the period. The faster movement of the hydraulic device is not a particular advantage because of heat generation and shock loading of the sewing thread, also a further advantage of the mechanical system is in being able to adjust the timing such that the hook picks up the needle thread as it is about to withdraw from the fabric, so providing a loop of thread to be picked up by the hook. However, a significant advantage of the hydraulic system is that the penetration of the needle through the fabric is not governed by the size of the slider crank mechanism and hook geometry, consequently the overall movement of the needle can be greatly reduced such that only a small percentage of the total movement takes place while not piercing the fabric. Using this criteria the needle could be re-designed on a much shorter stroke which in turn would relieve some of the more inherent disadvantages.

Figure 78 shows the relative motions of the thread take-up mechanisms. Here the hydraulic actuator motion more closely resembles the mechanical movement, but once again is faster with the dwells accuring prior to each displacement curve. The shape of the time displacement curve can be adjusted slightly by varying the supply pressure, but the points A and B on the curve are fixed by the geometry of the rotary valve.

A survey of the results shown in Tables 5, 6 and 7 show that the values of velocity and acceleration for the hydraulic devices are much higher, and more sensitive to system fluctuations. This was to be expected as the calculated values for the mechanical system were based upon a constant speed drive whilst the results for the hydraulic devices were practical results obtained in the laboratory. The main purpose for presenting these results was to enable comparisons to be drawn, and any future developments can be assessed using the data produced contrasting the two systems of motivation.

4.6. Testing the Rotary Valve.

While bench testing the hydraulic actuators, a drive for the rotary valve was obtained from a hydraulic servo-motor. This unit was standard laboratory equipment with a power rating in the region of 10 H.P. at 1200 revolutions per minute. This system, known as the Telehoist Rig, was used extensively throughout the tests performed on the rotary valves. The parameters of performance to be determined by the tests were the relationships between supply pressure, speed and torque.

4.6.1. Instrumentation.

The supply pressure to the rotary valve was assumed to be constant once an even cycling rate for the actuators had been established, thus enabling the supply pressure to be monitored using a Bourdon tube pressure gauge. The speed of rotation was scaled from a tacho-generator fixed to the back of the hydraulic motor which gave a signal voltage proportional to speed. The torque required to rotate the bobbin of the valve was the most difficult parameter to determine, this involved using a British Hovercraft Corporation torque transducer. This utilised a network of bonded foil strain gauges cemented to a high tensile torsion shaft. These gauges, which are connected in a full bridge circuit, produce an electrical bridge unbalanced proportional to torsional strain and the signal is passed via a system of silver sliprings into an indicator unit which is calibrated to give a direct dial reading in lbf - ft.

Having instrumentated the rotary valve, its performance could be assessed on recorded data and conclusions drawn as to the effect of any future modifications to the valve.

4.6.2. Modifications to Rotary Valve Design.

The first modification to the valve was to drill a series of pressure tappings into the cylinder such that pressure transducers could be used to measure transient pressures in the clearance between the bobbin and the cylinder. These recordings indicated that the pressure slots were creating high pressure areas in particular sectors of the valve, hence the second modification was to machine a series of radial grooves round the bobbin, as had been done previously on the knitting machine bobbin. These grooves helped to distribute the pressure round the bobbin thus allowing the bobbin to run more concentrically so reducing the torque requirements of the valve. The third modification was to measure the change in rotary valve performance due to the 'O' ring, originally introduced to prevent the high pressure supply oil going behind the end face of the bobbin and producing a ram action. The results obtained were most surprising in that the frictional resistance due to the 'O' ring absorbed considerably more power than the load created by the end pressure. This fact instantly necessitated a further modification whereby grooves were cut in the end caps of the bobbin to create pressure annulii. These in turn were linked to each other via an external pipe so equalising the pressure distribution.

4.6.3. Readings for Rotary Valve Performance

At this juncture a series of readings were obtained to give an indication of the rotary valve's performance with and without the 'O' ring. These results can be seen in Tables 8 and 9 together with the graphs shown in figures 79 and 80. The torque requirements for the rotary valve with an 'O' ring can be seen in figure 79 and the series of curves clearly indicate that the torque required to rotate the valve is a function of the supply pressure whilst being almost independent of rotational speed. Since the torque requirements for the valve double when using the 'O' ring seal, it must be presumed that the operating characteristics of the seal predominate, and in consequence account for the rather unexpected results. The family of curves in figure 80 (the torque requirements without the 'O' seal) are nearer to what would be expected, with the torque requirements being dependent upon rotational speed and supply pressure.

Marginal improvements in performance were obtained by changing the 'O' ring groove into a second pressure annulus in an attempt to balance the pressure distribution within the valve, and also to create a better centralising mechanism for the rotor. However, it became evident that no real improvement in performance was going to be achieved by experimental techniques, and it was at this juncture that the decision to embark upon a detailed study into the performance of rotary valves was taken (see Part 3).

T A B L E 8

RESULTS OBTAINED FROM THE SEWING MACHINE ROTARY VALVE WITH AN 'O' RING

THE RESULTS GIVE THE TORQUE LBF-IN REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS.

SPEED REVS/MIN / PRESSURE LBF/IN^2	50	100	200	500	800	1200
0	15.6	15.6	15.6	16.8	18.4	20.0
50	19.2	18.0	19.2	18.0	20.4	20.4
100	24.0	20.6	22.8	22.8	24.0	26.4
150	28.8	26.4	26.4	27.6	28.8	30.0
200	34.8	31.2	32.4	33.6	33.6	33.6
250	40.8	36.0	36.0	37.2	37.2	37.2
300	48.0	42.0	42.0	43.2	43.2	40.8
350	54.0	48.0	48.0	46.8	48.0	44.4
400	60.0	54.0	52.8	49.2	50.4	48.0

<u>T A B L E 9</u>

<u>RESULTS OBTAINED FROM THE SEWING MACHINE ROTARY VALVE WITHOUT AN 'O' RING</u>

THE RESULTS GIVE THE TORQUE LBF-IN REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS.

SPEED REVS/MIN / PRESSURE LBF/IN^2	50	100	200	500	800	1200
0	7.2	7.2	7.2	9.6	12.0	14.0
50	8.4	8.4	8.4	9.6	12.6	14.4
1100	9.6	9.6	9.6	12.0	13.8	15.6
150	10.8	10.8	12.0	13.2	15.6	16.8
200	14.4	13.2	15.6	15.6	18.0	19.2
250	16.8	15.6	18.0	19.2	20.4	21.6
300	19.2	18.0	20.4	21.6	22.8	24.0
350	21.6	21.6	22.8	25.2	26.4	27.6
400	24.0	24.0	26.4	28.8	30.0	31.2

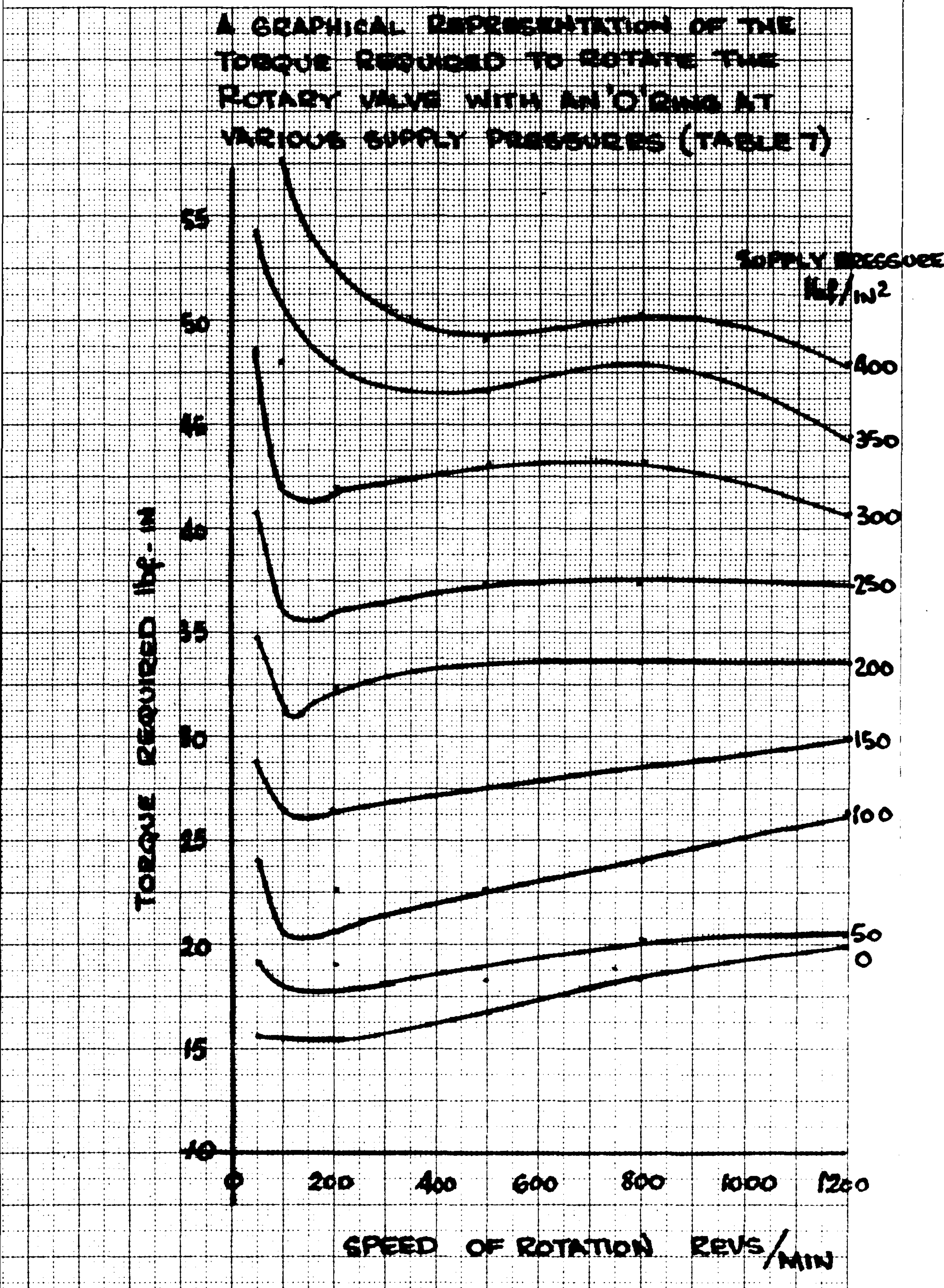

A GRAPHICAL REPRESENTATION OF THE TORQUE REQUIRED TO ROTATE THE ROTARY VALVE WITH AN 'O' RING AT VARIOUS SUPPLY PRESSURES (TABLE 7)
SUPPLY PRESSURE lbf/in^2
400
350
300
250
200
150
100
50
0
TORQUE REQUIRED lbf-in
55
50
45
40
35
30
25
20
15
10
0
200
400
600
800
1000
1200
SPEED OF ROTATION REVS/MIN

FIG 79

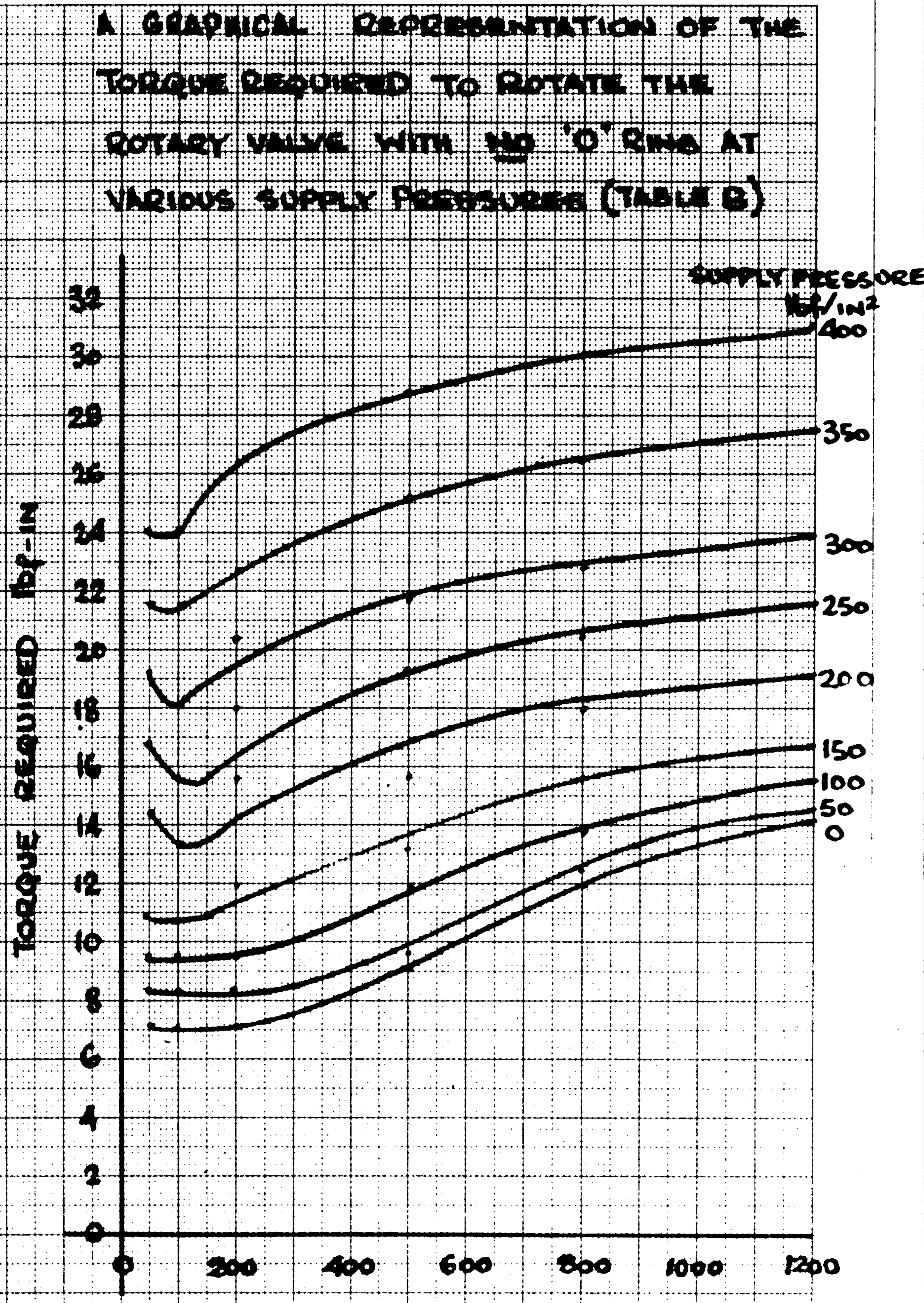

A GRAPHICAL REPRESENTATION OF THE TORQUE REQUIRED TO ROTATE THE ROTARY VALVE WITH NO 'O' RING AT VARIOUS SUPPLY PRESSURES (TABLE B)
SUPPLY PRESSURE lbf/in²
400
350
300
250
200
150
100
50
0
TORQUE REQUIRED lbf-IN
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
200
400
600
800
1000
1200
SPEED OF ROTATION REVS/MIN

FIG 80

4.7. Commissioning the Hydraulic Sewing Machine.

Prior to any sewing being performed upon the machine, the overall power requirements had to be estimated using the data obtained from the rotary valve. Since the original sewing machine motor was rated at half a horse power, the maximum input speed possible was 1,000 revolutions per minute. This input speed was obtained by adjusting the pulley dimensions on the motor and hook drives accordingly. The machine was then fully assembled with the thread carriers, needle and yarn, as can be seen in figure 71 in preparation for sewing.

4.7.1. Timing the Sequence of the Various Mechanisms.

The first attempts at producing a lock stitch were made by manually turning the rotor so that phase adjustments to the rotary valve and hook mechanism could be made. This adjustment consisted of rotating the bobbin of the rotary valve such that the needle actuator was retracted at the moment the hook had passed the eye and picked up the thread. Having succeeded in co-ordinating the relative motions of the needle, thread take-up, hook drive and feed dogs, an attempt to sew under power was made. It was realised that the clutch mechanism could not transmit the torque required to give even acceleration and in consequence tended to grab, producing needle breakage. To relieve this problem, the drive motor was replaced by a half horse-power motor, driving through a variable ratio Carter gearbox. This motor was installed and was capable of producing a constant speed drive that could be controlled at the gearbox.

4.7.2. Testing the Sewing Action.

The machine could now be tested for sewing performance. At low speeds, the stitch was formed with the two threads being satisfactorily interlocked, but as the speed was increased, the back of the fabric appeared to have a double loop. This double-looping was traced to the hook picking up the loose yarn for a second time, after the stitch had been formed. A solution was to advance the point at which the thread take-up mechanism was withdrawn, so removing the loose yarn before the hook had completed its first revolution. This modification involved drilling a second pair of supply ports in the cylinder body of the rotary valve such that the thread take-up actuator was advanced. This resulted in higher sewing speeds. One further problem existed, that of random yarn breakage. Here the problem was confined to the hook picking up the yarn from the needle and passing it behind the shuttle. In order to perform this operation, a slight resistance had to be introduced into the yarn to ensure that it was pulled into the neck of the hook. If the yarn remained at the point of the hook, the yarn was cut by the hook mechanism. Several attempts to solve this problem were made, the best two solutions being the introduction of a small spring latch or a pair of loose tension discs. However, the problem was basically due to the linear motion of a hydraulic actuator replacing the complex thread take-up motion, and is an area that requires further investigation. A pictorial record of the sewing machine has been made on the film entitled "The Development of a Hydraulic Sewing Machine".

This film shows the general layout of the sewing machine together with the rotary valve and sewing head. It also demonstrates the machines' ability to produce a lockstitch on a piece of fabric. It is envisaged that the film should be viewed as a complementary feature to this section.

4.8. Findings and Recommendations for the Application of Hydraulic Actuation Techniques to Sewing Machines.

4.8.1. Comments on the Hydraulic Sewing Machine.

The work undertaken to build a lockstitch sewing machine provided useful experience and served as an exercise to highlight the main criteria to be considered in future work. It has proved that sewing is possible by using hydraulic actuators to replace the linear mechanical motions, but further research would be required if a commercial machine were to be a reality.

4.8.2. Further Practical Usage of Hydraulic Actuation Techniques as Applied to Sewing Machines.

The following areas require special consideration as the basis for future developments:-

(i) The type of stitch that is most applicable to the media of hydraulic actuation must be selected. The lockstitch machine chosen had a limited sewing capacity before the shuttle has to be rewound thus restricting its application for automatic machines.

(ii) It must be investigated where the greater degree of flexibility offered by the hydraulic control media can best be exploited, for example, when producing left-hand chain stitch machines.

(iii) The further development of miniature hydraulic devices in order to obtain higher cycling rates would be necessary, so enabling the hydraulic machine to be comparable in speed with mechanical systems. This research would involve testing actuators with programmed hydraulic stops and could even involve introducing cushioning devices to protect the actuator at the extremities of the stroke.

(iv) Compound and multiple actuators could provide useful techniques for extending the effective cycle times. For instance, a compound actuator on the needle could shape the time displacement profile so that the small movement required to make the needle clear the hook had to be completed in the critical 180° of hook rotation. The remaining movement could be contained in the other 540° (the hook rotating twice per revolution of the rotary valve) hence providing a more even cycling profile. One of the major disadvantages of the thread take-up mechanism was the excessive amplitude that had to be accommodated. Consequently, a single actuator of two inch amplitude could be replaced with two actuators each of one inch. These could be programmed to operate in series and so effectively reduce the time required to complete the cycle. The use of multiple actuators could also provide extra control over the yarn, by introducing a further two fixed points on the thread take-up time displacement profile.

(v) A basic study must be performed on the stitch formation so that the hydraulic machine would be designed to effect the desired stitch formation and not simply to replace the existing mechanical motion. Taking the needle movement on the lockstitch machine; the fabric penetration is governed by the hook geometry with the crank dimensions governing the overall movement of the needle. Using hydraulic techniques a better utilization of needle movement could be obtained by ensuring that a high percentage of the total needle movement took place whilst the needle penetrated the fabric. This would allow a reduction in needle amplitude.

(vi) Auxilliary mechanisms would require further detailed thought. The mechanisms most directly effecting the overall performance are those used to attach the thread carrying devices to the actuators. These components would essentially be as light as possible in order to reduce the inertia forces, but also be able to withstand the high cyclic stresses to which they would be subjected.

Whilst this study has not produced a commercially viable sewing machine, it is hoped that this new sewing technique will provide a fresh approach to the concept of automatic sewing machines. The work performed in this field is very limited due to the time allocation for the various aspects of the overall project, but it has provided a foundation upon which further development can be based and it is believed that an extension of the work will be undertaken by Courtaulds.

PART 3

A DETAILED DESIGN STUDY AND ANALYSIS OF PULSE-GENERATING ROTARY VALVES

PART 3

A DETAILED DESIGN STUDY AND ANALYSIS OF PULSE-GENERATING ROTARY VALVES

5. ROTARY VALVE DESIGN

5.1. Rotary Valves

From the experimental application of rotary valves to a circular weft knitting machine and a lockstitch sewing machine, it was evident that the rotary valves revealed a new technique for generating a series of cyclic pulses. The rotary valves thus far designed have been solely concerned with producing a device capable of generating the pulses required to control the external system. However, having demonstrated this capability, a further study was required to enable the general design criteria to be resolved and a procedure specified as to how efficient valves may be designed. It was also envisaged that a method for predicting the power requirement for the valve should be evolved so permitting the drive mechanism for the rotor to be specified at the design stage.

Experience with rotary valves has demonstrated that out of balance of forces can be generated by the pressure and exhaust grooves cut in the bobbin for pulse generation purposes. Since the most desirable running condition appears to be when these forces are minimised (a low supply pressure), a method of construction has to be devised whereby these internal forces are constrained or balanced. They could be constrained or carried by bearings placed externally to the pulse-generating section, thereby producing a shaft running in a cylinder.

A DIAGRAM TO SHOW THE MOST SUITABLE BASIC LAYOUT FOR A ROTARY VALVE.

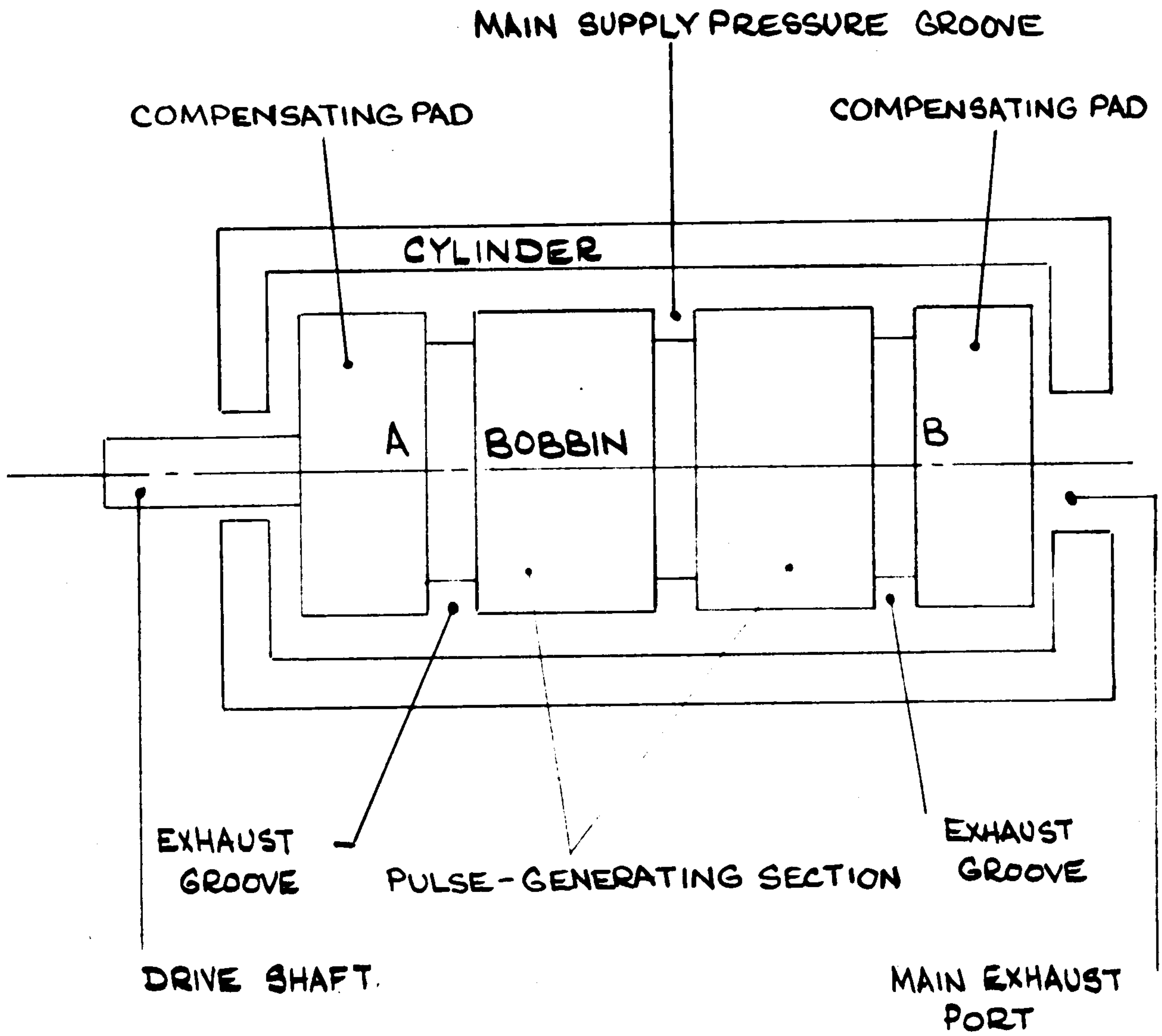


FIG 81

These bearings could be of any known type:- ball race, hydrostatic, or hydrodynamic providing that the load-carrying capacity was greater than the out-of-balance forces and that the eccentricity when loaded did not detract from the valve performance. A further examination of the rotary valve revealed that without the pulse-generating grooves, the valve would in effect be a plain hydro-dynamic bearing, running with no externally applied load. Under these conditions, the bobbin should run centrally in the cylinder (as a non-loaded shaft), and the torque requirements would be at a minimum. Consequently, the optimum valve performance could be obtained by regarding the valve as a complete hydrodynamic bearing, with the internal forces balanced at the design stage. The balancing would be achieved by considering the geometry and orientation of the pulse-generating pads, leaving only small residual forces to be counteracted by compensating pads. These will have to be individually designed for each particular slot geometry, and mounted at the end of the pulse-generating section.

5.1.1. Geometrical Layout of the Bobbin.

Therefore to design a balanced valve, the first consideration must be the geometrical layout internally, of ports and pads, in order to produce as symmetrical a design as possible. The suggested basic form for any future valves can be seen in figure 81. The bobbin, which is free to rotate about the central axis, is shown to be symmetrical about the main supply annulus. This central supply groove is fed, via the cylinder, by a high pressure source, serving as a common chamber to all the high pressure pads. The pulse-generating sections are placed to each side of the supply grooves and the exhaust pads connected to the central exhaust bore that runs along the axis of the bobbin. The geometry of the pulse-generating section is largely governed by the pulse requirements of the external system, but since the take-off ports can generally be placed at any point on the diameter of the cylinder, the orientation of one set of grooves with respect to a second set of grooves can be manipulated to produce a small resultant load. The most convenient way to reduce the resultant force is to make the pressure and exhaust pads symmetrical as possible across any diameter. Compensating pads placed at the bobbin extremeties can then be used to counteract the residual forces generated by the pulse-generating sections.

5.1.2. Balancing Techniques.

Having formulated this design strategy, a method of calculating the residual out-of-balance forces for any pad geometry had to be found, and then these forces used to calculate the physical pad dimensions.

The most common form of balancing pad used in general engineering practice is the hydrostatic bearing. The fluid film separating the journal and the bearing is maintained by a source of pressurised fluid external to the bearing with flow restrictions both in and external to the bearing. Therefore, the hydrostatic bearing pad provided a basis upon which to develop the compensating pads for the rotor.

The hydrostatic bearing requires a control device such as a restrictor or an orifice plate to limit the flow of fluid through the pads, thus preventing the complete system from being exhausted through a single pad. However, with the compensating pads the fluid flow rate will be many times greater than the internal leakage of the valve. Therefore, it could be assumed that the system would be constantly flow-saturated so acting as a self-compensating control device. Using this technique, the pressure at the compensating pads would vary in relation to the system pressure at the main take-off ports, so automatically restoring the pressure distribution balance as the demand on the rotary valve changes. A further advantage of using flow saturation would be that no pressure drop would be required across a control device so enabling the overall dimensions for the compensating bearing to be kept to a minimum.

The shape of the compensating pad would be a rectangular recess a few thousandths of an inch deep although it must have sufficient depth to not restrict the flow of fluid.

The assumption that the rotor will act as a hydrodynamic bearing provided a starting point from which an analytical solution could be developed.

5.2. Basic Theory And Assumptions.

The basic equations of fluid film lubrication for rigid bearing components are based upon the following fundamental relationships:-

(i) The continuity equation which represents the conservation of mass.

(ii) The equations of motion which represent the conservation of momentum.

(iii) The energy equation which represents the conservation of energy.

These three equations apply to all fluids, when written in their most general form. Osborne Reynolds, in 1886, derived from the first two equations a differential equation governing pressure distribution in a fluid film bearing. The derivation and simplification of this equation is standard Hydrodynamic theory and can be found in several text books (references 1 and 2) consequently it is not reproduced in this thesis.

When applying Reynold's Equation to a finite journal bearing and taking the assumptions which are known to be true for all ordinary lubrication conditions, then the equation may be written in two dimensions with steady motion as

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=6U\eta\frac{dh}{dx} \qquad \text{(1)}$$

providing that the following assumptions are made:-

(iv) Lubricant is Newtonian

(v) Fluid is incompressible

(vi) Flow is laminar

(vii) Fluid film is so thin that the pressure remains constant across the depth.

(viii) No slip occurs between fluid and bearing surfaces.

(ix) The viscosity of the film is uniform throughout.

(x) The effects of thermal and elastic distortion are neglected.

(xi) The shaft axis is parallel to the axis of the bearing.

(xii) Inertia and body forces are negligible.

(xiii) The curvature of the bearing surfaces are considered to be large compared with the film thickness so that the film may be unwrapped for analysis.

Thus, looking at the bearing at any instance in time when travelling with angular velocity, ω, the system will be of the form seen in figure 82 and it can be shown that the film thickness, h, at point θ round the bearing can be found from the relationship:-

$$h = c\,(1 + \varepsilon\cos(\theta - \phi)) \qquad \text{\_\_\_\_\_\_\_\_} (2)$$

where C is known as the radial clearance of the bearing and ε the eccentricity ratio defined as $\frac{e}{C}$ and ϕ the eccentricity angle.

Reynold's Equation in its most convenient form, is expressed in Cartesian Co-ordinates. Therefore, using the assumption that the bearing can be unwrapped and letting the arc length produced by angle θ be expressed as the x co-ordinate and the breadth of the bearing as the y co-ordinate, the bearing can be expressed as a pad, of length L = $2\pi R$ and breadth B. In order to use Reynold's Equation it is usual to express it in a dimensionless form:-

$$\bar{p} = \frac{p\,h_o^2}{\eta U L} \qquad \text{\_\_\_\_\_\_\_\_} (3)$$

A DIAGRAM TO SHOW THE NOTATION FOR A FULL JOURNAL BEARING.

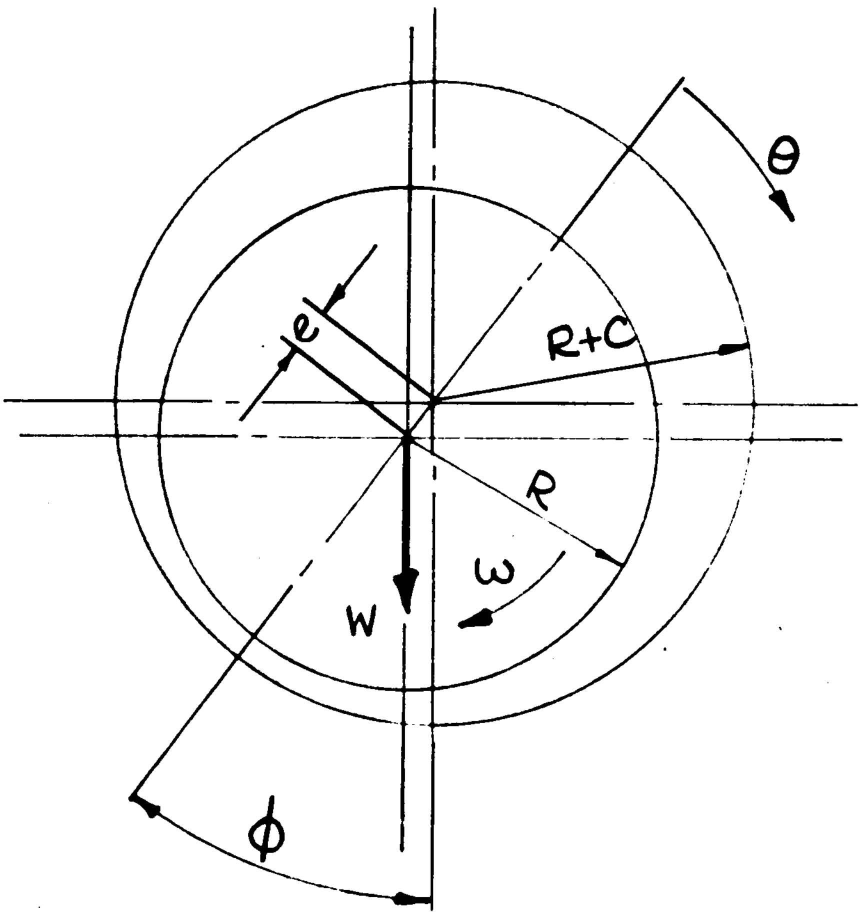


FIG. 82

$$\bar{y} = \frac{y}{B} \qquad (4)$$

$$\bar{x} = \frac{x}{L} \qquad (5)$$

$$\bar{h} = \frac{h}{h_o} \qquad (6)$$

(where ho is the minimum film thickness)

Expressing Reynold's Equation for a finite bearing in dimensionless form gives:-

$$\frac{\partial}{\partial x} \cdot \frac{1}{L}\left(\bar{h}^3 h_o^3 \frac{\partial \bar{p}}{\partial \bar{x}} \cdot \eta \frac{UL}{h_o^2} \cdot \frac{1}{L} \right) + \frac{\partial}{\partial y} \cdot \frac{1}{B}\left(\bar{h}^3 h_o^3 \frac{\partial \bar{p}}{\partial \bar{y}} \cdot \eta \frac{UL}{h_o^2} \cdot \frac{1}{B} \right) = 6\eta U \frac{d\bar{h}}{d\bar{x}} \cdot \frac{h_o}{L} \qquad (7)$$

which on rearranging produces:-

$$\frac{\partial}{\partial \bar{x}}\left(\bar{h}^3 \frac{\partial \bar{p}}{\partial \bar{x}} \right) + \frac{\partial}{\partial \bar{y}}\left(\frac{L^2}{B^2} \right)\left(\bar{h}^3 \frac{\partial \bar{p}}{\partial \bar{y}} \right) = 6 \frac{d\bar{h}}{d\bar{x}} \qquad (8)$$

Thus, if the values of ϕ and ε are either known or assumed, then the pressure distribution for a finite journal bearing can be found by solving the differential equation. However, in this particular instance, neither parameter was known, and since it is hoped to design a balanced rotor equivalent to an unloaded shaft, it was reasonable to assume that the bobbin would run almost centrally within the cylinder so approximating eccentricity e to zero, which in turn eliminates the angle ϕ (the eccentricity angle). Making such an assumption simplifies the Reynold's Equation because,

$$h = h_o = \text{constant}$$

thus giving the Laplace Equation

$$\frac{\partial^2 \bar{p}}{\partial \bar{x}^2} + \left(\frac{L^2}{B^2}\right)\frac{\partial^2 \bar{p}}{\partial \bar{y}^2} = 0 \qquad \text{(9)}$$

The solution of the Laplace Equation would enable the pressure distribution round the bearing to be found. This in turn could be used to calculate the magnitude and direction of the residual out-of-balance forces. Having compensated the system with hydrostatic pads, the assumption that the rotor acts as a non-loaded shaft should be true.

5.3. Solution of Laplace Equation for the Pressure Distribution round the Rotor.

The equation to be solved was the Laplace Equation with special boundary conditions depending upon the geometry of the slots required by the system. Several methods of solving the equation from series approximations to electrical analogue were available and fully established, but with the advent of the digital computer the most convenient solution would be a relaxation method, as all fluid film lubrication problems can be made to yield to them. First approximations can be obtained quickly and the numerical procedure pursued infinitely to achieve any required degree of accuracy. Using the relaxation process, the first step was to replace the differentials in the equation by finite difference approximations. The area to be considered was covered by a mesh and the method relies upon the fact that a function can be represented with sufficient accuracy over a small range by a quadratic expression.

Consider the variation of a function p in the direction of x and let the mesh size be $\propto$ (see figure 83i) thus it can be seen that:-

$$x = o \qquad p = p_o$$

$$x = +\propto \qquad p = p_{+\propto}$$

$$x = -\propto \qquad p = p_{-\propto}$$

And by using a finite difference form the rate of change of p at a point $x = -\frac{\propto}{2}$ can be expressed as,

$$\left(\frac{\partial \bar{p}}{\partial \bar{x}}\right)_{-\frac{\propto}{2},\ \propto \to 0} = \frac{p_o - p_{-\propto}}{\propto} \qquad \text{(10)}$$

Similarly at point $x = +\frac{\propto}{2}$

$$\left(\frac{\partial \bar{p}}{\partial \bar{x}}\right)_{+\frac{\propto}{2},\ \propto \to 0} = \frac{p_{+\propto} - p_o}{\propto} \qquad \text{(11)}$$

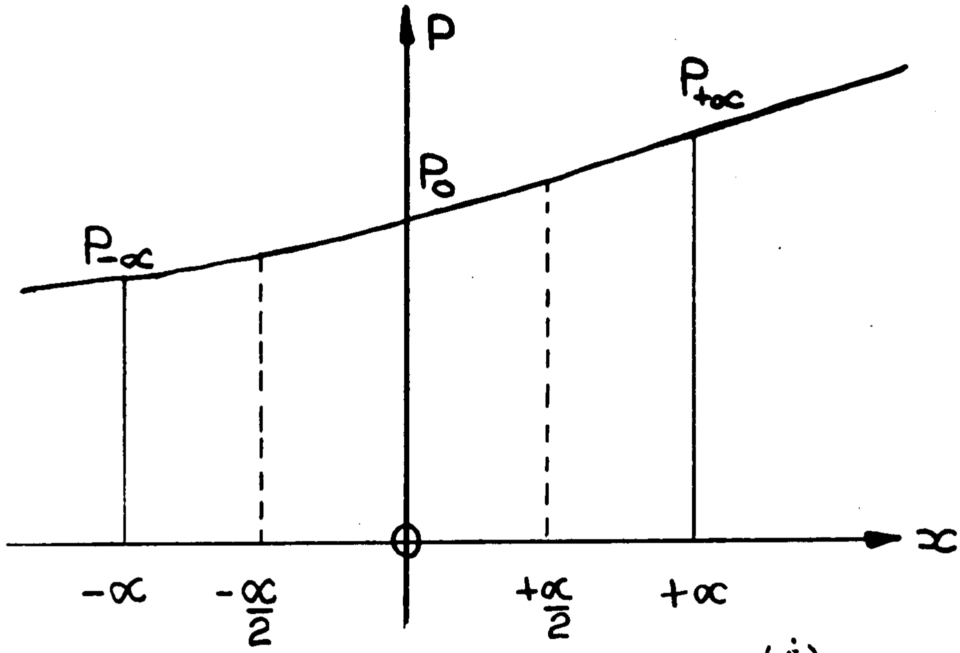
Again, by a similar finite difference representation the rate of change of $\left(\frac{\partial \bar{p}}{\partial \bar{x}}\right)$ at x = o can be expressed as

$$\left(\frac{\partial^2 \bar{p}}{\partial \bar{x}^2}\right)_o = \frac{\left(\frac{\partial \bar{p}}{\partial \bar{x}}\right)_{+\frac{\propto}{2}} - \left(\frac{\partial \bar{p}}{\partial \bar{x}}\right)_{-\frac{\propto}{2}}}{\propto} \qquad \text{(12)}$$

$$\therefore \left(\frac{\partial^2 \bar{p}}{\partial \bar{x}^2}\right)_o = \frac{p_{+\propto} + p_{-\propto} - 2p_o}{\propto^2} \qquad \text{(13)}$$

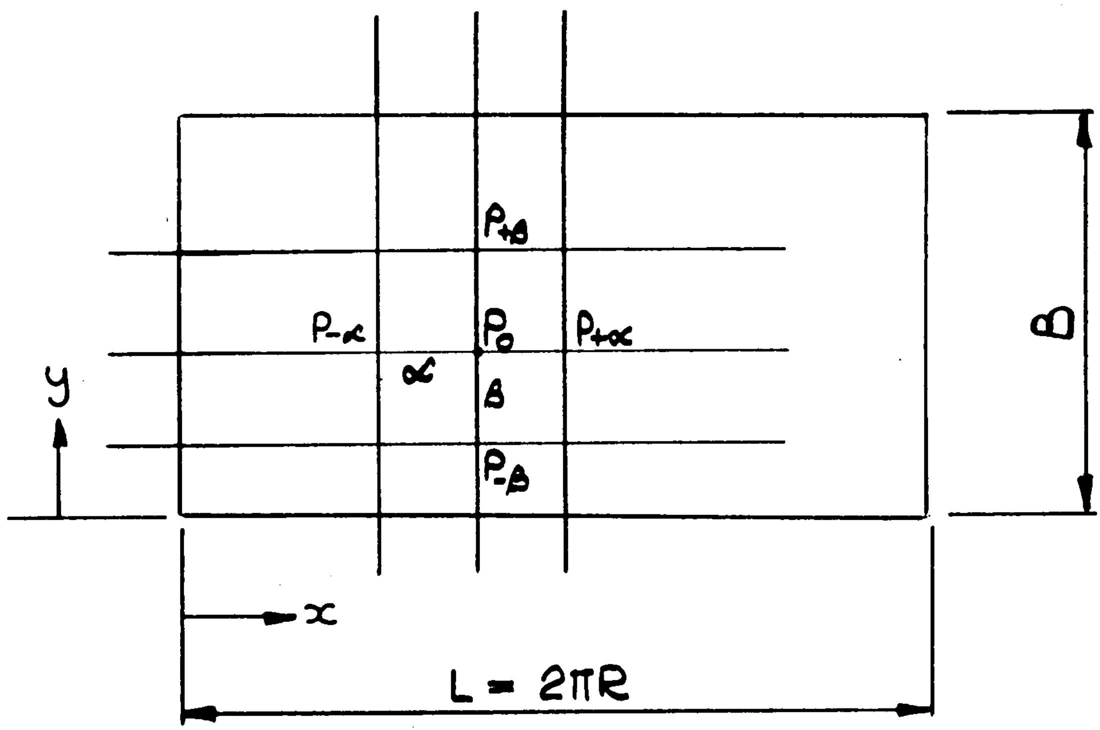
Also by considering the variation of a function p in the direction

A DIAGRAM TO SHOW HOW THE PRESSURE AT A POINT CAN BE REPRESENTED IN A FINITE DIFFERENCE NOTATION



(i)

A DIAGRAM TO SHOW HOW THE FINITE DIFFERENCE NOTATION CAN BE APPLIED TO THE PRESSURE DISTRIBUTION ON A PAD USING A GRID.



(ii)

FIG. 83

of y and letting the mesh size be β

then $$\left(\frac{\partial^2 \bar{p}}{\partial \bar{y}^2}\right)_0 = \frac{p_{+\beta} + p_{-\beta} - 2p_0}{\beta^2} \qquad \text{(14)}$$

These two expressions for $\frac{\partial^2 p}{\partial x^2}$ and $\frac{\partial^2 p}{\partial y^2}$ can be substituted directly into the Laplace Equation

$$\frac{\partial^2 \bar{p}}{\partial \bar{x}^2} + \left(\frac{L^2}{B^2}\right)\frac{\partial^2 \bar{p}}{\partial \bar{y}^2} = 0 \qquad \text{(15)}$$

thus

$$\frac{p_{+\alpha} + p_{-\alpha} - 2p_0}{\alpha^2} + \left(\frac{L^2}{B^2}\right)\left(\frac{p_{+\beta} + p_{-\beta} - 2p_0}{\beta^2}\right) = 0 \qquad \text{(16)}$$

$$\therefore \; p_0 = \frac{p_{+\alpha} + p_{-\alpha} + \frac{(L\alpha)^2}{(B\beta)^2}\left(p_{+\beta} + p_{-\beta}\right)}{2\left(1 + \frac{(L\alpha)^2}{(B\beta)^2}\right)} \qquad \text{(17)}$$

Therefore, by setting out a grid over the pad (as shown in figure 83ii) an approximation of the pressure at any point could be found, if the pressures around the point were known. Consequently to start the relaxation process in motion the boundary conditions must be known and also since the hydrodynamic action is being neglected, a pressure source must be introduced. If the system were being worked by hand, the centre point would be investigated first with $\alpha = \beta = \frac{1}{2}$ and then the grid size gradually increased to the desired mesh. Once the first approximations have been found, the relaxation procedure could be initiated. This technique would involve taking each point on the grid in turn, and putting the values first calculated into the general expression for the pressure distribution. These would not satisfy the equations exactly so the value of p_o would be adjusted such that the residual was equal to zero, hence satisfying the equation. This adjustment would affect the pressure at the points around p_o so the process would have to be repeated at each point across the whole grid. Once having traversed the grid, all the pressure values would have been altered, so repeating the process until the equations are satisfied to a sufficient degree of accuracy. This process may take as many as 100 iterations across the grid depending upon the accuracy required. To speed up the relaxation process, a technique of over-relaxation could be employed. This was put forward by Gauss-Seidel stating that:-

Value of p to be put onto grid

= Previous value of p + Relaxation Factor (A) $\times$

(Calculated value of p - Previous value of p)

This relaxation factor is a number between 1.0 and 2.0 and is set to the value that produces the most accurate answer for the minimum number of iterations round the grid.

It will be appreciated that only an outline of the process has been included in this thesis, as the techniques have been fully developed and can be found in references (4) and (5).

5.4. <u>Application of Laplace Equation to the Rotary Valve</u>

The approach adopted for the application of the Laplace Equation to the rotary valve involved using relaxation techniques on a digital computer. The computer programme can be used for solving the Laplace Equation for any size valve and pad geometry. In section 5.5. an outline will be given as to how the programme was evolved and how it can be used.

5.5. The Pulse-Generating Section of the Valve.

The positioning of the slots to be cut in the central section of a rotary valve is always dictated by the external system. e.g. if the system is supplied with a pressure pulse at a given instant, a slot must have been machined radially into the bobbin so producing a pressure pad. (This procedure has been dealt with in detail in Chapter 2.) The first consideration of any design must be pad geometry. This is most easily solved by drawing cross sectional views of the rotary valve through the pressure and exhaust pads, as shown in figure 4. This enables the train of pulses generated by valve station to be simulated. Once the pad dimensions have been established, the overall pad layout must be made as symmetrical as possible by adjusting the radial position of the slots. The overall breadth of the valve is governed by the number of slots that have to be cut in the valve bobbin to satisfy the system. The overall diameter of the valve should be made as small as possible to reduce such factors as inertia, torque requirements and material utilisation, but is primarily governed by the number of take-off points required by the system. The volume flow rate passed through the internal bores of the bobbin must not create too large a pressure drop across the ports. Experience has shown to err on the large side when deciding the diameter of the internal bores, as not all the factors that contribute to the total pressure drop can be accounted for in a mathematic model. The size of the bores required will also be related to the number of pulses required per revolution, and the speed at which the bobbin is rotating. However, a few simple calculations concerning the parameters of the system and the application of those parameters to the rotary valve would provide a basis upon which the diameter of the valve could be selected.

Having established the size of the pulse-generating section, these two most important facts still have to be determined:-

(i) The out-of-balance forces, at each end of the pulse-generating section, so that suitable pads can be designed to oppose them.

(ii) The minimum torque required to overcome the viscous frictional resistance of the bobbin.

Since both calculations were dependent upon the pressure distribution over the bobbin, the starting point must be to solve the Laplace Equation for a particular pad geometry. To solve the Equation using the computer the pad geometry was transposed onto the grid containing the complete pulse-generating section, and the equations solved for the given boundary conditions imposed. The easiest method of performing this procedure was to select a mesh size, for the Breadth of the section β and the developed Length of the section α; the number of grids could be any even number between 20 and 100, (though solutions for grids larger than 50 by 50 become rather lengthy even on a fast computer). This grid was then plotted, using a convenient scale, onto graph-paper such that the grid points could be numbered. When considering the breadth of the valve, the grid could be numbered from 1 to N + 1 where β is the grid size initially selected, and the boundary condition set by letting the pressure be equal to that in the exhaust line. (P MIN). When considering the boundary conditions for the developed length of the valve, the end conditions cannot be specified directly. Since it was a continuous band, the pressures at both ends of the section defined by length L must be the same. Consequently, the solution for the boundary conditions could be overcome by extending the two ends so that they overlapped. Thus when considering the grid points, the length L is described by the grid points 2 to N + 2, α being the grid size initially selected. It can be stated that the pressure at grid point 1 is equal to the pressure at grid point N + 1, the pressure at 2 is equal to the pressure at grid point N + 2 and the pressure at 3 is equal to the pressure at N + 3. Figure 84 shows the basic grid structure upon which the orientation of the slots on the rotary valve have to be transposed. This technique involves taking the cross sectional views of the rotary valve and mapping the various pressure and exhaust pads onto the grid drawn on the graph paper; a typical plot is shown in figure 85. This plot enables the position of the pressure pads to be expressed in co-ordinate values of I and J to be fed in as data for the computer. The number of pads and the position of the co-ordinates were all processed in a subroutine named PRESSURE POINTS (I, J, K) (which can be expanded by increasing the number of logic statements required; each statement identifies one slot in the rotary valve). This sub-routine consisted of a series of IF statements into which

DIAGRAM TO SHOW THE GRID, SUPERIMPOSED ON THE PULSE GENERATING SECTION OF THE ROTARY VALVE AND TO DEFINE THE GRID AT THE MESH POINTS

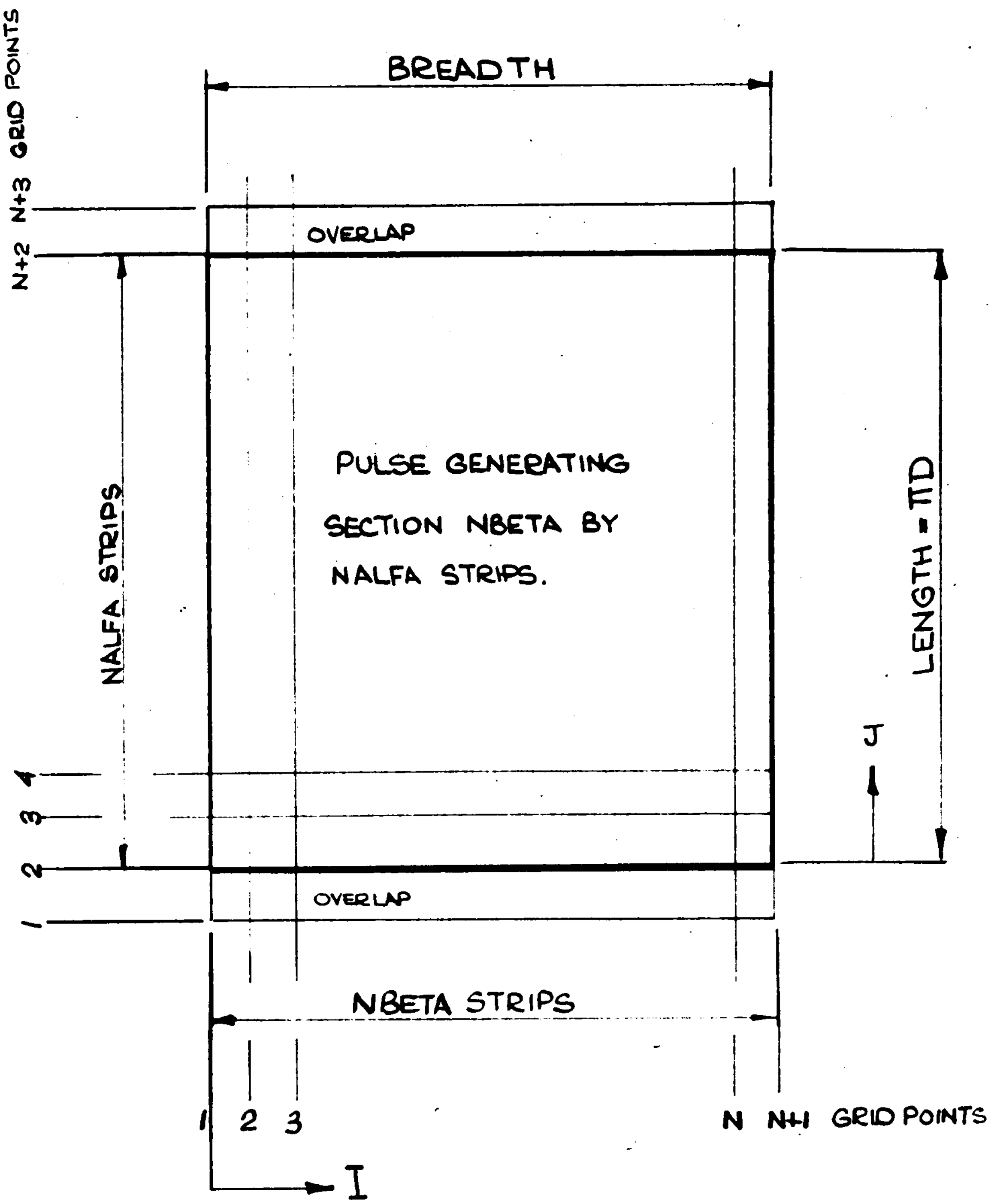


FIG 84

A DIAGRAM TO SHOW HOW THE PRESSURE AND EXHAUST PORTS ARE PLOTTED ON THE RELAXATION GRID TO FIND THE I AND J CO-ORDINATES OF THE SLOTS

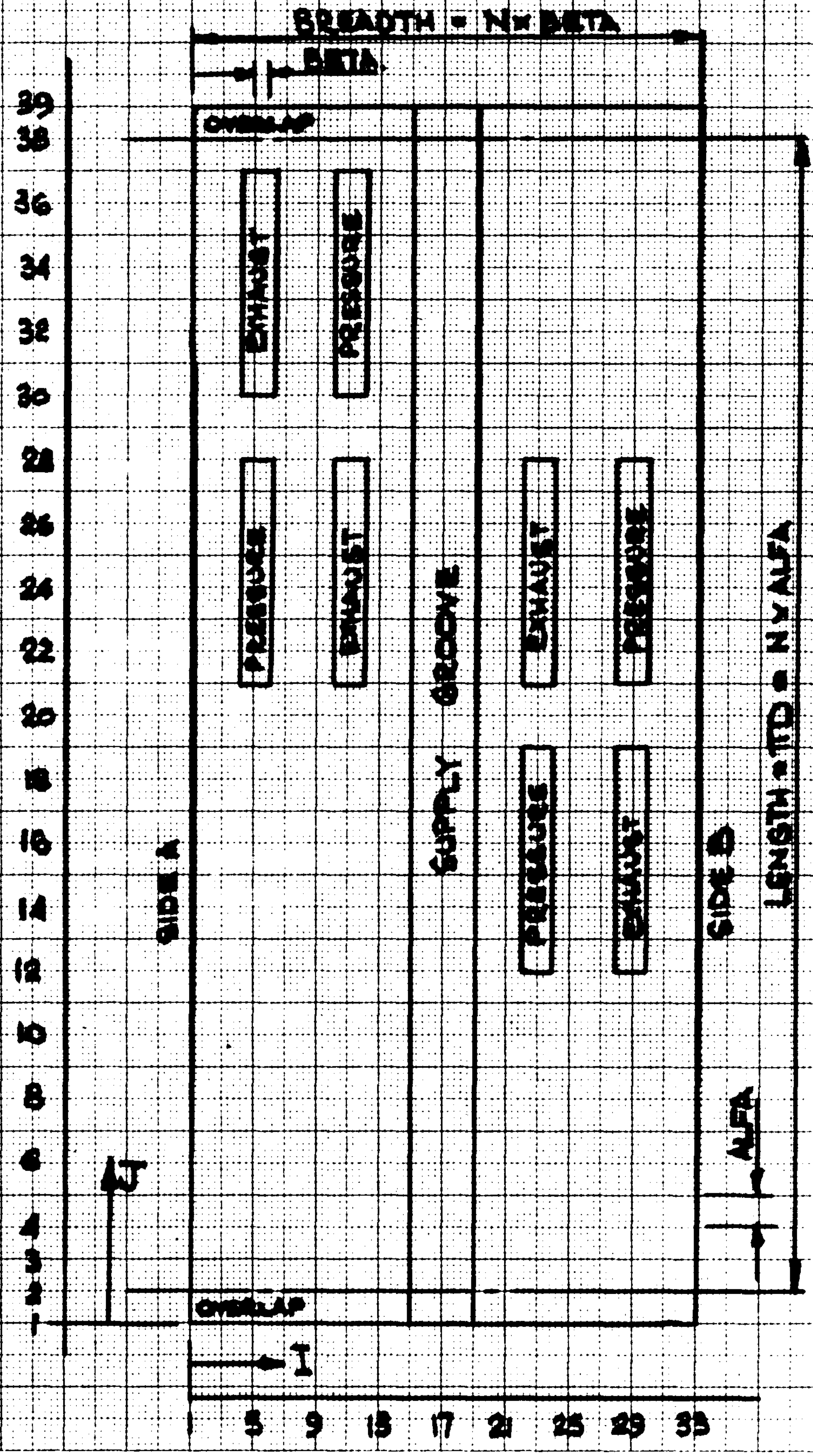


FIG. 85

the grid points of the slots had been transposed, and the pressure at the various slots were set to either PMAX or PMIN depending upon the state of the slot. The co-ordinates of a particular slot were punched onto a card with the I values followed by the J values using a 4I0 format. These cards were put into sequence; all the high pressure slots were followed by the low pressure slots. The last card was used to define the high pressure supply annulus. The value of NSLOT gave the instruction as to the number of co-ordinates to be read from the data cards.

The computer programme, compiled in FORTRAN IV FORMAT entitled MASTER PRESSURE OF SLOTS calculated the pressure distribution over the valve's surface, the out-of-balance load, and the minimum torque required to overcome viscous friction. The information to be supplied to the programme took the following form:-

(iii) DATA CARD 1, contained the fixed design parameters of the valve and the system

DIAMETER of the Rotor (in)

BREADTH of the pulse-generating section (in)

VISCOSITY of the oil to be used in the system (centipoises)

HO the radial clearance of the bearing (in)

These four values had been temporarily fixed at the initial design stage and in consequence were unlikely to be required as variables in the programme.

(iv) DATA CARD 2, set the desired accuracy to which the relaxation solution was required. An EPS value of 0.1 gives a largest error of 1% as:

$$EPS = \frac{P.\ NEW - P.\ OLD}{P.NEW}$$

The MAXIT figure usually set at a value of about 100 enables a result to be obtained after a given number of iterations round the relaxation loop. The maxit facility is mainly used when initiating the solution, and for checking the various parameters in the programme. For example, running the programme to fix the best value of A (the relaxation factor) for a particular grid size and pad geometry. This cannot be calculated directly, but the value of the residual after 20 iterations for three different values of A will enable it to be optimised and hence save computer time for a more accurate solution involving a larger number of iterations.

(v) DATA CARD 3 - 11 (depending upon the number of slots in the pad) contained the co-ordinates of a slot. Each card contained four numbers giving first the I grid co-ordinates, followed by the J co-ordinates. These pad co-ordinates are usually fixed once the geometry of the valve has been established but certain minor adjustments to the relative positions of adjacent sets of slots may enable the forces created by the pressure and exhaust slots to be balanced. It should be noted that if the outlet port is displaced through the same angle as the pulse-generating slots in the bobbin, then the pulses remain in the same sequence.

(vi) DATA CARD 12 contained the operating variables of the rotary valve: SPEED of valve rotation which was directly related to the system parameters. (revs/min). PMAX being the system pressure to perform the duty for which it was designed. (lbf/in^2) PMIN being the exhaust pressure; this is ideally set at zero, but in practice was always found to be at some absolute value, depending upon the size of the return lines back to tank. (lbf/in^2).

Once having compiled the required data in the units specified by the programme, the computer proceeded to calculate the solution to the Laplace Equation. The process of logic used in the programme is outlined as follows:-

(vii) The data is converted into compatible units to conform with the linear pad geometry.

(viii) The grid size is set to the selected value, and all the pressures at the mesh points are given an initial value of PMIN, (the exhaust pressure).

(ix) The iteration process is then performed in which the values of the pressure at the supply ports are set and the pressure distribution calculated, using the Gauss Seidel technique (see section 5.3).

Having calculated the pressure distribution to the desired accuracy, the information can be utilised to give an estimation as to the magnitude of the out-of-balance forces produced within the system.

(x) The force at each grid point can be calculated by using the pressure and the area over which it acts. Therefore, by taking the force on each element round the circumference of the bobbin, and converting each one into vertical and horizontal components, the total value of the force at each section across the valve in the I direction can be found.

(xi) The force at each section can then be summated. By taking moments about each end of the pad section in turn, and summing the moments, a value for the out-of-balance force at each end of the rotor can be found in both magnitude and direction.

The computer programme was compiled to give a print-out of the results calculated at each stage, so providing information for making design improvements to the pulse-generating section of the valve.

5.6. Torque Required to Turn the Pulse-Generating Section.

The torque expression can be derived from established theory, see references (1) and (2). The expression of the total force is given by integrating the value of the shear stress τ over the bearing surface:-

$$F = \iint \tau \, d\text{AREA} \qquad \text{(18)}$$

The expression for the shear stress τ can be shown to be:-

$$\tau = -\frac{h}{2}\left(\frac{\partial p}{\partial x}\right) - \eta\frac{U}{h} \qquad \text{(19)}$$

All the values are known, except the term $\frac{\partial p}{\partial x}$ which by using the finite difference relationship

$$\left(\frac{\partial p}{\partial x}\right)_{x=0} = \frac{p_{+\alpha} - p_{-\alpha}}{2\alpha} \qquad \text{(20)}$$

The Total Frictional force can be calculated by solving numerically the following expression:-

$$\text{Frictional Force} = \int_0^B\left(\int_0^L\left(\eta\frac{U}{h} + \frac{h}{4}\left(\frac{p_{+\alpha} - p_{-\alpha}}{\alpha}\right) dL\right)dB\right. \qquad \text{(21)}$$

which produces:-

Total Torque = Frictional Force x Radius of the Bobbin.

The programme to calculate the torque required was a logical continuation because the pressure distribution had been calculated at each mesh point and held in store, consequently the expression for the shear stress could be calculated over the complete grid using a single mathematical expression. The double integration was performed by applying Simpson's Rule in two directions (hence the even number of grids) to give the overall shear force. The torque required by the system was printed out at the end of the sequence.

This programme, the MASTER PRESSURE OF SLOTS, which can be seen in Appendix 1, took a simplified expression for the pressure distribution in a rotary valve and used it to calculate the overall load on the pulse-generating section. Having once established an approximate direction and magnitude for the out-of-balance load, the expression for the pressure distribution could be made more complex by introducing the eccentricity term which would invalidate the Laplace Equation and require the pressure distribution solution to be obtained by relaxing Reynold's Equation. The complexity of the theoretical study could be expanded further by allowing the centre of the bobbin to whirl about an eccentric centre, thus again increasing the Reynold's Expression for the pressure distribution. However, it was felt that the simplified expressions developed in this section made possible sufficiently accurate theoretical predictions upon which to base a practical design procedure.

5.7. The Design of Compensating Pressure Pads.

Once the magnitude and direction of the out-of-balance forces had been determined, two hydrostatic compensating pads had to be designed, such that the pressure distribution would balance the resultant force generated by the slots. A second computer programme MASTER COMPENSATING PADS was compiled. This programme used the same assumptions and techniques as the first programme but was self-checking, producing a result which gave the compensating bearing and pad dimensions.

The parameters fixed in this solution were that the full compensating pad covered a 180° arc of bearing surface, and also that the recess perimeter was between 0.4 and 0.6 of the bearing perimeter. (This value was recommended by O'Donoghue (11))

When calculating the dimensions of the compensating pad, the direction of the balancing force was neglected because it had already been determined, and since the balancing pad was symmetrical, it's centre had to be designed to oppose the out-of-balance force produced by the pulse-generating section.

The DATA supplied to the computer was basically the same as for the first programme, with the addition of EHO, which was the clearance of the end plates on the rotary valve. This information was used to calculate the viscous friction produced by the end caps. The only other amendment was the replacement of the data containing the co-ordinates of the pressure slots by a value of the out-of-balance force.

The basic sequence of the programme was as follows:-

(i) To convert the input data into compatible units and adjust the parameters of the system to conform to a linear pad geometry.

(ii) To set the grid to a convenient size, though if the grid area $\alpha \times \beta$ used to calculate out-of-balance forces was large, no benefit would be gained by using a very fine mesh.

(iii) The pad co-ordinates are then calculated and set such that the pad perimeter will be half the bearing perimeter.

(iv) An estimate as to the breadth of the compensating pad is then made using the assumption that a linear pressure gradient will be generated across the bearing surface.

(v) The pressure is calculated over the bearing surface, and this pressure is summed over the area to produce a value of the resultant force, (as in MASTER PRESSURE OF SLOTS programme).

(vi) This calculated resultant force is then compared with the known force to be balanced, and the pad size is then adjusted by a single co-ordinate across the breadth of the bearing to either increase or decrease as required, the force generated by the compensating pad. The force generated by this new size pressure pad is recalculated and compared again with the known out-of-balance force. This process is repeated until the difference between the calculated value and the known value changes sign, indicating that the bearing pad dimension must lie between the two co-ordinates. The breadth of the pressure pad can then be calculated using the assumption that the resultant force will vary linearly with respect to the pad size, between the two adjacent grid points.

(vii) A check is then made as to the ratio of the bearing perimeter to the pad perimeter. If this lies outside the optimum value, then the breadth of the bearing is readjusted using the calculated pad size, and the complete calculation repeated.

(viii) Having optimised the pad size and bearing breadth, these two values are printed out and can be used directly in the design of the rotary valve.

(ix) The remaining section of the programme is directed to calculating the torque required to overcome the frictional resistance generated by the compensating pad. (This process is again as described in section 5.6). This torque value can be added to the viscous friction produced by the complementary plain bearing covering the remaining 180° arc of the compensating pad.

(x) To finalise the calculation, the frictional torque due to the end covers is added to the programme (The basic derivation for this can be found in Appendix V).

It must be remembered when using this second programme that different compensating pads were required at each end of the rotary valve, so the reactions at ends A and B had to be fed in as data, and the final print-out gave full details of both compensating bearings. The complete computer programme for the compensating pad bearings can be seen in Appendix II. As with the first programme, the mathematical model could be increased in complexity. If rotary valves find a direct commercial application, then a further study using mathematical techniques will be justified, however it was felt that these two programmes enabled valve performance to be predicted to a limited degree of accuracy, and would serve as useful tools when designing a rotary valve for a particular application.

5.8. Verification of the Predicted Theory by Practical Tests.

Once having developed a basis upon which rotary valves could be designed, a comparison had to be made between the results predicted by the analytical approach and the parameters found in experimental tests. The most effective method of performing this comparison was to design a rotary valve, based on the pulse requirements of the sewing machine, using the computer programmes previously outlined. Tests on the valve could then be performed enabling the predicted results to be examined in the light of practical results obtained.

5.8.1. Design data for Test Rotary Valve.

The first task was to design the pulse-generating section to suit the system previously specified. The systems' requirements were translated into design parameters so providing the following data for the computer:-

(i) Diameter of the rotor - 2 inches

(ii) Breadth of the pulse-generating section - 4 inches

These two parameters were governed by the earlier rotary valve.

(iii) The Viscosity of the oil - 90 centipoise

(iv) The radial clearance of the bearing - 0.001 inches.

The two values outlined above have been based on an ideal specification.

(v) The co-ordinates of the pressure and exhaust pads on the grid:-

4	66	21	28
10	12	30	37
22	24	12	19
28	30	21	28
4	6	30	37
10	12	21	28
22	24	21	28
28	30	12	19
15	19		

The technique for setting out the relaxation grid and superimposing the pressure and exhaust ports has been detailed in section 5.5, with the results being shown in figure 85. Here the actual ports are plotted onto a grid and the co-ordinates can be scaled directly from the plot to give the input data.

(vi) Speed of rotation - 500 revolutions per minute

(vii) Pressure at the feed slot - 300 lbf/in^2

(viii) Pressure at the exhaust slot - 0.01lbf/in^2

These three input parameters are variable under different running conditions. However, for the purposes of the initial design, an optimum value for each parameter had to be established. These values were not critical provided that they were consistent for both computer programmes and thought to represent the mean operating conditions for the valve.

5.8.2. Computer Predictions for the Out-Of-Balance Forces.

The computer then calculated the magnitude and direction of the resultant end forces:-

Total reaction at A = 30.586 lbf at an angle of -43.97 degrees

Vertical Component - ve Horizontal Component +ve

Total reaction at B = 30.678 lbf at an angle of -45.493 degrees

Vertical Component - ve Horizontal Component +ve.

These two reactions have to be examined in context with the horizontal and vertical reactions, to enable the correct quadrant for the balancing pad to be determined. The balancing pad was then designed in order to produce a resultant force equal to the reaction required to balance the pulse-generating section of the valve. For example, the angle of the reaction at end A was -44 degrees with a -ve vertical component and +ve horizontal component, thus producing an angle of 136 degrees for the direction of the total reaction. When the pressure compensating pad was introduced, it had to produce a resultant force in this direction so necessitating its introduction with a radial mid-position of 316 degrees.

The same parameters for the diameter of the valve, viscosity of the oil, radial clearance, speed of rotation, pressure at the feed slots, pressure at the exhaust slots plus the out-of-balance forces calculated by the first programme, were introduced into the second computer programme, MASTER COMPENSATING PADS. This produced the following print-out for end A:-

Length of bearing - 3.1415 in-

Breadth of bearing - .0865 in

Length of high pressure pad - 1.5708 in

Angle subtended by the pad - 90.0 degrees

Breadth of the high pressure pad - 0.0469 in

and for the end B:-

Length of bearing - 31.415 in

Breadth of bearing - 0.0867 in

Length of High pressure pad - 1.5708 in

Angle subtended by the pad - 90.00 degrees

Breadth of the high pressure pad - 0.0471 in.

This information to be used for balancing the rotor, enabled the design to be completed, and a series of detail drawings made. However, when surveying the overall sizes of the compensating pads it was obvious that this particular system was almost self-balancing, a desirable feature in itself in a rotary valve except for when a performance assessment for balancing pads was required. This situation was resolved by rotating the third pair of pressure and exhaust slots through a phase change of 90 degrees. This modified the pad co-ordinate input data to:-

4	6	21	28
10	12	30	37
22	24	21	28*
28	30	21	28
4	6	30	37
10	12	21	28
22	24	30	37*
28	30	12	19
15	19		

(* denoting the modified pads)

This new pad geometry produced a marked effect upon the overall balance within the system and gave the following results:- Total Reaction at A - 130.23 lbf at an angle of 13.668 degrees

Vertical Component - ve   Horizontal Component - ve

Total Reaction at B - 286.110 lbf at an angle of 29.270 degrees

Vertical Component - ve   Horizontal Component - ve

The corresponding pad geometries were calculated in the second programme and found to be:-

	End A	End B
Length of Bearing	3.141 in	3.141 in
Breadth of Bearing	0.368 in	.809 in
Length of High Pressure Pad	1.57 in	1.57 in

	End A	End B
Angle subtended by the pad	90.00 deg	90.00 deg
Breadth of High Pressure Pad	0.179 in	0.331 in

A decision to build the rotary valve with interchangeable bobbins was made. The design of bobbin 1 was for the self-balancing rotor, bobbin 2 had inherent out-of-balance forces. The detail drawings for the manufacture of these two bobbins can be seen in figure 86 drawing number J.D.G. 130, and figure 87, drawing number J.D.G. 132. Also the valve cylinder, drawing number J.D.G. 131 in figure 88. The end caps were identical to those used on the sewing machine valve seen in figure 70.

The materials used in manufacture were similar to those used on the earlier rotary valve. The only difference was that the bobbins were finished using a grinding operation for a better surface finish. Photographs of the test bobbins can be seen in figure 89 (Bobbin 1) and figure 90 (Bobbin 2). These photographs enable the compensating pads to be compared and the overall assembly of the rotary valve is shown.

5.8.3. Results obtained from Test Rotary Valve.

The rotary valve was assembled (with Bobbin 1 as the rotor) and the complete valve mounted onto the Telehoist Rig. The drive for the valve was obtained from the hydraulic servo-motor with the torque transducer interposed, thus enabling a torque measurement to be obtained. The oil supply to the valve was taken from an independent hydraulic power pack, and a supply line pressure tapping made to a Bourdon tube pressure gauge. The input speed was again monitored using the tacho-

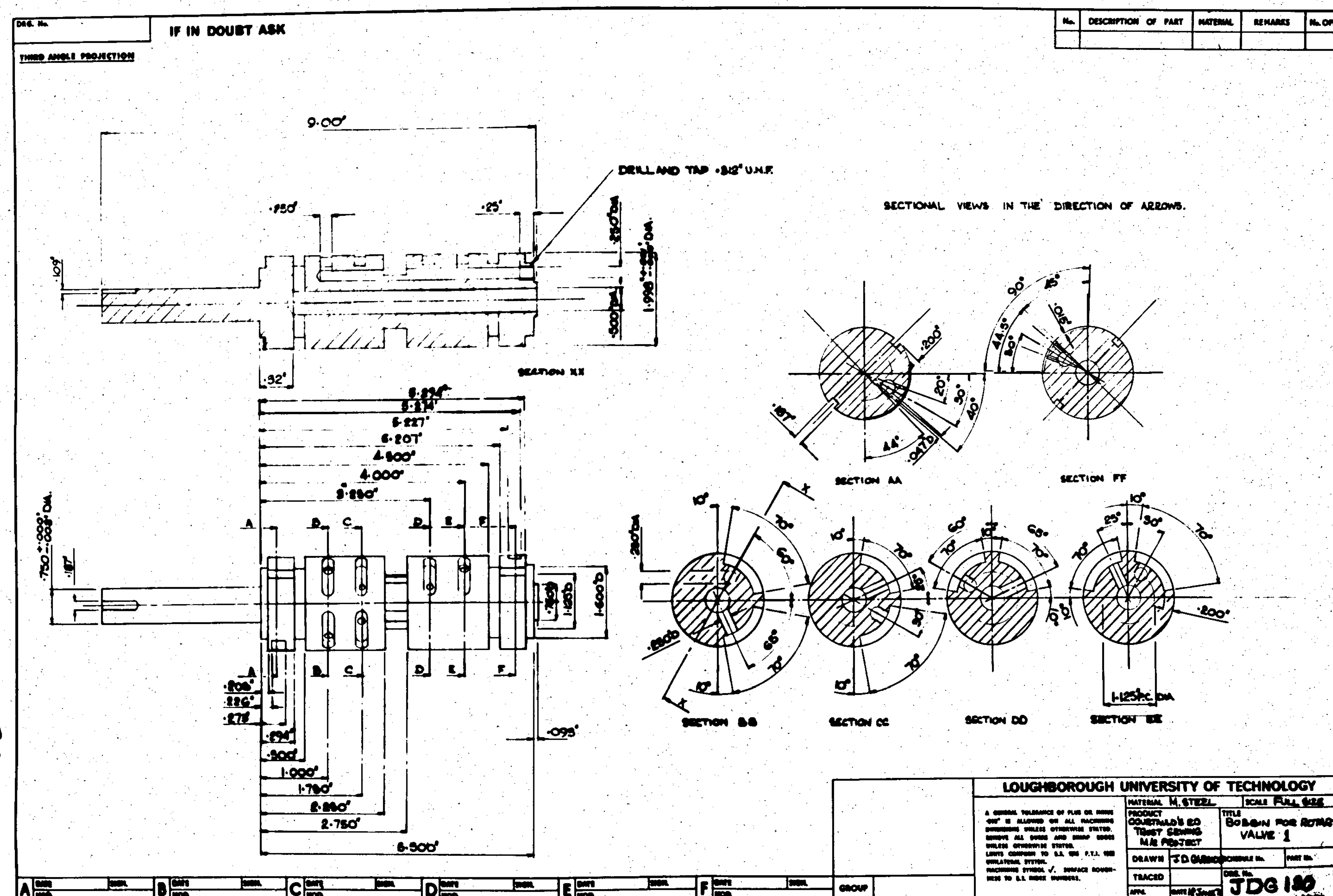

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. | DESCRIPTION OF PART | MATERIAL | REMARKS | No. OFF
DRILL AND TAP ·312" U.N.F.
SECTIONAL VIEWS IN THE DIRECTION OF ARROWS.
SECTION XX
SECTION AA
SECTION FF
SECTION BB
SECTION CC
SECTION DD
SECTION EE
1·125 P.C. DIA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL M. STEEL
SCALE FULL SIZE
PRODUCT COURTAULD'S 2D TWIST SEWING M/C PROJECT
TITLE BOBBIN FOR ROTARY VALVE 1
DRAWN J.D. GARSIDE
TRACED
DRG. No. JDG 180
GROUP

FIG. 86

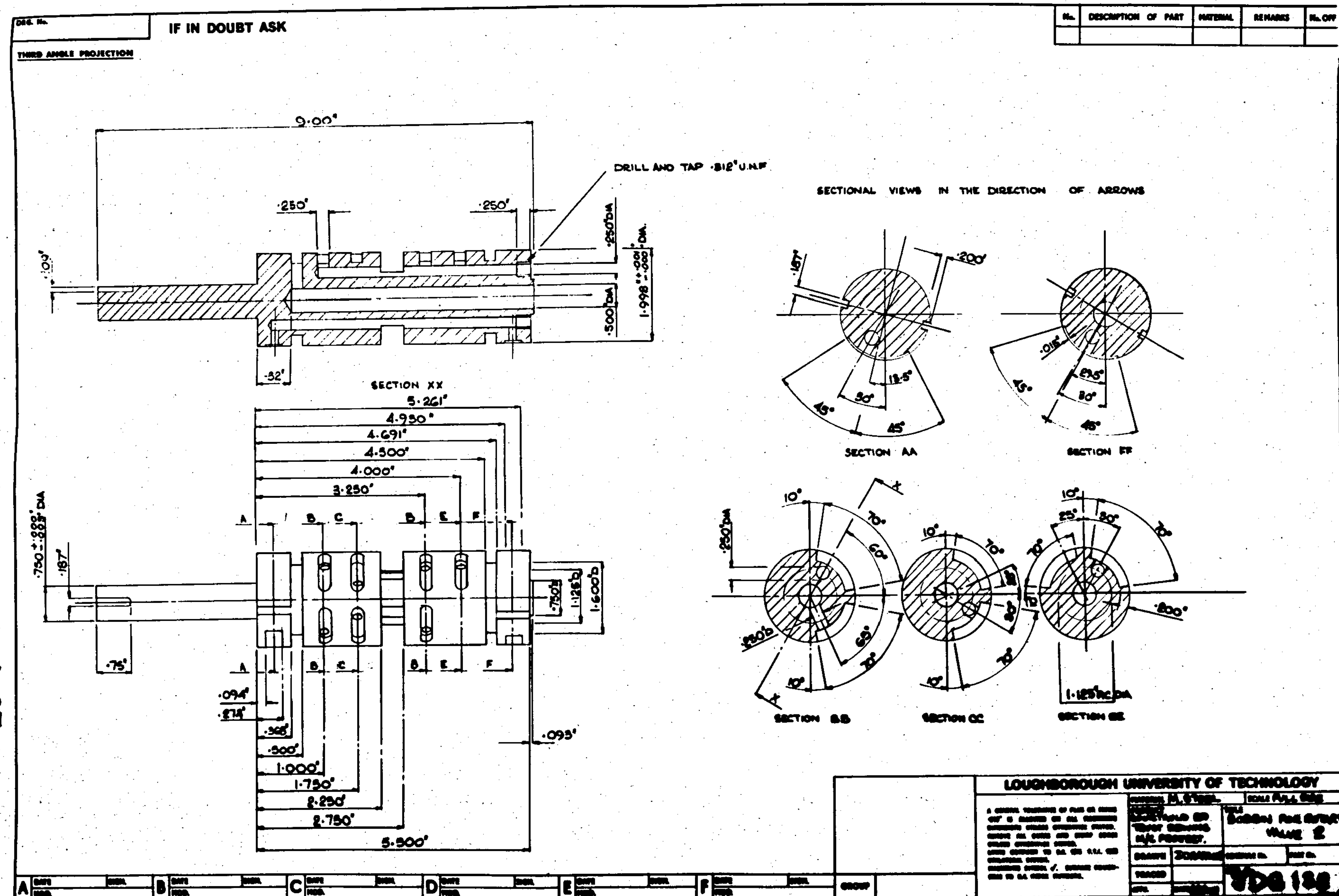

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
DESCRIPTION OF PART
MATERIAL
REMARKS
9·00"
DRILL AND TAP ·312" U.N.F.
·250"
·250"
·250" DIA
·500" DIA
·32"
SECTION XX
5·261"
4·950"
4·691"
4·500"
4·000"
3·250"
·75"
·094"
·500"
1·000"
1·750"
2·250"
2·750"
5·500"
·093"
·750"
1·125"
1·600"
SECTIONAL VIEWS IN THE DIRECTION OF ARROWS
·200"
45°
30°
13·5°
SECTION AA
SECTION FF
10°
70°
60°
·250" DIA
SECTION BB
SECTION CC
SECTION EE
·800"
1·125" PC DIA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
SCALE FULL SIZE
DRAWN
TRACED
GROUP

FIG. 87

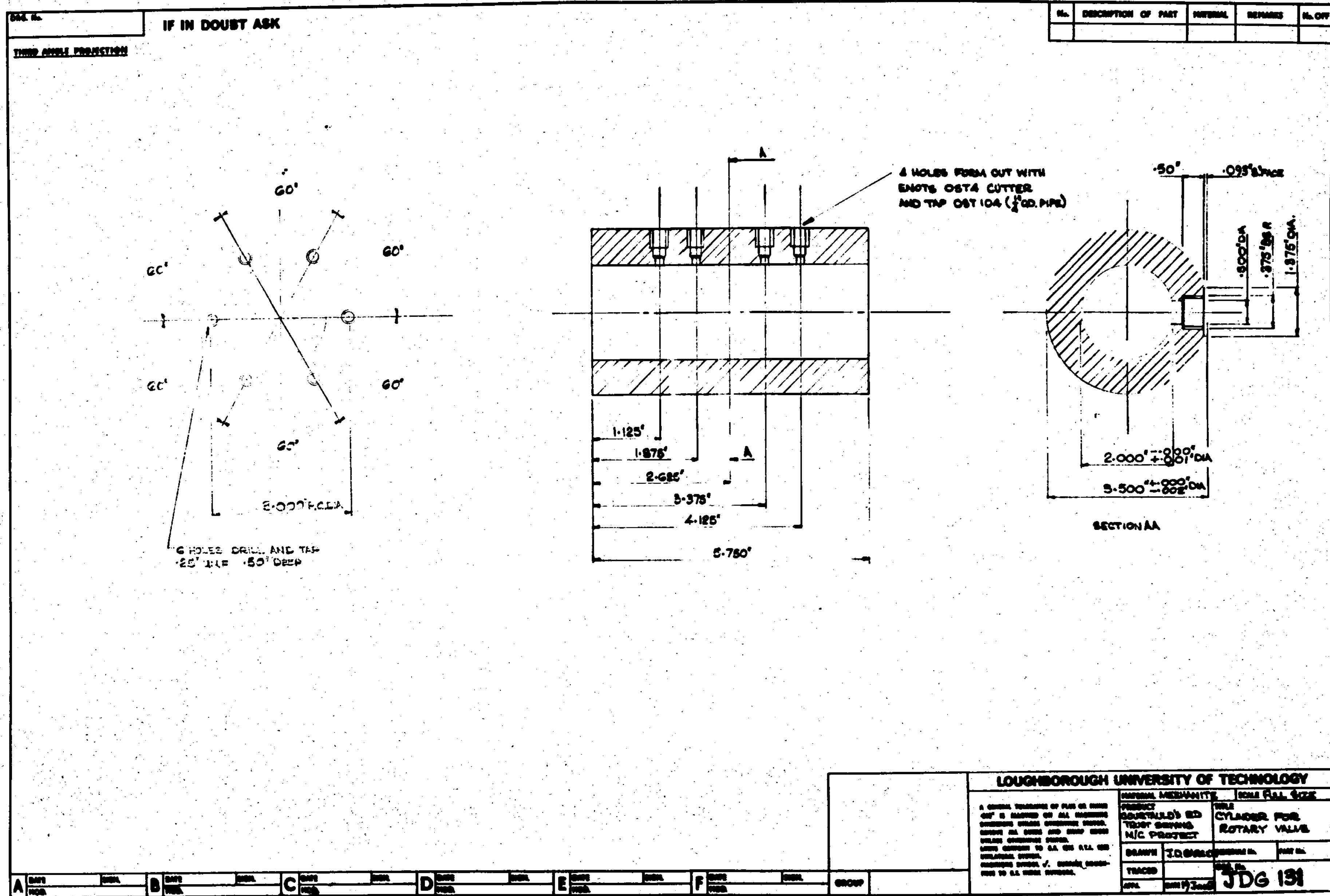

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. DESCRIPTION OF PART MATERIAL REMARKS No. OFF
4 HOLES FORM OUT WITH
ENOTS OST4 CUTTER
AND TAP OST 104 (¼" OD. PIPE)
6 HOLES DRILL AND TAP
·25" UNF ·50" DEEP
1·125"
1·875"
2·625"
3·375"
4·125"
5·750"
·50"
2·000" DIA
3·500" DIA
1·375" DIA
·500" DIA
SECTION AA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL MEEHANITE
SCALE FULL SIZE
COURTAULDS ED
TWIST DRIVING
N/C PROJECT
CYLINDER FOR
ROTARY VALVE
DRAWN J.D.GIBBS
TRACED
JDG 131
GROUP

FIG. 88

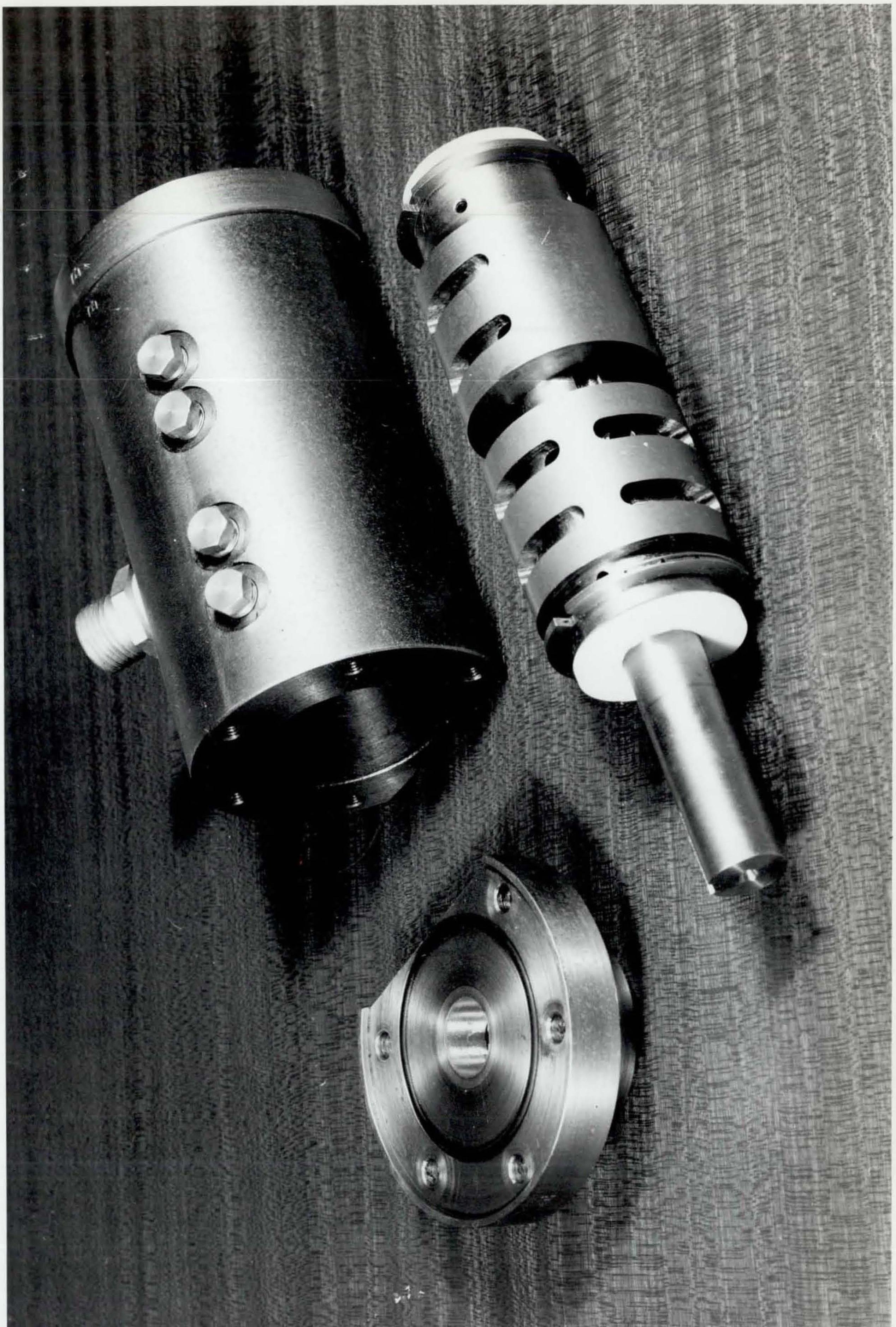


FIG. 89 TEST ROTARY VALVE, BOBBIN 1, SELF BALANCING

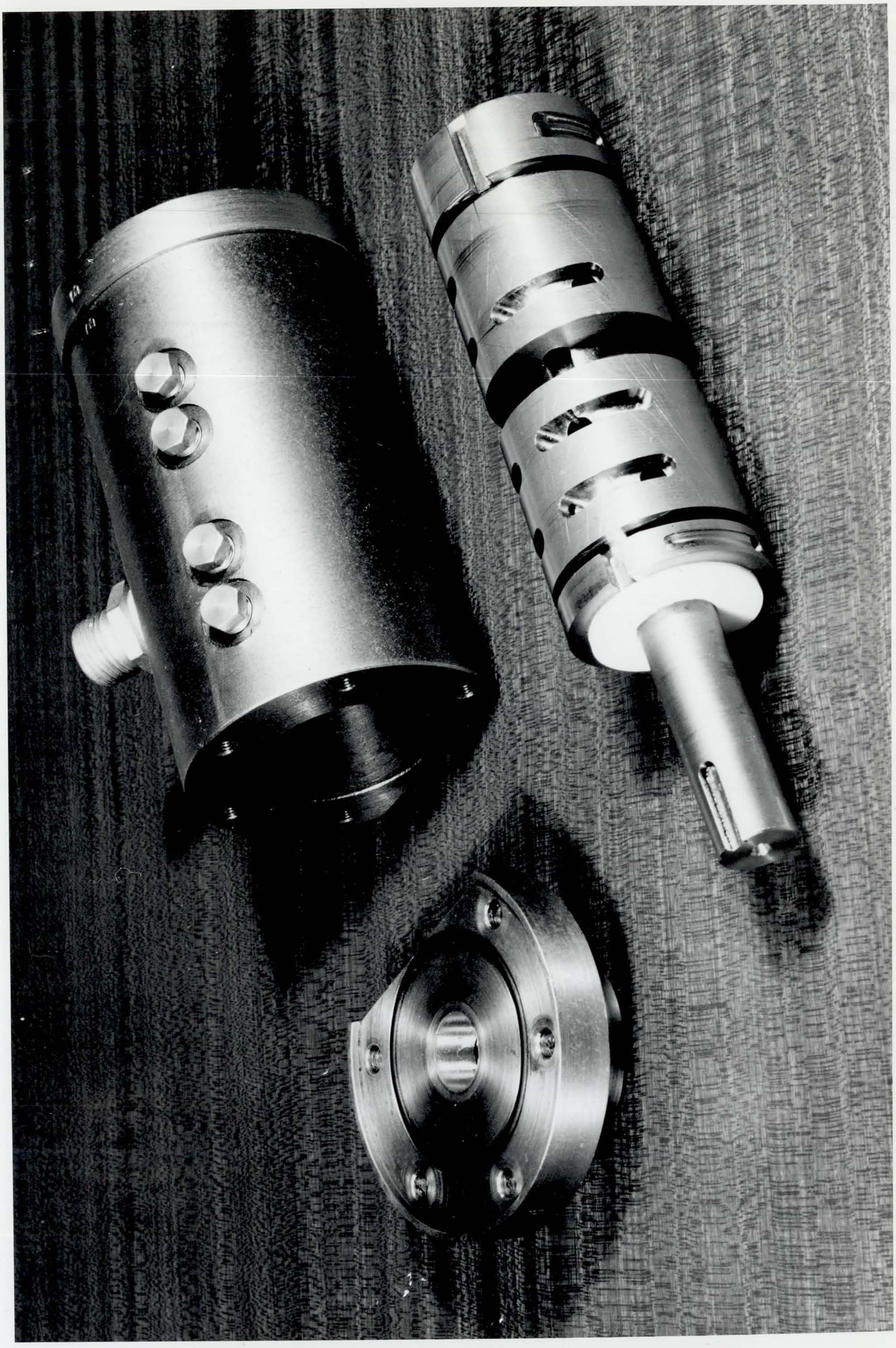


FIG. 90 TEST ROTARY VALVE, BOBBIN 2, WITH COMPENSATING PADS

generator on the back of the hydraulic motor, so enabling the three main parameters of rotary valve performance to be measured. To obtain performance data under the same conditions as those assumed in the theoretical predictions, the supply ports from the valve were blocked thus creating an ideal running condition with no pressure fluctuation. The complete system was run for a period of three hours to enable the rotor to lap into the cylinder and to allow the shaft seal to polish the shaft. It was during this running-in period that the torque fluctuations could be examined under constant running conditions and it was noted that the torque requirement was gradually decreasing. A second run from cold under the same conditions revealed a similar performance pattern, thus indicating that a system parameter was again changing. A series of viscosity measurements were taken of the Tellus 27 hydraulic oil, revealing dramatic viscosity fluctuation in relation to a rise in temperature.

To obtain meaningful results, the system was run for a short period prior to the readings being taken, enabling an ambient running temperature to be obtained. The viscosity of the oil was then measured and the following readings of valve performance noted (see Table 10). The speed of rotation of the valve was adjusted to a known value, and the supply pressure increased by increments of 100 lbf/in^2 to enable torque requirements to be tabulated. This procedure was repeated over a range of valve input speeds. A second viscosity reading was then obtained to enable a mean value to be found, and to be used in calculating the predicted results. A further test

TABLE 10

PRACTICAL RESULTS OBTAINED FROM ROTOR 1, SEE DRAWING J.D.G. 130. THE RESULTS GIVE THE TORQUE (LBF-IN) REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS.

PRESSURE LBF/IN^2 \ SPEED REVS/MIN	100	200	300	400	500	600	700	800	900	1000
100	3.0	5.5	7.2	9.0	9.9	10.8	11.4	11.7	12.0	12.3
200	3.1	5.6	7.3	9.0	10.0	10.9	11.4	11.7	12.0	12.3
300	3.2	5.6	7.3	9.1	10.0	11.0	11.5	11.7	12.0	12.3
400	6.0	5.6	7.3	9.1	10.1	11.0	11.5	11.7	12.0	12.4
500	8.4	5.7	7.4	9.2	10.1	11.0	11.5	11.7	12.0	12.5
600	9.6	6.0	7.5	9.6	10.4	11.4	11.7	11.7	12.0	12.6
700	10.8	8.4	9.6	10.2	11.0	11.7	12.0	12.0	12.2	12.7

of valve performance was made by removing the plugs from the valve distribution ports and replacing them with .25 inch diameter tubing going directly back to tank. Under these operating conditions, severe pressure fluctuations were introduced into the valve and it was thought that valve performance may be impaired. However, a series of readings were obtained, taking first the valve under no flow conditions followed by the readings for maximum flow conditions and no meaningful increase in torque requirements could be found. This indicated that the pressure fluctuations had a direct influence on the pressure distribution within the valve, and in consequence the system remained in balance. The rotor within the valve was replaced by Bobbin 2 and a similar test procedure conducted. The results of performance were noted and can be seen in Table 11. These results have also been plotted in graph form and can be seen in figures 91 and 92. The most noticeable feature is that the torque curve is almost independent of supply pressure, indicating that the internal forces must have been partially balanced. However, these results will be discussed in greater detail when comparisons are drawn with the predicted results.

5.8.4. Input Data for Computer Programme used in Theoretical Predictions.

To obtain the theoretical predictions, the physical parameters of the system had to be supplied as input data to the computer programme. This involved measuring the diameter of the bobbin and the cylinder to obtain an accurate value for the radial clearance. The Department had no standard measuring

TABLE 11

PRACTICAL RESULTS OBTAINED FROM ROTOR 2, SEE DRAWING J.D.G. 132, THE RESULTS GIVE THE TORQUE (LBF-IN) REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS.

PRESSURE LBF/IN^2 \ SPEED REVS/MIN	100	200	300	400	500	600	700	800	900	1000
100	3.6	6.9	8.4	10.2	11.7	12.7	14.6	15.6	18.0	20.5
200	4.2	7.1	8.6	10.2	11.7	12.8	14.6	15.6	18.0	20.5
300	4.3	7.1	8.6	10.3	11.8	12.8	14.7	15.7	18.5	20.8
400	5.4	7.1	8.6	10.3	11.8	12.9	14.7	15.7	18.7	21.0
500	6.6	7.2	8.7	10.3	11.9	12.9	14.7	15.7	18.8	21.3
600	9.0	7.2	8.7	10.4	12.0	13.0	14.8	15.8	19.0	21.5
700	9.6	7.4	8.8	10.4	12.0	13.1	14.8	15.9	19.2	21.7

A GRAPHICAL REPRESENTATION OF THE PRACTICAL RESULTS OBTAINED FROM BOBBIN 1.

THE RESULTS SHOW THE TORQUE REQUIRED TO ROTATE THE BOBBIN AT VARIOUS SUPPLY PRESSURES.

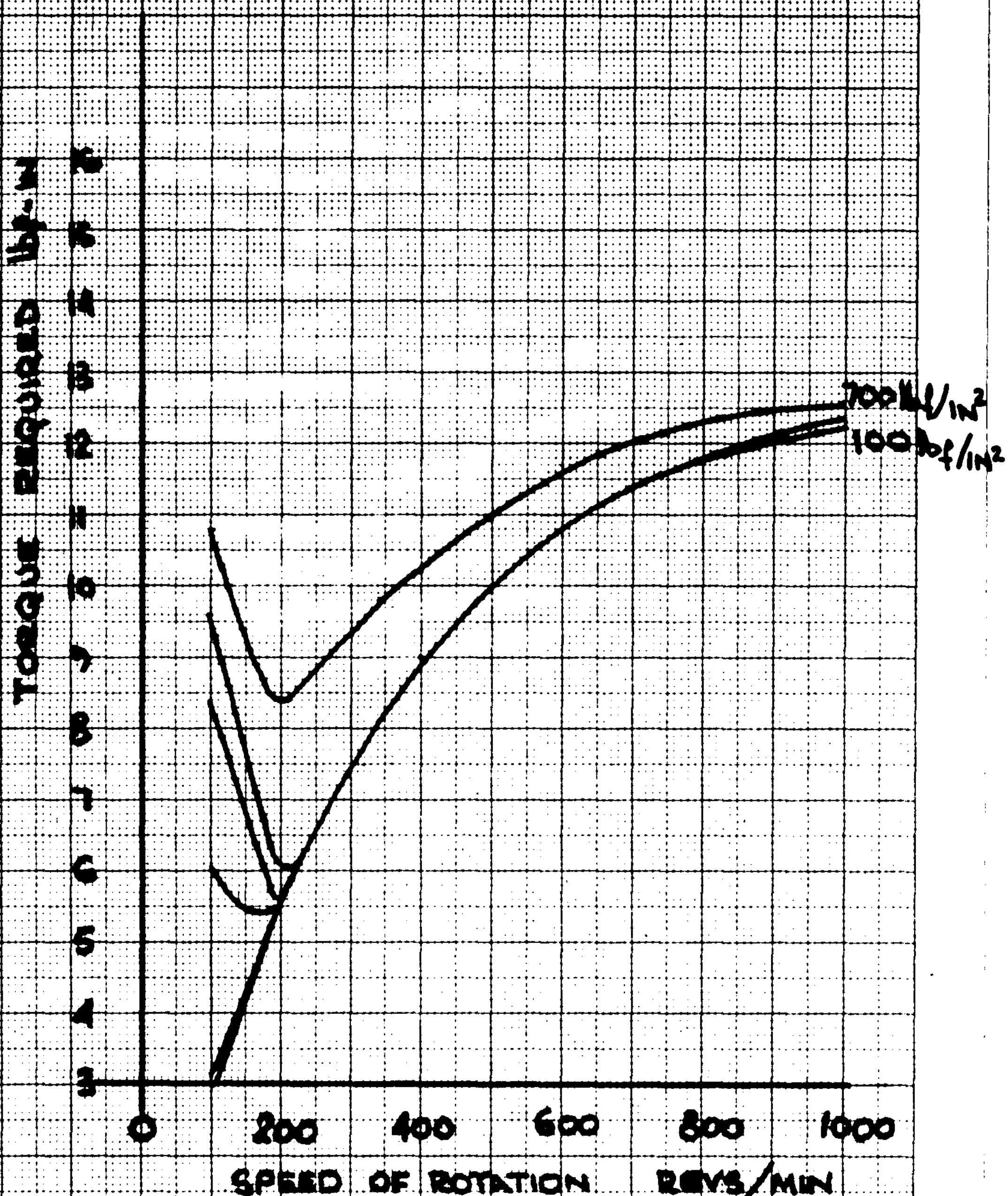


FIG. 91

A GRAPHICAL REPRESENTATION OF THE PRACTICAL RESULTS OBTAINED FROM BOBBIN 2. THE RESULTS SHOW THE TORQUE REQUIRED TO ROTATE THE BOBBIN AT VARIOUS SUPPLY PRESSURES.

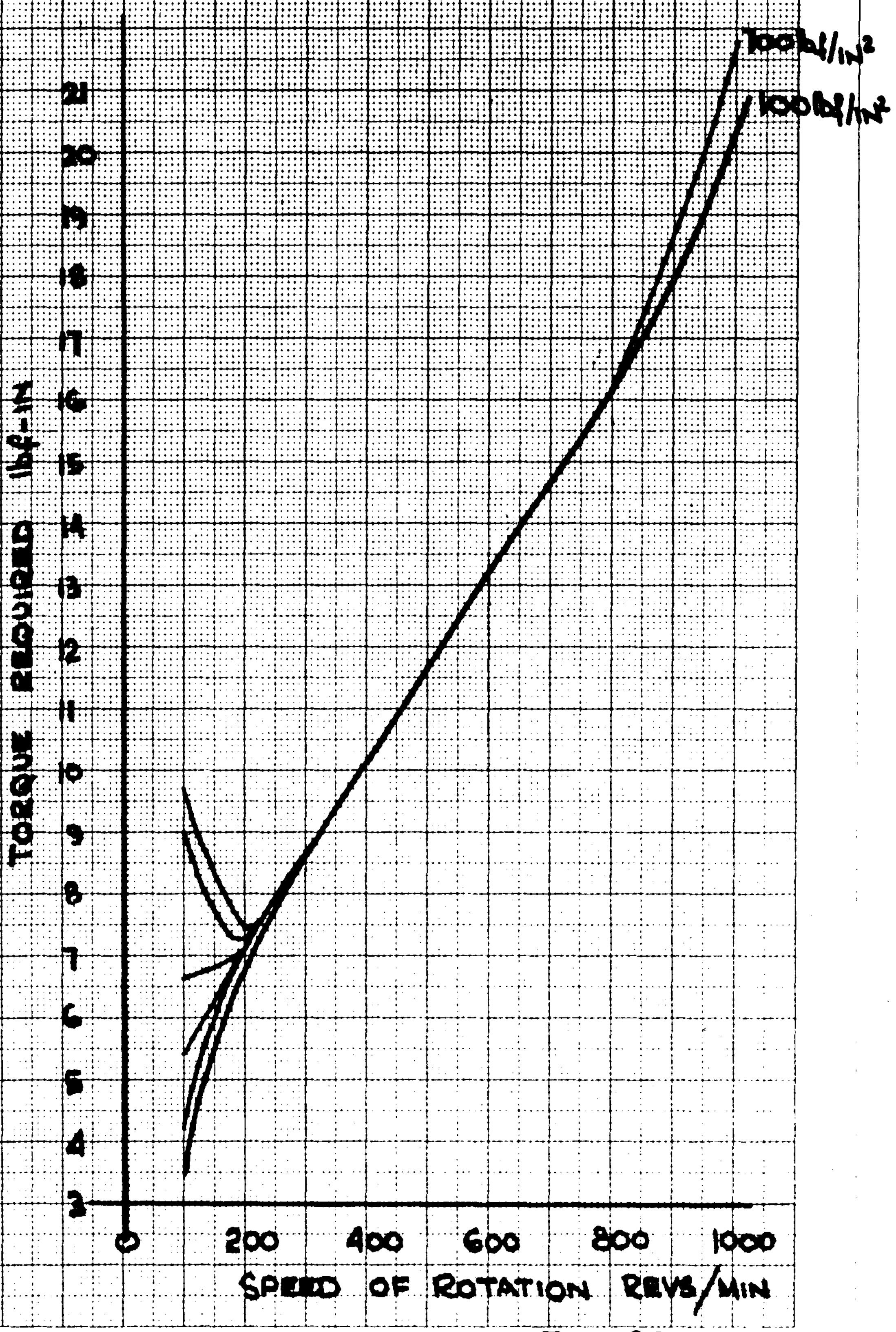


FIG. 92

equipment so the following results were obtained with the aid of the Production Engineering Department. The measurement taken after a considerable stabilizing period was the degree of roundness using a Rank Taylor Hobson, Talyrond which gave the following results;

Cylinder Bore:	0.0003 inches ovality
Bobbin 1:	0.00005 inches ovality
Bobbin 2:	0.00005 inches ovality

These measurements were necessary to confirm that all the components were round. The internal cylinder bore was measured using a three point micrometer and a series of readings gave values of 2.0016, 2.0020, 2.0018 for a measure of the internal diameter, producing a mean value of 2.0018 inches diameter. The diameter of the bobbins were measured using an optical comparitor technique and produced readings of:-

Bobbin 1:	2.0006 inches diameter
Bobbin 2:	2.0004 inches diameter.

The end clearance within the rotary valve was measured using a depth gauge micrometer and found to be .008 inches giving an end clearance of .004 inches.

5.8.5. Physical Parameters affecting Input Data.

The physical dimensions for the radial clearance measured could not be used directly because no account had been taken of the dimensional changes that could occur under operating conditions. For example, the effects of thermal expansion of the two components had to be examined to prove that it had no significant effect upon the radial clearance. In this instance using steel and cast iron, the difference between the two values for the coefficient of expansion was 1.5×10^{-6} per deg C so this factor was neglected. A second parameter to be examined was the effect of internal supply pressure in the valve. The expansion of the radial clearance was calculated using the Strength of Materials Theory for thick cylinders. This theory is well-established and can be seen in reference (10). Applying this theory to the rotary valve cylinder:-

The radial stress $$\sigma_R = A_1 \pm \frac{B_1}{R^2} \qquad \text{\_\_\_\_\_\_\_\_} (22)$$

where R = the radius

Using the dimensions for the cylinder to find the constants of integration A_1 and B_1

σ_R = -p when R = 1.00 in

where p = the internal pressure

σ_R = 0 when R = 1.187 in.

(The effective thickness of the cylinder was .187 in due to the flat surface being machined for mounting purposes).

Thus, when the supply pressure to the valve was 350 lbf/in^2 (the average supply pressure)

A_1 = 1,030

B_1 = -1,450

which produces a maximum hoop stress σ_C = 2,480 lbf/in^2 at the inner surface of the cylinder.

The increase in radius of the cylinder δR was given by the expression

$$\delta R = \frac{R}{M}\left[\sigma_C - \mu(\sigma_R)\right] \qquad \text{\_\_\_\_\_\_\_\_} (23)$$

where M = Modulus of Elasticity = 20×10^6 lbf/in

μ = Poisson's ratio = .26

δR = .00013 (expansion)

The effect of pressure on the bobbin could be calculated using the fact that it was a solid cylinder and the maximum hoop stress would be $\sigma_c = -p$ = 350 lbf/in^2 giving R = -0.00002 (compression) for an Elasticity Modulus of 28×10^6 lbf/in^2. This gave a total increase in radial clearance of 0.00015 inches which was added to the measured radial clearance of the valve.

This produced average values for the radial clearance of Bobbin 1 as .00075 inches and Bobbin 2 as .00085 inches.

Had the expansive effect of applying the internal pressure been realised at the design stage, an alternative method of securing the test valve prior to fitting it to the sewing machine would have been adopted, so that the cylinder wall would not have been weakened for the test. However, by increasing the radial clearance the amount calculated at the mean supply pressure, partially compensated for this discrepancy. An alternative method of measuring the radial clearance of the valve would have been to secure the cylinder, and jack the bobbin through the extremities of its movement. This movement, which could be measured with a dial gauge would give a direct reading for the mean radial clearance with all the discrepancies taken into account. Viscosity readings of the hydraulic oil had been taken during the practical tests and from these values a mean figure of 35 centipoise was selected as the average value.

5.8.6. Torque Predictions.

In order to provide a series of torque predictions at various supply pressures, the two computer programmes were modified to calculate the pressure distribution for unit supply pressure, and to scale the results accordingly by means of a 'DO' loop. This reduced the overall computing time, because the relaxation routine had only to be completed on the initial unit pressure, as opposed to re-calculating the complete solution for each individual pressure level. The programme for the compensating pads (which calculated pad dimensions) had to be further modified to accept pad dimensions as data and to calculate the values of the torque required to overcome the viscous friction. These two modified programmes are shown in Appendices III and IV.

A summary of the results from the computer programme can be seen in Tables 12 and 13. Table 12 and figure 93 show the predicted results for Rotor 1. These results which give the predicted torque required to turn the rotor, are given in tabular and graph forms. Similar results for Bobbin 2 are shown in Table 13 and figure 94. Table 14 shows the power requirement for Bobbin 2.

5.8.7. Comparison of Predicted and Practical Results

Comparing the results shown in figures 91 and 92 it can be seen that the general range of values is similar, and that the increase in torque due to pressure is as predicted in the theoretical solution. The variation in torque at speeds below 200 revolutions per minute is due to the rotor not running

TABLE 12

THEORETICAL PREDICTION FOR ROTOR 1, THE RESULTS GIVING THE TORQUE (LBF-IN) REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS

PRESSURE LBF/IN^2 \ SPEED REV/MIN	100	200	300	400	500	600	700	800	900	1000
100	2.06	3.97	5.86	7.76	9.66	11.56	13.46	15.36	17.26	19.16
200	2.24	4.13	6.04	7.93	9.82	11.74	13.63	15.53	17.43	19.34
300	2.40	4.30	6.20	8.10	10.00	11.90	13.81	15.70	17.60	19.50
400	2.57	4.47	6.37	8.27	10.18	12.07	13.97	15.87	17.77	19.67
500	2.74	4.64	6.54	8.44	10.34	12.24	14.14	16.04	17.94	19.84
600	2.91	4.81	6.71	8.61	10.51	12.41	14.31	16.21	18.11	20.01
700	3.08	4.98	6.88	8,78	10.68	12.58	14.48	16.38	18.28	20.18

TABLE 13

THEORETICAL PREDICTIONS FOR ROTOR 2, THE RESULTS GIVING THE TORQUE (LB-IN) REQUIRED TO ROTATE THE BOBBIN AT THE INDICATED SPEED AND PRESSURE SETTINGS.

PRESSURE LBF/IN^2 \ SPEED REV/MIN	100	200	300	400	500	600	700	800	900	1000
100	2.34	4.42	6.49	8.57	10.65	12.72	14.80	16.88	18.95	21.03
200	2.61	4.69	6.76	8.84	10.92	12.99	15.07	17.15	19.22	21.30
300	2.88	4.96	7.04	9.11	11.19	13.27	15.35	17.42	19.50	21.57
400	3.16	5.23	7.31	9.38	11.46	13.54	15.61	17.69	19.77	21.84
500	3.43	5.50	7.58	9.66	11.73	13.81	15.88	17.96	20.04	22.11
600	3.70	5.77	7.85	9.93	12.00	14.08	16.16	18.24	20.31	22.38
700	3.97	6.05	8.12	10.20	12.28	14.35	16.43	18.51	20.58	22.66

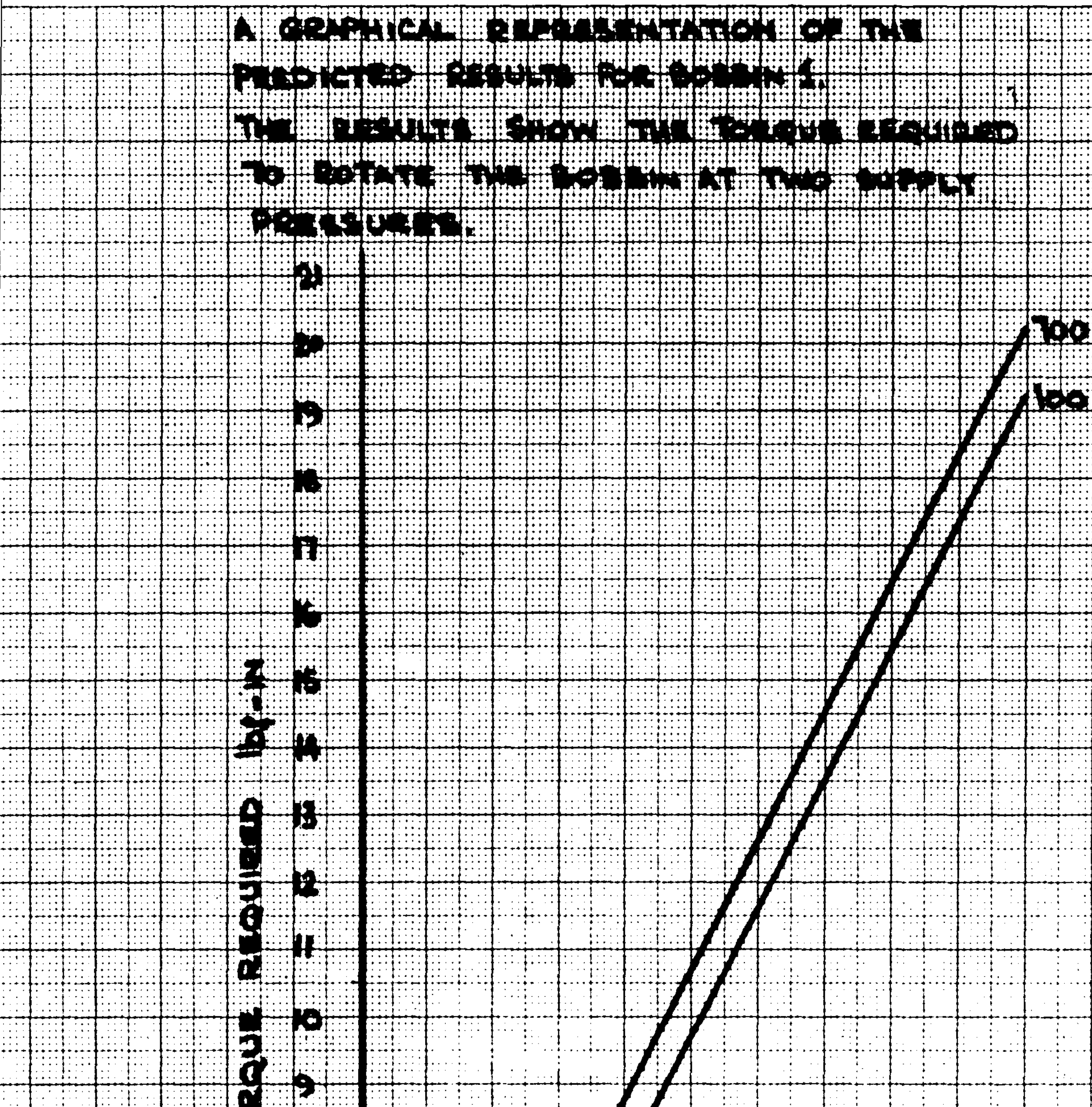

A GRAPHICAL REPRESENTATION OF THE PREDICTED RESULTS FOR BOBBIN 1.
THE RESULTS SHOW THE TORQUE REQUIRED TO ROTATE THE BOBBIN AT TWO SUPPLY PRESSURES.
TORQUE REQUIRED lbf-in
SPEED OF ROTATION REVS/MIN
700 lbf/in²
100 lbf/in²
0
200
400
600
800
1000
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21

FIG. 93

A GRAPHICAL REPRESENTATION OF THE PREDICTED RESULTS FOR BOBBIN 2. THE RESULTS SHOW THE TORQUE REQUIRED TO ROTATE THE BOBBIN AT TWO SUPPLY PRESSURES.

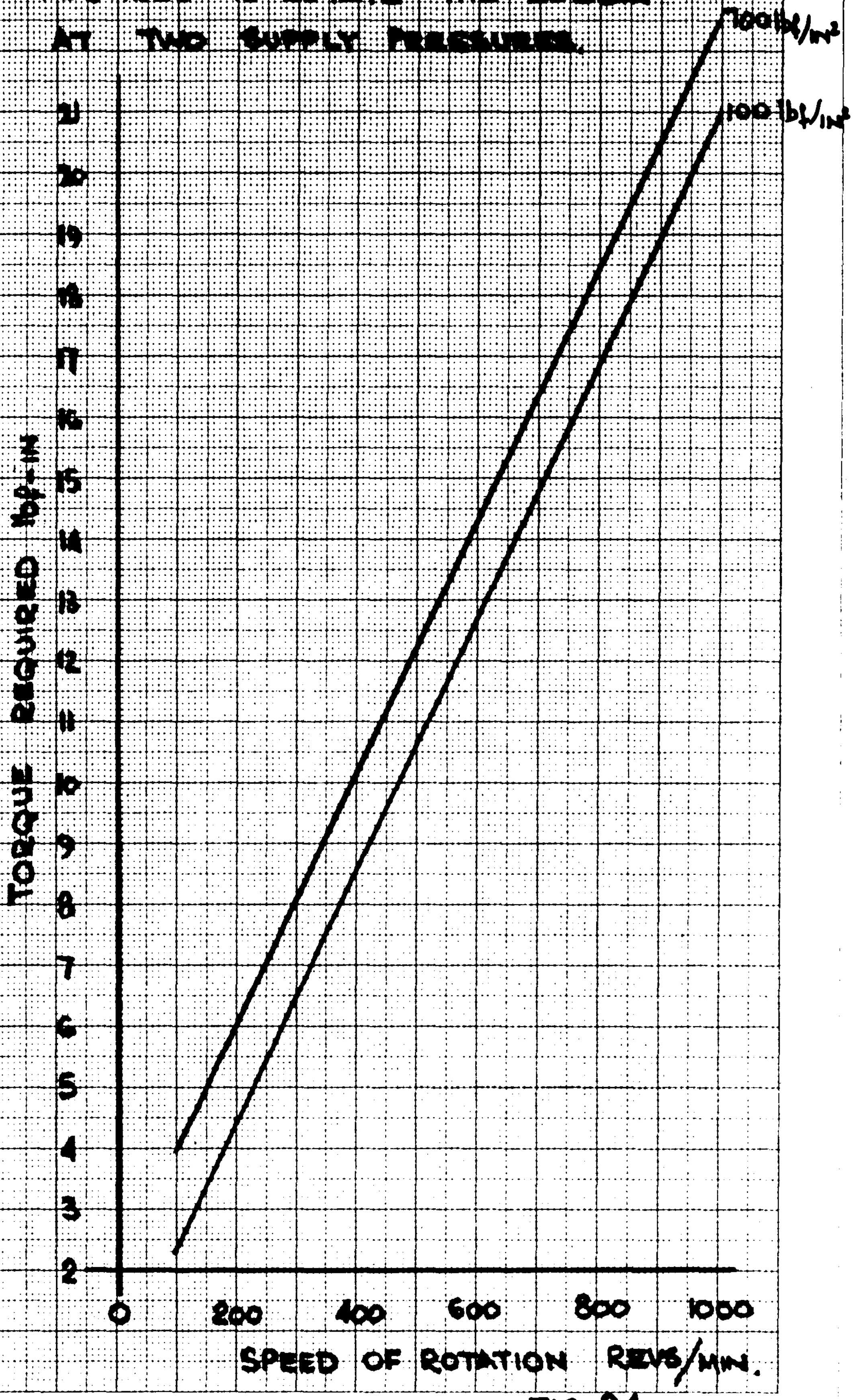


FIG. 94

T A B L E 14

THE PREDICTED HORSE POWER REQUIRED TO ROTATE BOBBIN 2. WITH SUPPLY PRESSURES OF 100 LBF/IN^2 AND 700 LBF/IN^2 AT VARIOUS INPUT SPEEDS

SPEED OF ROTARY VALVE REVS/MIN	VALVE SUPPLY PRESSURE 100 LBF/IN^2		VALVE SUPPLY PRESSURE 700 LBF/IN^2	
	TORQUE LBF - IN	POWER	TORQUE LBF - IN	POWER
100	2.34	.0037	3.97	.0062
200	4.42	.0140	6.05	.0191
300	6.49	.0308	8.12	.03866
400	8.57	.0543	10.20	.0647
500	10.65	.0844	12.28	.0973
600	12.72	.1210	14.35	.1365
700	14.80	.1643	16.43	.1824
800	16.88	.2141	18.51	.2348
900	18.95	.2704	20.58	.2937
1000	21.03	.3335	22.66	.3593

centrally as assumed by the theory, and at theselower speeds the hydrodynamic action within the valve must be causing the increased torque. The curvature as seen on the practical results graph, can only be explained by a viscosity change. These results were obtained in a single run starting with low speeds and using the pressure setting as the variable, consequently the oil was gradually changing in viscosity. In the second series of tests with Bobbin 2, the practical and theoretical results are in much closer agreement and this could be due to obtaining the torque readings at two separate sessions thereby keeping the fluid temperature within a narrower band. The first series of tests were partially repeated and in fact found to be in closer agreement with the theoretical predictions, but the results were not modified because they are typical of the operating results to be found in practice under normal working conditions.

The results from the second rotor, Bobbin 2, are the more significant because the pressure-generating section had an inherent out-of-balance that had to be compensated by end pads. These results show that the compensation technique is working as predicted and the torque requirements for a rotary valve of this type can be calculated by using simple hydrodynamic bearing theory. Figure 95 which shows the theoretical and practical results plotted on the same graph, gives an indication as to the correlation between the results. The torque requirement at speeds below 200 revolutions per minute again indicates that the rotor is not running centrally within the bore, but once having achieved this critical speed, the curve closely resembles

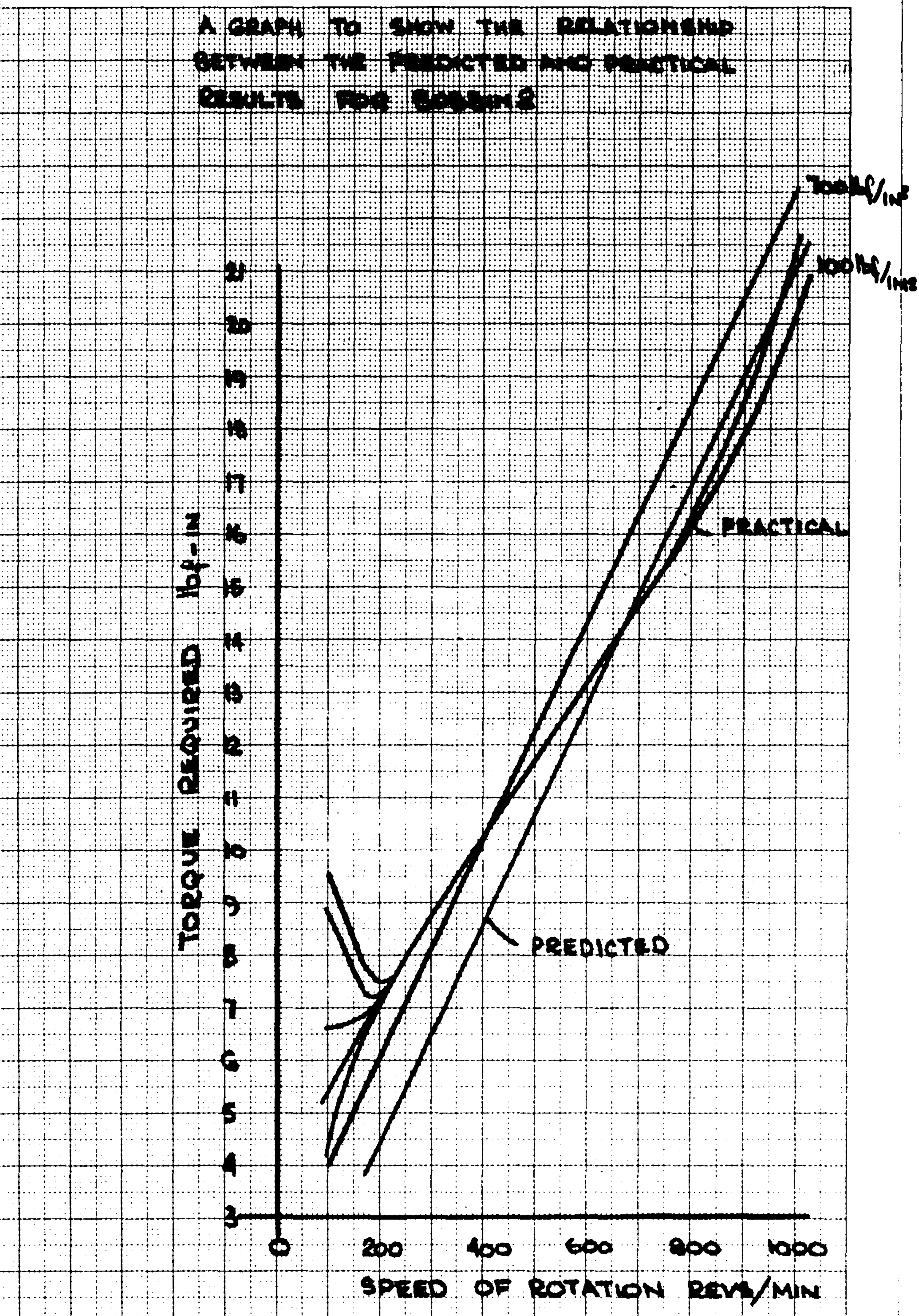

A GRAPH TO SHOW THE RELATIONSHIP BETWEEN THE PREDICTED AND PRACTICAL RESULTS FOR BOBBIN 2
700 lbf/in²
1000 lbf/in²
PRACTICAL
PREDICTED
TORQUE REQUIRED lbf-in
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
0
200
400
600
800
1000
SPEED OF ROTATION REVS/MIN

FIG. 95

the predicted line. These results also represent the optimum performance for the rotary valve because the theory assumes ideal bearing running conditions, that is, of a non-loaded shaft, so indicating that the design of the rotary valves using this technique has been optimised. A measure of the improvement in performance can be gauged by comparing the resultsfor the sewing machine rotor shown in figure 80 with the newly-designed Rotor 2. The physical linear dimension for the sewing machine rotor is much shorter, thus reducing the contact area, but the torque meter readings taken during practical tests are:-

At a speed of 600 revolutions per minute with a supply pressure of 400 lbf/in^2:-

For the Sewing Machine Rotor: 29.2 lbf/in

For the Rotor with Bobbin 2: 13.2 lbf/in

This represents an over 100% improvement in valve performance and fully justifies the time spent in analysing the performance of rotary valves.

5.9. Further Work Based on Rotary Valves.

Throughout this project, rotary pulse-generating valves have been applied to novel hydraulic systems and if this type of work were to be continued because of a commercial demand for them, then further analytical work maybe justified. This work has clearly shown how analytical design can improve the performance of the valve and any future valve designers need to use the compensating pad technique for balancing. This involves using the digital computer programmes given in Appendices I and II. The predicted values of the torque requirement would be a useful guide in selecting the size of the drive required to turn the valve, but to finally select and specify a drive prior to knowing the actual physical parameters of the valve could be rather difficult. This reservation is made based on the experience of measuring actual running clearances for the bobbin, and in obtaining a realistic value for the viscosity of the oil under true working conditions. While it is comparitively easy to measure to a tenth of one thousandths of an inch the machining of components to that tolerance is comparatively difficult. It must also be remembered that machining marks and ovality in components cannot always be measured with normal measuring instruments, thus making for discrepancies in radial clearance. A useful empirical basis upon which initial calculations could be made (when it is expected that the bobbin will be a smooth running fit in the cylinder), would be to allow .0007 inches of clearance per inch of radius.

One aspect of the rotary valve which has yet to be examined is its flow characteristics. This work would fall into two catagories:- internal and external flow conditions.

5.9.1. Internal Flow.

The study of internal flow, (leakage) would have to be based on an analytical solution for the pressure distribution round the rotary valve. The mathematical model of the pressure variations within the radial clearance could be used to predict the leakage flow both radially and axially. (The basic equations can be found in Ref 1.).

This study would involve similar techniques to those used for predicting the torque requirement of a rotary valve.

A design study could then follow enabling the radial clearance within the valve to be optimised. The power required to rotate the valve, for various values of radial clearance at a constant speed and supply pressure, could be compared with the power required to supply the leakage flow due to radial clearance of the valve. This aspect of rotary valve performance is at present being undertaken by a final year student at Loughborough University and will provide more data upon which to base future rotary valve designs.

5.9.2. External Flow.

Examining the external flow would involve a study of the transient flow patterns from the supply ports, so enabling the flow function to be expressed correctly for use in classical control theory. A further development could be to derive an overall expression for predicting the total delivery of fluid by the valve, per unit time. This expression would essentially be based upon the internal dimensions of the valve and the type of fluid being delivered. Future developments in valve compensation techniques could involve replacing the compensating pads by other fixed bearings i.e. ball racers, or full hydrostatic bearings. These may offer advantages in particular applications, but it is thought that generally the compensating pad would serve adequately. Other general fields of analytical and practical work could be into examining the effects of individual system parameters such as temperature rise, and internal pressure. Both these parameters could alter the physical dimensions of the valve, so would be worthy of a specialised study. For applications involving high rotor speeds, a system of mechanical balancing, similar to that already used on electrical motors, would have to be devised. This mechanical balancing would reduce the centrifugal forces within the valve, so improving the overall dynamic performance.

To conclude, this study has resulted in the derivation of a design technique that can be applied to produce all pad geometries and dimensions of pulse-generating rotary valves. A general layout plus a method of obtaining optimum performance has been given so enabling this type of valve to be designed using a technological approach in future applications.

6. SUMMARY OF THE MAIN FINDINGS AND CONCLUSIONS

6.1. HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

6.2. HYDRAULIC LOCKSTITCH SEWING MACHINE

6.3. PULSE-GENERATING ROTARY VALVES

6. SUMMARY OF THE MAIN FINDINGS AND CONCLUSIONS

The research undertaken in this project has been concerned with replacing linear mechanical mechanisms by miniature hydraulic actuators. The project, sponsored by Courtauld's Educational Trust Fund, has been directed towards textile machinery, in particular, Knitting and Sewing Machines. However, the project basically demonstrates how the well-established mechanical motions of either a cam or linkage can be replaced by miniature hydraulic actuators.

6.1. Hydraulic Circular Weft Knitting Machine

The first example of this technique was demonstrated on a prototype ninety-six needle hydraulic circular weft knitting machine. This machine had ninety-six independent actuators, placed in a circular configuration at four actuators per inch, controlled by a single pulse-generating rotary valve. This valve powered 25% of the needles at any particular instance, and was capable of sequencing each actuator for ten movements per second. The prototype machine built and tested in the Department provided the basis for the following conclusions:-

(i) Knitting speeds above the accepted maximum running speeds for conventional machines were obtainable. The maximum velocity for needles passing through a cam is regarded as 65 inches per second, which represents a 3.75 inch diameter hosiery machine running at 230 revolutions per minute. The hydraulic devices attached to the needles were capable of driving the prototype machine at a speed equivalent to 50% above this accepted limit and still produce a satisfactory knitted fabric. This increase in speed was due to two factors:-

(ii) The actuators could move the needles faster than was possible using a cam track.

(iii) Only one needle was in the process of knitting at any particular instance.

Tests performed on single hydraulic actuators had demonstrated that cycling rates of 50 hertz were possible, without damaging the actuators. This type of motion was feasible using hydraulic devices, because the driving force, i.e. stored energy in the hydraulic fluid, has a cushioning effect when first applied to the actuator piston, compared to the direct impact of a butt onto a 45° gradient. This comparatively damped movement reduced the shock loading transmitted to both the needle and the yarn. Research into new parabolic cam profiles for knitting machines has been undertaken at various institutions, but the actual increase of cam velocities achieved over the last twenty years on commercial machines is less than 30%. Using the hydraulic actuator technique speed restriction imposed by the cam track has been removed because the equivalent cam velocity of 65 inches per second represents a cycling rate of 3 hertz for each actuator on the prototype machine.

A further advantage of hydraulic actuators was the geometry of the time displacement profile described by the needle. It was noticed that as the needles operated in sequence, one actuator completed its movement in a particular part of the cycle prior to the next one starting. This individual needle movement when pulling the new yarn through the old loop meant knitting could be produced on a single needle. The advantage gained by this action was that problems associated with "pull back" no longer existed. On conventional cam driven machines, the knitting action takes place as the needles traverse the last slope of the cam. (i.e. travelling from the tuck to the miss position). By the nature of the cam track, more than one needle must be moving down this slope at any one instance, thus new yarn when introduced has to be pulled through the hooks of several needles. This function introduces tension in the yarn and research outside this University has shown that 70% of yarn required to form the new loop is obtained from the yarn carrier, while the remaining 30% is pulled from previously formed loops. This undesirable tension imposes restrictions on knitting speeds, because at high operating rates permanent damage may be caused to the needles. Using the single knitting action of hydraulic devices, no yarn restrictions are imparted by other needles, permitting all the yarn required to form the loop to be pulled directly from the yarn carrier.

When knitting had been established using hydraulically-powered needles, a further significant advantage was realised:-

(iv) The number of needles per knitting station could be greatly reduced.

On a conventional knitting machine, the number of needles per knitting station is governed by the parameters of the needle and the length of cam track necessary to move the needle through the complete knitting cycle. This usually creates knitting stations of 30 - 40 needles. Using hydraulic actuators, the displacement was a function of time, and not cam geometry, and since higher velocities for needle movement were possible, the number of needles per knitting station could be reduced. When knitting, the prototype machine had six needles in the extended knitting position followed by six needles in the mid or tuck position, with the other eighty-four needles in the miss position. This arrangement could be modified to; two needles extended, with two needles in the tuck position without altering the dynamic performance of the actuators. (Remembering that each actuator is stationary after completing a movement). Then by allowing four needles to occupy the miss state before repetition, the complete knitting station would comprise of eight needles. Using this configuration, the ninety-six needle machine could be fitted with twelve feeders with a resultant increase in fabric production without increasing the speed of the actuator movement. An equivalent cam machine to the prototype hydraulic machine would be limited to a maximum of four feeders, thus producing only a third as much fabric. Taking the technique to its ultimate limit could provide a knitting station per needle, so creating a circular warp knitting machine. This feature was of major importance, because it permitted a substantial increase in production from the hydraulic machine, using current yarn handling techniques. It is believed that even if it were possible to double the cam velocity on mechanical machines, the associated yarn handling problems would prevent commercial exploitation.

(v) The use of hydraulic actuators permits new control techniques to be applied to the aspect of programming needles for patterning purposes.

Using an actuator per needle, the control of the displacement between the knit, tuck and miss modes can be effected by on-off devices in the supply paths. While the full practical aspects of programming have to be investigated in future work, it was evident from the tests performed that individual needle selection was feasible using the type of actuator already developed.

The practical hardware used in this section of the research can be seen in a film titled "Development of a Hydraulic Knitting Machine". This film shows the general layout of the rig together with a demonstration of knitting at speeds ranging from sixty, to one hundred and eighty rows per minute.

Any benefits to the textile industry derived from this project have yet to be realised. The initial experiments in applying miniature hydraulic actuation techniques to knitting machines provided an indication that it could form the basis for a future generation of knitting machines provided that sufficient time was spent in development.

Future practical work on the prototype machine must be concentrated in three sections:-

(vi) Adjust the time displacement profile of the actuator.

This would be converted to a direct in and out motion. Using this motion, attempts to produce a plain knitted fabric would provide useful knowledge for assessing further developments. If this motion could be used successfully, then future machines required for knitting single jersey could be simplified by using a single hydraulic piston in the actuator. This would reduce the complexity of the hydraulic control aspect considerably.

(vii) Verify that patterned fabrics could be knitted on a hydraulic knitting machine.

To verify that patterned fabrics could be knitted, several solenoid on-off valves could be introduced into the system thereby allowing manual selection of the time-displacement profile.

(viii) Check that the system would operate as a multi-feeder machine.

This check could be made by changing the rotor of the valve, or by designing an integrated valve and actuator block system. The purpose of this test would be to prove that a knitting station could be constituted by as little as eight needles.

All these future areas of research on the prototype hydraulic machine are based upon the rig that has already been built and are currently being investigated.

(ix) Examine rotary valve design procedures.

This work has already been performed and will be reported later in this summary.

(x) Devise and develop a method of integrating the actuator block and rotary valve.

The most convenient method of integrating these two components would be to use the actuator block as the rotor of the valve, with a ported collar around it to supply the hydro-static pressure. As the needle block was rotated the actuators would perform in the desired manner. A design for an integrated system is at present under consideration and details can be found in Part 4.

Once the various aspects of the project have been assessed then a feasibility study will be required to highlight the commercial potential of the numerous advantages offered by hydraulic knitting techniques. It is then possible that the specification for a commercial machine could be derived, and the work to develop it undertaken by a knitting machine manufacturer.

Further ideas to demonstrate the extra versatility of knitting with hydraulic actuators:-

(xi) The bed configuration would no longer be restricted to a circular or flat by necessity of the cam drive. Positioning of the actuators would enable any desired shape to be adopted. This might have applications when knitting a complete garment.

(xii) The needle orientation of the machine could be varied to suit the yarn.

(xiii) The gauge of the machine could be varied around the periphery so allowing different parts of e.g. garments, to be more flexible.

(xiv) To a limited degree, the needles could be moved raidally enabling e.g. garments to be shaped.

(xv) Complete control over each individual needle could be incorporated so enabling infinite patterns to be knitted.

(xvi) Larger gauge machines could be employed e.g. knitting carpets, exploiting the advantages of infinite pattern selection.

Finally, the work presented in this thesis has demonatrated that the advantages to be gained by applying hydraulic actuation techniques to knitting machines could be considerable, provided that initial research is backed by a full development programme to assess their true commercial potential.

A second application of miniature hydraulic actuation techniques to be considered was for a sewing machine. Such machines contain a number of complex machanical mechanisms that culminate in a linear motion.

6.2. Hydraulic Lockstitch Sewing Machine

The technique of a hydraulic sewing action was demonstrated using a lockstitch sewing machine with the needle and thread take-up mechanisms being replaced by miniature hydraulic devices. A pulse-generating rotary valve again acted as the control medium.

This study proved that a lockstitch could be produced on a hydraulic sewing machine and a full demonstration together with the overall layout of the rig can be seen in the film "Development of a Hydraulic Sewing Machine". The main purpose for building the lockstitch machine was to show that sewing using hydraulic sewing heads was possible, and to provide information upon which to base future sewing machine applications.

The benefits to be gained by using hydraulic actuation techniques on sewing machines could be in the following areas:-

(xvii) A decoupling of the mechanical linkage between the needle and thread take-up mechanisms and the hook drive.

This feature would be useful on a moveable co-ordinate sewing head. If a sewing station producing say bed quilts or regular shapes requiring sewing to take place away from the edge of the fabric, then a moveable automatic sewing head would be advantageous. Using hydraulic actuation techniques the sewing head could be compactly made, housing only two acutators and linked only to the base of the machine by four flexible hydraulic connections, provided that external controll systems maintained a register between the needle and the hook.

(xviii) Removing the restrictions of the throat.

This is linked with (xvii) and would provide a method of building deep throated machines without having to alter the size of the mechanisms. The only necessary modification would be increased pipe length from the actuators to the rotary valve.

(xix) Provide a method of reversing the direction of individual components.

In particular types of machines, i.e. chain stitch and overlocking machines, the nature of the mechanical mechanism prevents individual elements from being reversed to form left-hand machines. These opposite hand machines are used when sewing two edges of a garment at the same time. While these machines are available, they tend to be very expensive and unreliable, therefore the application of hydraulic devices would provide a useful alternative method. Using miniature hydraulic actuators, the phasing and direction of the various components can be adjusted by the positioning of the part-circumferencial grooves in the rotary valve.

(xx) Removing the restrictions imposed by the mechanical mechanisms.

It was found when examining the needle slider crank mechnnism that only 25% of the total needle amplitude was required to form the lockstitch. The remaining

75% of the needle movement took place without the needle piercing the fabric in order to re-orientate the needle in readiness for the next stitch. The one inch movement required by the slider crank mechanism could be replaced by a hydraulic actuator having an amplitude of a quarter of an inch. This illustrated that not only must the mechanical motions be examined, but also the mechanics of sewing.

Therefore, the work presented in this thesis is intended to serve as an introduction to hydraulic sewing techniques and to provide a basis for future developments. Advantages could be obtained using hydraulic sewing units, but more research will be necessary before a viable alternative to mechanical systems can be produced.

6.3. Pulse-Generating Rotary Valves

The necessity for the third section of this project had arisen from the previous developments. The two applications of pulse-generating rotary valves has established that they function effectively and possess characteristics not found in conventional hydraulic control devices. Therefore, a design procedure was evolved in order to rationalise the general layout for the valve and establish a technique for optimising its performance.

Until this juncture, rotary valves had been designed solely to generate the pulses required by the external system. It was evident that the power requirements to rotate the bobbin were excessive, increasing proportionally to the supply pressure to the valve. The high power required could only be attributed to one factor, that of an unequal pressure distribution in the radial clearance between the bobbin and the cylinder. Experimental techniques of machining radial clearance grooves to sustain a more equal pressure distribution had produced marginal improvements in performance, but it was evident that to obtain significant benefits an analytical approach would be required. Estimates regarding the volume flow rates distributed by the valve could be obtained using standard theory for flow in pipes, but no direct method of calculating the torque required to rotate the valve bobbin was available. This resulted in an analytical study to:-

(xxi) Develop a method of balancing the internal forces of the valve.

(xxii) Provide a technique for estimating the power required to rotate the valve bobbin.

The uneven pressure distribution in the radial clearance of the valve was inherent due to the radial part-circumferential grooves that generated the pressure and exhaust pulses. The effect of this pressure distribution could be represented by a resultant force of unknown magnitude and direction. Therefore the initial task was to derive a method of calculating the magnitude and direction of this unbalanced force in order to compensate for it. The annulment of this force could be attempted by using load-carrying bearings (i.e hydrostatic or roller bearings) or by using high pressure pads to create equal and opposite compensating forces. The second method was adopted because the out-of-balance force compensation could be made integral with the rotor. Thus, the general layout for the rotor was to site the pulse-generating section between two compensating pads.

To calculate the size and radial position of the compensating pads, the solution for the pressure distribution in the radial clearance was calculated by solving a differential equation derived from theory related to hydrodynamic bearings. The solution was obtained by using a relaxation procedure on a digital computer. Knowing the pressure distribution, the forces on the rotor could be calculated. By resolving the elements of force in a horizontal and vertical direction and by taking moments about each end of the rotor, the magnitude and direction of the resultant end reactions could be found. The magnitude of the force was then used to calculate the dimension for the compensating pad, again using a solution to the pressure distribution equation. This design technique was tested by designing a rotary valve and results successfully showed that the power requirement for the rotor, when compared with similar valves designed previously, was halved.

Having calculated the pressure distribution around the valve, estimation of the torque requirements to rotate the bobbin was necessary. This involved using the differential pressures between each element in a general expression and summing to find the total torque required. This calculation was also performed on the computer.

The correlation between calculated and practical results agreed to within an error of 10% when account had been taken of the variable physical parameters experienced in practice. Parameters for radial clearance and viscosity had to be assumed constant in the theoretical expression. In practice, the radial clearance (which had to be expressed to an accuracy of

1×10^{-5} inches) was difficult to measure to such fine limits and no method of making allowances for the ovality and taper on the components was feasible. Other system variables affecting the radial clearance were the expansion of the cylinder due to the internal supply pressure and the expansion of the components due to thermal effects. The value for the viscosity of the oil had to be averaged for the series of tests and assumed constant. Viscosity readings showing a 60% fluctuation due to temperature rise in the oil, were recorded, (although during the series of tests for Bobbin 2 care was taken to maintain a constant temperature). These variations in valve parameters could be allowed for in the computer programme, but would increase considerably the complexity of the pressure distribution differential equation. This was not deemed necessary in this application due to the large number of physical variables. To solve an up-graded equation would take much longer on the computer so pricing this design technique beyond the commercial user.

The design technique presented could be applied to all sizes of rotary valve with all forms of pad geometry. The practical results taken from the two bobbins demonstrated that the major assumptions made to simplify the analytical solution were justified. One assumption was that the bobbin ran centrally in the cylinder under optimum running conditions. Since the practical results for rotor speeds above 200 revolutions per minute support the calculated values, this design technique can be considered a suitable method upon which to base future developments. The predicted values for the torque requirements will serve as an indication as to the size of motor required to turn the valve, provided that sufficient details concerning the input parameters for the computer programme are known.

As a result of this research, new techniques for producing the linear motions required on a knitting and a sewing machine have been evolved. These techniques, while still needing further investigation, have shown that they could form the basis for a future generation of machines, and if adopted, could provide commercial benefits to the textile industry. As a secondary function, it is hoped that engineers in other industries not conversant with miniature hydraulic devices will be inspired into assessing the possibility of adopting hydraulic devices to replace conventional mechanical mechanisms.

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2.	1966	A. CAMERON	"Principles of Lubric-ation	Longmans
3.	1970	F.M. STANSFIELD	"Hydrostatic Bearings"	Machinery
4.	1960	S.E. FORSYTHE & W.R. WASOW	"Finite-Difference Methods for Partial Differential Equations"	Wiley
5.	1965	G. SMITH	"Numerical Solutions to Partial Differential Equations"	Oxford
6.	1969	E. MAYER	"Mechanical Seals"	Iliffe
7.	1960	J.F. BLACKBURN G. REETHOF & J.L. SHEARER	"Fluid Power Control"	Wiley
8.	1966	P.J. HOWARD	"Theory of Machines"	Macdonald
9.	1965	D. McCRACKEN	"A Guide to Fortran IV Programming"	Wiley
10.	1959	A. MORLEY	"Strength of Materials"	Longmans
11.	1969	J.P. O'DONOGHUE & W.B. ROWE	"Hydrostatic Bearing Design"	Machine Design
12.	1963	H.C. RIPPEL	Ten Part series on "Design of Hydrostatic Bearings"	Machine Design
13.	1957	A.A. RAIMONDI & J. BOYD	"An Analysis of Orifice and Capillary Compensated Hydrostatic Bearings"	Lubrication Engineering
14.	1965	D. DAWSON	Notes from a Bearing Design Course	Leeds University
15.	1966	J. KNAPTON & D. MUNDEN	"A Study of the Mechanism of Loop Formation on Weft-Knitting Machinery"	Textile Research Journal
16.	1966	JAMES & J. KNAPTON	"The Dynamics of Weft Knitting and the Measure-ment of Wool Yarn Knitting Performance"	Knitted Outerwear Times

17.	1966	J. KNAPTON	"The Dynamics of Weft-Knitting -A Mathematical Analysis"	Textile Research Journal
18.	1970	J.D. GARSIDE	"The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine"	Loughborough University of Technology
19.	1972	J.D. GARSIDE & T.P. PRIESTLEY	Provisional Patent Application No. 36572/71 Hydraulic Actuators	

APPENDIX I

Computer programme for calculating the out of balance forces generated by the pressure and exhaust grooves in a rotary valve.

```
      MASTER PRESSURE OF SLOTS
      DIMENSION FORCE(60),THETA(60),SVFO(60),SHFO(60),TOR(60,60),STOR(60
     1)
      COMMON/PP/P(60,60),PBMA,PBMI,IS(100)
      READ(1,20)DIA,BRED,VISCO,HO
   20 FORMAT(4F0.0)
      READ(1,21)EPS,MAXIT
   21 FORMAT(F10.7,I4)
C         DIAMETER OF VALUE IN
C         BREADTH OF PORT SECTION IN
C         VISCOSITY OF OIL CENTIPOISES
C         SPEED OF ROTATION  REVOLUTIONS PER MIN
C         PMAX IS PRESSURE AT HIGH PRESSURE SLOTS LBF/IN2
C         PMIN IS PRESSURE AT EXHAUST SLOTS LBF/IN2
C         HO IS THE RADIAL CLEARANCE OF THE BEARING IN
      PI = 3.14159
      NSLOT = 34
      READ(1,22) (IS(I),I = 1,NSLOT)
   22 FORMAT (4I0)
      WRITE(2,23)DIA,BRED,VISCO,HO,EPS,MAXIT
   23 FORMAT(1H1,21H DIAMETER OF VALVE = ,F10.5,5H INS,6X,13H BREADTH
     1 = ,F10.5,5H INS,//20H VISCOSITY OF OIL = ,F10.5,15H CENTIPOISE
     2  ,6X,20H RADIAL CLEARANCE = ,F10.7,5H INS,//
     311H EPSILON = ,F10.7,6X,32H MAXIMUM NUMBER OF ITERATIONS = ,I4,//)
      WRITE(2,24) (IS(I),I = 1,NSLOT)
   24 FORMAT (78H GRID POINTS ON THE SURFACE OF THE VALVE FOR PRESSURE
     1 EXHAUST AND FEED SLOTS,//, (4(6X,I3)/)//)
   50 READ(1,25) SPEED,PMAX,PMIN
   25 FORMAT (3F0.0)
      WRITE(2,28) SPEED,PMAX,PMIN
   28 FORMAT (21H SPEED OF ROTATION = ,F10.5,14H REVS PER MIN ,//
     126HPRESSURE AT FEED SLOTS = ,F10.5,12H LBF PER IN2,6X,//
     229H PRESSURE AT EXHAUST SLOTS = ,F10.5,12H LBF PER IN2,//)
      RLEN = PI* DIA
      VIS  = VISCO* 1.45* 0.0000001
      VEL  = SPEED * 2.0*PI*(DIA/2.0)/ 60.0
      PBMA = PMAX* HO**2 /(VIS*VEL*RLEN)
      PBMI = PMIN* HO**2 /(VIS*VEL*RLEN)
C         SET THE GRID SIZE
      NALFA = 36
      NBETA = 32
C         SET ALL THE PRESSURES ON THE GRID EQUAL TO ZERO
      DO 1 I = 1,NBETA+1
      DO 1 J = 1,NALFA+3
    1 P(I,J) = PBMI
      ITN     = 0.0
      A = 1.6
   10 D = 0.0
      ITN = ITN+1
      ALFA = 1.0/NALFA
      BETA = 1.0/NBETA
```

```
      DO 2 I = 2,NBETA
      DO 2 J = 2,NALFA +2
      CALL PRESSURE POINTS (I,J,K)
      IF (K .EQ. 1) GO TO 2
      P(I,1) = P(I,NALFA+1)
      P(I,NALFA+3) = (I,3)
      PPLA = P(I,J+1)
      PMNA = P(I,J-1)
      PPLB = P(I-1,J)
      PMNB = P(I+1,J)
      DENOM = 2.0*(1.0+(RLEN*ALFA/(BRED* BETA))**2)
      RNUM = PPLA+PMNA +(RLEN*ALFA/(BRED*BETA))**2*(PPLB+PMNB)
      PTEMP = RNUM/DENOM
      PNEW = (1.0-A)* P(I,J) + A* PTEMP
      IF (PNEW .EQ. 0.0) GO TO 600
      RESID = ABS(1.-P(I,J)/PNEW)
      IF (RESID .GT. D) D = RESID
600    CONTINUE
      P(I,J) = PNEW
    2 CONTINUE
      WRITE (2,260) ITN ,D
260   FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,/)
      IF (ITN .GT. MAXIT) GO TO 3
      IF (D .GE. EPS) GO TO 10
    3 DO 18 I = 1,NBETA+1
      DO 18 J = 1,NALFA+3
      P(I,J) = P(I,J)* VIS * VEL* RLEN /(HO**2)
   18 CONTINUE
      WRITE(2,26) ITN, D, ((P(I,J) ,J = 2,NALFA+1), I = 1,NBETA+1)
   26 FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,
     1///22H PRESSURE DISTRIBUTION,//,( 9 (1X,F 8.3)/)///)
          TO CALCULATE THE HORIZONTAL AND VERTICAL FORCES
      DO 4 I = 2,NBETA+1
      SVFO(I) = 0.0
      SHFO(I) = 0.0
      DO 5 J  = 2,NALFA+1
      DTFO    = P(I,J)*ALFA*RLEN*BETA*BRED
      VERFO   = DTFO *COS((J-2)*2.0*PI*ALFA)
      HORFO   = DTFO *SIN((J-2)*2.0*PI*ALFA)
      SVFO(I) = SVFO(I)+VERFO
      SHFO(I) = SHFO(I)+HORFO
    5 CONTINUE
      FORCE(I) = SQRT((SVFO(I)**2)+(SHFO(I)**2))
      IF (FORCE (I) . EQ. 0.0 ) GO TO 4
      THETA(I) = (180.0/PI)*ATAN(SHFO(I)/SVFO(I))
    4 CONTINUE
          TO CALCULATE THE TOTAL LOAD
      TOSVFO = 0.0
      TOSHFO = 0.0
      DO 6 I = 2,NBETA+1
      TOSVFO = TOSVFO +SVFO(I)
      TOSHFO = TOSHFO +SHFO(I)
    6 CONTINUE
      TOFOR = SQRT(( TOSVFO**2)+(TOSHFO**2))
      TOTHET = (180.0 /PI)* ATAN (TOSHFO /TOSVFO)
      WRITE (2,31)
```

```
   31 FORMAT(1H1,4X,6H INDEX,6X,16H TOTAL FORCE LBF,6X,
     121H HORIZONTAL FORCE LBF,6X,19H VERTICAL FORCE LBF,6X,
     214H THETA DEGREES,//)
      WRITE (2,32)(I,FORCE(I),SHFO(I),SVFO(I),THETA(I), I = 2,NBETA+1)
   32 FORMAT((6X,I4,9X,F10.5,13X,F10.5,15X,F10.5,12X,F10.5,/)//)
      WRITE(2,33) TOFOR,TOTHET,TOSHFO,TOSVFO
   33 FORMAT(7X,17H SUM OF FORCES = ,F10.5,5H LBF,8X,
     130H DIRECTION OF FORCE (THETA) = ,F10.5,8H DEGREES,//,7X,
     227H SUM OF HORIZONTAL FORCE = ,F10.5,5H LBF,8X,25H SUM OF VERTICAL
     3  FORCE = ,F10.5,5H LBF,//)
C          TO FIND THE END REACTIONS
      TVMOA = 0.0
      THMOA = 0.0
      TVMOB = 0.0
      THMOB = 0.0
      DO 7 I = 2,NBETA+1
      VMOA = SVFO(I) *((I-1)* BETA*BRED )
      HMOA = SHFO(I) *((I-1)* BETA*BRED )
      TVMOA = TVMOA + VMOA
      THMOA = THMOA + HMOA
      VMOB  = SVFO(I)*(NBETA+1-I)*BETA*BRED
      HMOB  = SHFO(I)*(NBETA+1-I)*BETA*BRED
      TVMOB = TVMOB + VMOB
      THMOB = THMOB + HMOB
    7 CONTINUE
      VREA  = TVMOB /BRED
      HREA  = THMOB /BRED
      VREB  = TVMOA /BRED
      HREB  = THMOA /BRED
      TOREA = SQRT((VREA**2)+(HREA**2))
      REATH = (180.0/PI)*ATAN(HREA/VREA)
      TOREB = SQRT((VREB**2)+(HREB**2))
      REBTH = (180.0/PI)*ATAN(HREB/VREB)
      WRITE(2,34) TOREA ,VREA,HREA,REATH,TOREB,VREB,HREB,REBTH
   34 FORMAT(6X,22H TOTAL REACTION AT A = ,F10.5,5H LBF,6X,
     125H VERTICAL REACTION AT A = ,F10.5,5H LBF,6X,//,6X,
     228H HORIZONTAL REACTION AT A = ,F10.5,5H LBF,6X,21H ANGLE OF RE-
     3ACTION = ,F10.5,9H DEGREES,//,6X,22H TOTAL REACTION AT B = ,F10.5,
     45H LBF,6X,
     525H VERTICAL REACTION AT B = ,F10.5,5H LBF,6X,//,6X,
     628H HORIZONTAL REACTION AT B = ,F10.5,5H LBF,6X,21H ANGLE OF RE-
     7ACTION = ,F10.5,9H DEGREES,// )
C          TO FIND THE TORQUE REQUIRED
      DO 8 I = 1,NBETA+1
      DO 8 J = 2,NALFA+2
      PD = (P(I,J+1)-P(I,J-1))
      TOR(I,J) = ABS(PD)*HO/(4.0*ALFA*RLEN)+(VIS*VEL/HO)
    8 CONTINUE
      DO 11 I = 1,NBETA+1
      ODD = 0.0
      EVEN = 0.0
      DO 12 J = 3,NALFA+1,2
   12 EVEN = EVEN +TOR(I,J)
      DO 13 J = 4,NALFA,2
   13 ODD = ODD +TOR(I,J)
      STOR(I) = (ALFA /3.0)*(TOR(I,2)+4.0*EVEN+2.0*ODD +TOR(I,NALFA+2))*
```

```
     1RLEN
  11 CONTINUE
     ODD = 0.0
     EVEN = 0.0
     DO 14 I = 2,NBETA,2
  14 EVEN = EVEN + STOR(I)
     DO 15 I = 3,NBETA-1,2
  15 ODD = ODD + STOR(I)
     FOTOR = (BETA/3.0)*(STOR(1)+4.0*EVEN+2.0*ODD +STOR(NBETA+1))*BRED
     TOTOR = FOTOR *DIA /2.0
     WRITE (2,35) TOTOR
  35 FORMAT(6X,16H TOTAL TORQUE = ,F10.5,8H LBF IN,//)
     GO TO 50
     STOP
     END

     SUBROUTINE PRESSURE POINTS(I,J,K)
     COMMON /PP/ P(60,60),PBMA,PBMI, IS(100)
C        POSITION OF HIGH PRESSURE POINTS
     IF(I.GE.IS(1). AND.I.LE.IS(2).AND.J.GE.IS(3).AND.J.LE.IS(4))GO TO 100
     IF(I.GE.IS(5).AND.I.LE.IS(6).AND.J.GE.IS(7).AND.J.LE.IS(8)) GO TO 100
     IF(I.GE.IS(9).AND.I.LE.IS(10).AND.J.GE.IS(11).AND.J.LE.IS(12))
    1 GO TO 100
     IF(I.GE.IS(13).AND.I.LE.IS(14).AND.J.GE.IS(15).AND.J.LE.IS(16))
    1 GO TO 100
C          POSITION OF LOW PRESSURE POINTS
     IF(I.GE.IS(17).AND.I.LE.IS(18).AND.J.GE.IS(19).AND.J.LE.IS(20))
    1 GO TO 200
     IF(I.GE.IS(21).AND.I.LE.IS(22).AND.J.GE.IS(23).AND.J.LE.IS(24))
    1 GO TO 200
     IF(I.GE.IS(25).AND.I.LE.IS(26).AND.J.GE.IS(27).AND.J.LE.IS(28))
    1 GO TO 200
     IF(I.GE.IS(29).AND.I.LE.IS(30).AND.J.GE.IS(31).AND.J.LE.IS(32))
    1 GO TO 200
C          POSITION OF SUPPLY PORT
     IF(I.GE.IS(33).AND.I.LE.IS(34)) GO TO 100
C          CHECK TO MAKE PRESSURE +VE
     IF(P(I,J).LT.0.0) P(I,J) = 0.0
     K = 2
     GO TO 300
 100 P(I,J) = PBMA
     K = 1
     GO TO 300
 200 P(I,J) = PBMI
     K = 1
 300 RETURN
     END
```

APPENDIX II

Computer programme for designing the compensating pad dimensions once the out-of-balance forces in a rotary valve have been determined.

```
      MASTER COMPENSATING PADS
      DIMENSION FORCE(60),P(60,60),SVFO(60),SHFO(60),TOR(60,60),STOR(60)
     1,THETA(60)
      READ(1,21) DIA,VISCO,HO,EHO
   21 FORMAT (4F0.0)
      READ(1,20) EPS,MAXIT
   20 FORMAT (F10.7,I4)
C          DIAMETER OF VALVE IN
C          VISCOSITY OF OIL CENTIPOISES
C          SPEED OF ROTATION REVOLUTIONS PER MIN
C          PMAX IS PRESSURE AT HIGH PRESSURE SLOTS LBF/IN2
C          PMIN IS PRESSURE AT EXHAUST SLOTS LBF/IN2
C          HO IS THE RADIAL CLEARANCE OF THE BEARING IN
C          EHO IS THE END CLEARANCE OF THE BEARING
      PI = 3.14159
      WRITE (2,23) DIA,VISCO,HO,EPS,MAXIT
   23 FORMAT (1H1,21H DIAMETER OF VALVE = ,F10.5,5H INS,
     1//20H VISCOSITY OF OIL = ,F10.5,15H CENTIPOISE   ,6X,
     220H RADIAL CLEARANCE = ,F10.7,5H INS,//11H EPSILON = ,F10,7,6X,
     332H MAXIMUM NUMBER OF ITERATIONS = ,I4,//)
      M6 = 1
   50 READ(1,25) SPEED,PMAX,PMIN,RLOAD
   25 FORMAT (4F0.0)
      WRITE (2,28) SPEED,PMAX,PMIN,RLOAD
   28 FORMAT (21H SPEED OF ROTATION = ,F10.5,14H REVS PER MIN ,//
     126H PRESSURE AT FEED SLOTS = ,F10.5,12H LBF PER IN2,6X,//
     229H PRESSURE AT EXHAUST SLOTS = ,F10.5,12H LBF PER IN2,//
     323H OUT OF BALANCE LOAD = ,F10.5,5H LBF,//)
      RLEN = PI*DIA
      PLEN = PI*DIA/2.0
      VIS  = VISCO*1.45* 0.0000001
      VEL  = SPEED *2.0*PI*(DIA/2.0)/60.0
      PBMA = PMAX* HO**2 /(VIS*VEL*PLEN)
      PBMI = PMIN* HO**2 /(VIS*VEL*PLEN)
C            SET THE GRID SIZE       P
      NALFA = 20
      NBETA = 20
      MK = 0
      A  = 1.8
C           SET THE PAD SIZE AND VALUE OF BREADTH
      ALFA = 1.0 /NALFA
      BETA = 1.0 /NBETA
   83 IPAD1 = (NBETA/4)+1
      IPAD2 = ((3*NBETA)/4)+1
      JPAD1 = (NALFA /4)+1
      JPAD2 = ((3*NALFA)/4)+1
      M1 = 0
      M2 = 0
      KK = 0
      IF(MK.EQ.1)GO TO 101
      PPLJ = (JPAD2 -JPAD1)*ALFA*PLEN
```

```
C       TO FIND BREADTH OF VALVE ASSUMING LINEAR PRESSURE GRADIENT
    BRED = (RLOAD/PMAX)/( 0.75*1.50*PPLJ *0.667)
C       SET ALL THE PRESSURES ON THE GRID TO EXHAUST PRESSURE
101 DO 1 I = 1,NBETA+1
    DO 1 J = 1,NALFA+1
  1 P(I,J) = PBMI
    ITN = 0
 10 D = 0.0
    ITN = ITN +1
    DO 2 I = 2,NBETA
    DO 2 J = 2,NALFA
    IF(I.GE.IPAD1.AND .I.LE.IPAD2.AND.J.GE.JPAD1.AND.J.LE.JPAD2)GO TO 60
    K = 2
    GO TO 61
 60 P(I,J) = PBMA
    K = 1
 61 IF(K.EQ.1) GO TO 2
    PPLA = P(I,J+1)
    PMNA = P(I,J-1)
    PPLB = P(I-1,J)
    PMNB = P(I+1,J)
    DENOM = 2.0 *(1.0 +(PLEN*ALFA/(BRED* BETA))**2)
    RNUM = PPLA +PMNA +(PLEN* ALFA /(BRED *BETA))**2*(PPLB +PMNB)
    PTEMP = RNUM /DENOM
    PNEW = (1.0 -A)* P(I,J) +A*PTEMP
    IF (PNEW.EQ.0.0) GO TO 62
    RESID = ABS(1.0 -P(I,J)/PNEW)
 62 CONTINUE
    IF (RESID .GT.D) D = RESID
    P(I,J) = PNEW
  2 CONTINUE
    WRITE (2,63) ITN,D
 63 FORMAT(24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,/)
    IF (ITN.GT.MAXIT) GO TO 3
    IF ( D.GE.EPS) GO TO 10
  3 DO 64 I = 1,NBETA+1
    DO 64 J = 1,NALFA+1
 64 P(I,J)  = P(I,J)*VIS*VEL* PLEN /(HO**2)
    WRITE (2,26) ITN,D, ((P(I,J), J = 2,NALFA+1, I = 1,NBETA+1)
 26 FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,
   1///22H PRESSURE DISTRIBUTION,//,(10 (1X,F8.3)/)///)
C       TO CALCULATE THE HORIZONTAL AND VERTICAL FORCES
    DO 4 I = 2,NBETA+1
    SVFO(I) = 0.0
    SHFO(I) = 0.0
    DO 5 J  = 2,NALFA+1
    DTFO = P(I,J)*ALFA*PLEN*BETA *BRED
    VERFO = DTFO*COS((J-1)* PI *ALFA)
    HORFO = DTFO*SIN((J-1)* PI *ALFA)
    SVFO(I) = SVFO(I) + VERFO
    SHFO(I) = SHFO(I) + HORFO
  5 CONTINUE
    FORCE(I) = SQRT((SVFO(I)**2)+(SHFO(I)**2))
    IF (FORCE (I).EQ. 0.0) GO TO 4
    THETA(I) = (180.0/PI)*ATAN(SHFO(I)/SVFO(I))
  4 CONTINUE
```

```
C         TO CALCULATE THE TOTAL LOAD
      TOSVFO = 0.0
      TOSHFO = 0.0
      DO 6 I = 2,NBETA+1
      TOSVFO = TOSVFO+SVFO(I)
      TOSHFO = TOSHFO+SHFO(I)
    6 CONTINUE
      TOTFO = SQRT((TOSVFO**2)+(TOSHFO**2))
      TOTHET = (180.0/PI)*ATAN(TOSHFO /TOSVFO)
      WRITE (2,33) TOTFO,TOTHET,TOSHFO,TOSVFO
   33 FORMAT(7X,17H SUM OF FORCES = ,F10.5,5H LBF,8X,
     130H DIRECTION OF FORCE. (THETA) = ,F10.5,8H DEGREES,//.7X,
     227H SUM OF HORIZONTAL FORCE = ,F10.5,5H LBF,8X,
     325H SUM OF VERTICAL FORCE = ,F10.5,5H LBF,//)
      WRITE(2,41) IPAD1, I PAD2, JPAD1, JPAD2.
   41 FORMAT (7X,33HGRID POINTS ACROSS THE BEARING = ,I3,4X,I3,//,7X,
     132HGRID POINTS ROUND THE BEARING = ,I3,4X,I3,//)
C         TO ADJUST THE PAD SIZE
      IF((RLOAD -TOTFO).LT.0.0) M1 = 1
      IF((RLOAD -TOTFO).GE.0.0) M2 = 1
      IF((M1-M2).EQ.0) GO TO 71
      RELOAD = RLOAD - TOTFO
      WRITE(2,87) RELOAD
   87 FORMAT(6X,23H OUT OF BALANCE LOAD = ,F10.6,5H LBF,/)
      IF(KK.EQ.1) GO TO 72
      IF(RELOAD) 73,74,75
   73 IPAD1 =  IPAD 1 + 1
      KK = 1
      MM = 1
      GO TO 101
   75 IPAD1 = IPAD1 -1
      KK = 1
      MM = 2
      GO TO 101
   72 IF(RELOAD) 76,74,77
   76 IPAD2 = IPAD2 -1
      KK = 0
      MM = 1
      GO TO 101
   77 IPAD2 = IPAD2 +1
      KK = 0
      MM = 2
      GO TO 101
   71 REMLOAD = RLOAD - TOTFO
      IF(MM.EQ.1) GO TO 78
      BRPAD = ((IPAD2-IPAD1)+(REMLOAD/(RELOAD-REMLOAD)))*BETA*BRED
      GO TO 74
   78 BRPAD = ((IPAD2 -IPAD1)+(REMLOAD/(REMLOAD -RELOAD)))*BETA*BRED
   74 PERPAD = (2.0* BRPAD)+((JPAD2-JPAD1)*PLEN*ALFA*2.0)
      PERBER = 2.0*(PLEN+BRED)
      RATIO = PERPAD/PERBER
      IF(RATIO.LT.0.4.AND.RATIO.GT.0.6) GO TO 81
      GO TO 82
   81 BRED = BRPAD *2.0
      MK = 1
      GO TO 83
```

```
   82 ANPPLJ = (PPLJ/(DIA/2.0))*180.0/PI
      WRITE(2,85) RELOAD,REMLOAD,PLEN,BRED,PPLJ,ANPPLJ,BRPAD
   85 FORMAT(6X,63H LIMIT OF OUT OF BALANCE LOAD AT NEXT TO LAST GRID
    1 POINT = ,F10.5,5H LBF,/6X,55H LIMIT OF OUT OF BALANCE LOAD AT THE
    2 LAST GRID POINT = ,F10.5,5H LBF,/6X
    3 21H LENGTH OF BEARING = ,F10.5,5H INS,6X,22H BREADTH OF BEARING =
    4 ,F10.5,5H INS,/6X,
    5 31H LENGTH OF HIGH PRESSURE PAD = ,F10.5,5H INS,6X,
    6 30H ANGLE SUBTENDED BY THE PAD = ,F10.6,8H DEGREES,/6X,
    7 36H BREADTH OF THE HIGH PRESSURE PAD = ,F10.5,5H INS,///)
C        TO FIND THE TORQUE REQUIRED OVER THE PAD SEGMENT
      DO 8 I = 1,NBETA+1
      DO 8 J = 1,NALFA+1
      IF ( J.EQ. 1) GO TO 103
      IF ( J.EQ. NALFA+1 ) GO TO 104
      PD = (P(I,J+1)-P(I,J-1))
      GO TO 106
  103 PD = (P(I,J+1)-PMIN)
      GO TC 106
  104 PD = ( PMIN -P(I,J-1))
  106 TOR(I,J) = ABS(PD)*HO/(4.0*ALFA*PLEN)+(VIS*VEL/HO)
    8 CONTINUE
      WRITE (2,120) ((TOR(I,J), J = 2,21), I = 1,21)
  120 FORMAT (6X,5H TOR ,//,(10 (1X,F8.3)/) ///)
      DO 11 I = 1,NBETA+1
      ODD = 0.0
      EVEN = 0.0
      DO 12 J = 2,NALFA-2
   12 EVEN = EVEN + TOR(I,J)
      DO 13 J = 3,NALFA-1,2
   13 ODD = ODD +TOR(I,J)
      STOR(I) = (ALFA /3.0)*(TOR(I,1)+4.0*EVEN+2.0*ODD+TOR(I,NALFA+1))*
    1 PLEN
   11 CONTINUE
      ODD = 0.0
      EVEN = 0.0
      DO 14 I =2,NBETA,2
   14 EVEN = EVEN +STOR(I)
      DO 15 I = 3,NBETA-1,2
   15 ODD = ODD +STOR(I)
      FOTOR = (BETA/3.0)*(STOR(1)+4.0*EVEN+2.0*ODD +STOR(NBETA+1))*BRED
C        TO FIND THE TORQUE DUE TO THE PLAIN SIDE OF BEARING
      FOPLB = (VIS*VEL/HO)  * BRED*(RLEN-PLEN)
      TOTOR = (FOTOR + FOPLB)*DIA/ 2.0
      TOPAD = FOTOR *DIA/ 2.0
      TOPLB = FOPLB *DIA/ 2.0
      WRITE (2,17) TOTOR ,TOPAD,TOPLB
   17 FORMAT(6X,16H TOTAL TORQUE = ,F10.5,8H LBF IN,/6X,
      130H TORQUE DUE TO PRESSURE PAD = ,F13.9,8H LBF IN,/6X,
      231 TORQUE DUE TO PLAIN BEARING = ,F13.9,8H LBF IN,//)
      IF (M6 .EQ. 2) GO TO 102
      M6 = 2
      GO TO 50
C        TO CALCULATE THE FRICTIONAL TORQUE DUE TO THE END COVERS
  102 RAD1 = DIA/2.0
      RAD2 = RAD1 *0.333
```

```
      DIA2 = 2.0*RAD2
      ANGVEL = (SPEED /60.0)* 2.0*PI
      TOREND = PI*ANGVEL*VIS* (RAD1**4 -RAD2**4)/( 2.0* EHO)
      TOTEND = 2.0* TOREND
      WRITE (2,18) TOREND ,TOTEND, DIA,DIA2
   18 FORMAT(6X,25H TORQUE DUE TO END PAD = ,F10.5,8H LBF IN,//,
     16X,37H TOTAL TORQUE DUE TO BOTH END PADS = ,F10.5,8H LBF IN,//,
     26X,27H OUTSIDE DIAMETER OF PAD = ,F10.5,5H INS,6X,
     326H INSIDE DIAMETER OF PAD = ,F10.5,5H INS,//)
      HP = ANGVEL* 1.0 /(550.0 *12.0)
      WRITE (2,42)  HP
   42 FORMAT(6X,44H THE HORSE POWER PER UNIT TORQUE (LBF IN) = ,F10.5,/)
      STOP
      END
```

APPENDIX III

Computer programme to estimate the torque required to rotate the centre section of the rotary valve at various supply pressures and rotor speeds.

```
      MASTER PRESSURE OF SLOT 2
      DIMENSION FORCE(60),THETA(60),SVFO(60),SHFO(60),TOR(60,60),STOR(60
     1),R(60,60)
      COMMON/PP/P(60,60),PBMA,PBMI,IS(100)
      READ(1,20)DIA,BRED,VISCO,HO
   20 FORMAT(4F0.0)
      READ(1,21)EPS,MAXIT
   21 FORMAT(F10.7,I4)
C         DIAMETER OF VALVE IN
C         BREADTH OF PORT SECTION IN
C         VISCOSITY OF OIL CENTIPOISES
C         SPEED OF ROTATION REVOLUTIONS PER MIN
C         PMAX IS PRESSURE AT HIGH PRESSURE SLOTS LBF /IN2
C         PMIN IS PRESSURE AT EXHAUST SLOTS LBF /IN2
C         HO IS THE RADIAL CLEARANCE OF THE BEARING IN
      PI = 3.14159
      NSLOT = 34
      READ(1,22) (IS(I),I = 1,NSLOT)
   22 FORMAT (4I0)
      WRITE(2,23)DIA,BRED,VISCO,HO,EPS,MAXIT
   23 FORMAT(1H1,21H DIAMETER OF VALVE = ,F10.5,5H INS,6X,13H BREADTH
     1 = ,F10.5,5H INS,//20H VISCOSITY OF OIL = ,F10.5,15H CENTIPOISE
     2 ,6X,20H RADIAL CLEARANCE = ,F10.7,5H INS,//
     311H EPSILON = ,F10.7,6X,32H MAXIMUM NUMBER OF ITERATIONS = ,I4//)
      WRITE(2,24) (IS(I),I = 1,NSLOT)
   24 FORMAT (78H GRID POINTS ON THE SURFACE OF THE VALVE FOR PRESSURE
     1 EXHAUST AND FEED SLOTS,//. (4(6X,I3)/)//)
      SPEED = 1.0
      PMAX  = 1.0
      PMIN  = PMAX *0.1
      RLEN  = PI* DIA
      VIS   = VISCO* 1.45* 0.0000001
      VEL   = SPEED * 2.0*PI*(DIA/2.0)/ 60.0
      PBMA  = PMAX* HO**2 /(VIS*VEL*RLEN)
      PBMI  = PMIN* HO**2/(VIS*VEL*RLEN)
C          SET THE GRID SIZE
      NALFA = 36
      NBETA = 32
C          SET ALL THE PRESSURES ON THE GRID EQUAL TO ZERO
      DO 1 I = 1,NBETA+1
      DO 1 J = 1,NALFA+3
    1  P(I,J) = PBMI
      ITN = 0.0
      A = 1.6
   10 D = 0.0
      ITN = ITN+1
      ALFA = 1.0 /NALFA
      BETA = 1.0 /NBETA
      DO 2 I = 2,NBETA
      DO 2 J = 2,NALFA +2
      CALL PRESSURE POINTS (I,J,K)
```

```
      IF (K .EQ. 1) GO TO 2
      P(I,1) = P(I,NALFA+1)
      P(I,NALFA-3) = P (I,3)
      PPLA  = P(I,J+1)
      PMNA  = P(I,J-1)
      PPLB  = P(I-1,J)
      PMNB  = P(I+1,J)
      DENOM = 2.0*(1.0+(RLEN*ALFA/BRED* BETA))**2)
      RNUM  = PPLA+PMNA +(RLEN*ALFA /(BRED*BETA))**2*(PPLB+PMNB)
      PTEMP = RNUM /DENOM
      PNEW  = (1.0-A)* P(I,J) + A* PTEMP
      IF (PNEW .EQ. 0.0) GO TO 600
      RESID = ABS(1.-P(I,J)/PNEW)
      IF (RESID .GT. D) D = RESID
600    CONTINUE
      P(I,J) = PNEW
    2 CONTINUE
      WRITE (2,260) ITN ,D
260   FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,/)
      IF (ITN .GT. MAXIT) GO TO 3
      IF (D .GE. EPS) GO TO 10
    3 DO 18 I = 1,NBETA+1
      DO 18 J = 1,NALFA+3
      P(I,J)  = P(I,J)* VIS * VEL* RLEN /(HO**2)
   18 CONTINUE
      WRITE (2,26) ITN, D, ((P(I,J) ,J = 2,NALFA+1), I = 1,NBETA+1)
   26 FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,
     1///47H PRESSURE DISTRIBUTION FOR UNIT SUPPLY PRESSURE,//,
     2( 9 (3X,F8.6)/)///)
C        TO CALCULATE THE HORIZONTAL AND VERTICAL FORCES
      DO 90 M = 50,700,50
      DO 91 I = 1,NBETA+1
      DO 91 J = 1,NALFA+3
      R(I,J)  = M*P(I,J)
   91 CONTINUE
      PPMAX   = M*PMAX
      PPMIN   = PPMAX*0.1
      DO 4 I  = 2,NBETA+1
      SVFO(I) = 0.0
      SHFO(I) = 0.0
      DO 5 J  = 2,NALFA+1
      DTFO    = R(I,J)*ALFA*RLEN*BETA*BRED
      VERFO   = DTFO *COS((J-2)*2.0*PI*ALFA)
      HORFO   = DTFO *SIN((J-2)*2.0*PI*ALFA)
      SVFO(I) = SVFO(I)+VERFO
      SHFO(I) = SHFO(I)+HORFO
    5 CONTINUE
      FORCE(I) = SQRT((SVFO(I)**2)+(SHFO(I)**2))
      IF (FORCE (I) . EQ. 0.0 ) GO TO 4
      THETA(I) = (180.0/PI)*ATAN(SHFO(I)/SVFO(I))
    4 CONTINUE
C        TO CALCULATE THE TOTAL LOAD
      TOSVFO = 0.0
      TOSHFO = 0.0
      DO 6 I  = 2,NBETA+1
      TOSVFO = TOSVFO +SVFO(I)
```

```
      TOSHFO = TOSHFO +SHFO(I)
    6 CONTINUE
      TOFOR = SQRT(( TOSVFO**2)+(TOSHFO**2))
      TOTHET = (180.0 /PI)* ATAN (TOSHFO /TOSVFO)
      WRITE(2,29) PPMAX , PPMIN
   29 FORMAT (6X,26H PRESSURE AT FEED SLOTS = ,F10.5,12H LBF PER IN2,//,
     16X,29H PRESSURE AT EXHAUST SLOTS = ,F10.5,12H LBF PER IN2,//)
      WRITE(2,33) TOFOR, TOTHET, TOSHFO,TOSVFO
   33 FORMAT(7X,17H SUM OF FORCES = ,F10.5,5H LBF,8X,
     130H DIRECTION OF FORCE (THETA) = ,F10.5,8H DEGREES,//,7X,
     227H SUM OF HORIZONTAL FORCE = ,F10.5,5H LBF,8X,25H SUM OF VERTICAL
     3FORCE = ,F10.5,5H LBF,//)
C        TO FIND THE END REACTIONS
      TVMOA = 0.0
      THMOA = 0.0
      TVMOR = 0.0
      THMOB = 0.0
      DO 7 I = 2,NBETA+1
      VMOA = SVFO(I) *((I-1)* BETA*BRED )
      HMOA = SHFO(I) *((I-1)* BETA*BRED  )
      TVMOA = TVMOA + VMOA
      THMOA = THMOA + HMOA
      VMOB = SVFO(I)*(NBETA+1-I)*BETA*BRED
      HMOB = SHFO(I)*(NBETA+1-I)*BETA*BRED
      TVMOB = TVMOB + VMOB
      THMOB = THMOB + HMOB
    7 CONTINUE
      VREA = TVMOB /BRED
      HREA = THMOB /BRED
      VREB = TVMOA /BRED
      HREB = THMOA /BRED
      TOREA = SQRT((VREA**2)+(HREA**2))
      REATH = (180.0/PI)*ATAN(HREA/VREA)
      TOREB = SQRT((VREB**2)+(HREB**2))
      REBTH = (180.0/PI)*ATAN(HREB/VREB)
      WRITE(2,34) TOREA ,VREA,HREA,REATH,TOREB,VREB,HREB,REBTH
   34 FORMAT(6X,22H TOTAL REACTION AT A = ,F10.5,5H LBF,6X,
     125H VERTICAL REACTION AT A = ,F10.5,5H LBF,6X,//,6X,
     228H HORIZONTAL REACTION AT A = ,F10.5,5H LBF,6X,21H ANGLE OF RE
     3ACTION = ,F10.5,9H DEGREES,//,6X,22H TOTAL REACTION AT B = ,F10.5,
     45H LBF,6X,
     525H VERTICAL REACTION AT B = ,F10.5,5H LBF,6X,//,6X,
     628H HORIZONTAL REACTION AT B = ,F10.5,5H LBF,6X,21H ANGLE OF RE
     7ACTION = ,F10.5,9H DEGREES,//)
C           TO FIND THE TORQUE REQUIRED
      DO 90 N = 50,1000,50
      REV = N
      VELO = N*VEL
      DO 8 I = 1,NBETA+1
      DO 8 J = 2,NALFA+2
      PD = (R(I,J-1)-R(I,J-1))
      TOR(I,J) = ABS(PD)*HO/(4.0*ALFA*RLEN)+(VIS*VELO/HO)
    8 CONTINUE
      DO 11 I = 1,NBETA+1
      ODD = 0.0
      EVEN = 0.0
```

```
   DO 12 J = 3,NALFA+1,2
12 EVEN = EVEN +TOR(I,J)
   DO 13 J = 4,NALFA,2
13 ODD = ODD +TOR(I,J)
   STOR(I) = (ALFA /3.0)*(TOR(I,2)+4.0*EVEN+2.0*ODD +TOR(I,NALFA+2))*
  1RLEN
11 CONTINUE
   ODD = 0.0
   EVEN = 0.0
   DO 14 I = 2,NBETA,2
14 EVEN = EVEN + STOR(I)
   DO 15 I = 3,NBETA-1,2
15 ODD = ODD + STOR(I)
   FOTOR = (BETA/3.0)*(STOR(1)+4.0*EVEN+2.0*ODD +STOR(NBETA+1))*BRED
   TOTOR = FOTOR *DIA /2.0
   WRITE(2,28) REV
28 FORMAT (21H SPEED OF ROTATION = ,F10.5,14H REVS PER MIN)
   WRITE (2,35) TOTOR
35 FORMAT(9X,16H TOTAL TORQUE = ,F10.5,8H LBF IN,//)
90 CONTINUE
   STOP END
```

APPENDIX IV

Computer programme to calculate the torque required to rotate the compensating pads and end bearings of a rotary valve, at various supply pressures and rotor speeds.

```
      MASTER COMPENSATING PADS 2
      DIMENSION FORCE(60),P(60,60),SVFO(60),SHFO(60),TOR(60,60),STOR(60)
     1,THETA(60),R(60,60)
      READ(1,21) DIA,VISCO,HO,EHO
   21 FORMAT (4F0.0)
      READ(1,20) EPS,MAXIT
   20 FORMAT (F10.7,I4)
C            DIAMETER OF VALVE IN
C            VISCOSITY OF OIL CENTIPOISES
C            SPEED OF ROTATION REVOLUTIONS PER MIN
C            PMAX IS PRESSURE AT HIGH PRESSURE SLOTS LBF/IN2
C            PMIN IS PRESSURE AT EXHAUST SLOTS LBF/IN2
C            HO IS THE RADIAL CLEARANCE OF THE BEARING IN
C            EHO IS THE END CLEARANCE OF THE BEARING
      PI = 3.14159
      WRITE (2,23) DIA,VISCO,HO,EPS,MAXIT
   23 FORMAT (1H1,21H DIAMETER OF VALVE = ,F10.5,5H INS,
     1//20H VISCOSITY OF OIL = ,F10.5,15H CENTIPOISE    ,6X,
     220H RADIAL CLEARANCE = ,F10.7,5H INS,//11H EPSILON = ,F10.7,6X
     332H MAXIMUM NUMBER OF ITERATIONS = ,I4,//)
      M6 = 1
   50 READ (1,25) IPAD1,IPAD2,JPAD1,JPAD2,BRED
   25 FORMAT (4I0,F0.0)
      WRITE(2,41) IPAD1,IPAD2, JPAD1, JPAD2,BRED
   41 FORMAT (7X,33HGRID POINTS ACROSS THE BEARING = ,I3,4X,I3,//,7X,
     132HGRID POINTS ROUND THE BEARING = ,I3,4X,I3,//,
     27H,22H BREADTH OF BEARING = ,F10.5,5H INS,//)
      PMAX = 1.0
      PMIN = PMAX *0.1
      SPEED = 1.0
      RLEN  = PI*DIA
      PLEN  = PI*DIA/2.0
      VIS   = VISCO*1.45* 0.0000001
      VEL   = SPEED *2.0*PI*(DIA/2.0)/60.0
      PBMA  = PMAX* HO**2 /(VIS*VEL*PLEN)
      PBMI  = PMIN* HO**2 /(VIS*VEL* PLEN)
C            SET THE GRID SIZE
      NALFA = 20
      NBETA = 20
      A = 1.8
C         SET THE PAD SIZE AND VALUE OF BREADTH
      ALFA = 1.0 /NALFA
      BETA = 1.0 /NBETA
C         SET ALL THE PRESSURES ON THE GRID TO EXHAUST PRESSURE
  101 DO 1 I = 1,NBETA+1
      DO 1 J = 1,NALFA+1
    1 P(I,J) = PBMI
      ITN = 0
   10 D = 0.0
      ITN = ITN +1
```

```
      DO 2 I = 2,NBETA
      DO 2 J = 2,NALFA
      IF(I. GE.IPAD1.AND .I.LE.IPAD2.AND.J.GE.JPAD1.AND.J.LE.JPAD2)GO TO 60
      K = 2
      GO TO 61
   60 P(I,J) = PBMA
      K = 1
   61 IF(K. EQ.1) GO TO 2
      PPLA = P(I,J+1)
      PMNA = P(I,J-1)
      PPLB = P(I-1,J)
      PMNB = P(I+1,J)
      DENOM = 2.0 *(1.0 + (PLEN*ALFA/(BRED* BETA))**2)
      RNUM = PPLA +PMNA +(PLEN* ALFA /(BRED *BETA))**2*(PPLB +PMNB)
      PTEMP = RNUM /DENOM
      PNEW = (1.0 -A)* P(I,J) +A*PTEMP
      IF (PNEW.EQ.0.0) GO TO 62
      RESID = ABS(1.0 -P(I,J)/PNEW)
   62 CONTINUE
      IF (RESID .GT.D) D = RESID
      P(I,J) = PNEW
    2 CONTINUE
      WRITE (2,63) ITN,D
   63 FORMAT(24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,/)
      IF (ITN.GT.MAXIT) GO TO 3
      IF ( D.GE.EPS) GO TO 10
    3 DO 64 I = 1,NBETA+1
      DO 64 J = 1,NALFA+1
   64 P(I,J) = P(I,J)*VIS*VEL* PLEN /(HO**2)
      WRITE (2,26) ITN,D, ((P(I,J), J = 2,NALFA+1), I = 1,NBETA+1)
   26 FORMAT (24H NUMBER OF ITERATIONS = ,I4,6X,12H RESIDUAL = ,F12.8,
     1///47H PRESSURE DISTRIBUTION FOR UNIT SUPPLY PRESSURE,//,
     2( 10(3X,F8.6)/)///)
C         TO CALCULATE THE HORIZONTAL AND VERTICAL FORCES
      DO 90 M = 50,700,50
      DO 91 I = 1,NBETA+1
      DO 91 J = 1,NALFA+1
      R(I,J) = M*P(I,J)
   91 CONTINUE
      PPMAX = M*PMAX
      PPMIN = PPMAX*0.1
      DO 4 I = 2,NBETA+1
      SVFO(I) = 0.0
      SHFO(I) = 0.0
      DO 5 J  = 2,NALFA+1
      DTFO = R(I,J)*ALFA*PLEN*BETA*BRED
      VERFO = DTFO*COS((J-1)* PI *ALFA)
      HORFO = DTFO*SIN((J-1)* PI *ALFA)
      SVFO(I) = SVFO(I) + VERFO
      SHFO(I) = SHFO(I) + HORFO
    5 CONTINUE
      FORCE(I) = SQRT((SVFO(I)**2)+(SHFO(I)**2))
      IF(FORCE (I).EQ. 0.0) GO TO 4
      THETA(I) = (180.0/PI)*ATAN(SHFO(I)/SVFO(I))
    4 CONTINUE
C         TO CALCULATE THE TOTAL LOAD
```

```
      TOSVFO = 0.0
      TOSHFO = 0.0
      DO 6 I = 2,NBETA+1
      TOSVFO = TOSVFO+SVFO(I)
      TOSHFO = TOSHFO+SHFO(I)
    6 CONTINUE
      TOTFO  = SQRT((TOSVFO**2)+(TOSHFO**2))
      TOTHET = (180.0/PI)*ATAN(TOSHFO /TOSVFO)
      WRITE(2,29) PPMAX , PPMIN
   29 FORMAT (6X,26H PRESSURE AT FEED SLOTS = ,F10.5,12H LBF PER IN2,//,
     16X,29H PRESSURE AT EXHAUST SLOTS = ,F10.5,12H LBF PER IN2,//)
      WRITE (2,33) TOTFO,TOTHET,TOSHFO,TOSVFO
   33 FORMAT(7X,17H SUM OF FORCES = ,F10.5,5H LBF,8X
     130H DIRECTION OF FORCE (THETA) = ,F10.5,8H DEGREES,//,7X,
     227H SUM OF HORIZONTAL FORCE = ,F10.5,5H LBF,8X,
     325H SUM OF VERTICAL FORCE = ,F10.5,5H LBF,//)
C        TO FIND THE TORQUE REQUIRED OVER THE PAD SEGMENT.
      DO 90 N = 50,1000,50
      REV = N
      VELO = N*VEL
      DO 8 I = 1,NBETA+1
      DO 8 J = 1,NALFA+1
      IF ( J.EQ. 1) GO TO 103
      IF ( J.EQ. NALFA+1 ) GO TO 104
      PD = (R(I,J+1)-R(I,J-1))
      GO TO 106
  103 PD = (R(I,J+1)-PPMIN)
      GO TO 106
  104 PD = ( PPMIN -R(I,J-1))
  106 TOR(I,J) = ABS(PD)*HO/(4.0*ALFA*PLEN)+(VIS*VELO/HO)
    8 CONTINUE
      DO 11 I = 1,NBETA+1
      ODD = 0.0
      EVEN = 0.0
      DO 12 J = 2,NALFA,2
   12 EVEN = EVEN + TOR(I,J)
      DO 13 J = 3,NALFA-1,2
   13 ODD = ODD +TOR(I,J)
      STOR(I) = (ALFA /3.0)*(TOR(I,1)+4.0*EVEN+2.0*ODD+TOR(I,NALFA+1))*
     1PLEN
   11 CONTINUE
      ODD = 0.0
      EVEN = 0.0
      DO 14 I = 2,NBETA,2
   14 EVEN = EVEN +STOR(I)
      DO 15 I = 3,NBETA-1,2
   15 ODD = ODD +STOR(I)
      FOTOR = (BETA/3.0)*(STOR(I)+4.0*EVEN+2.0*ODD +STOR(NBETA+1))*BRED
C        TO FIND THE TORQUE DUE TO THE PLAIN SIDE OF BEARING
      FOPLB = (VIS* VELO/HO)* BRED*(RLEN-PLEN)
      TOTOR = (FOTOR +FOPLB)*DIA/ 2.0
      TOPAD = FOTOR *DIA/ 2.0
      TOPLR = FOPLB *DIA/ 2.0
      WRITE (2,28) REV
   28 FORMAT (21H SPEED OF ROTATION = ,F10.5,14H REVS PER MIN ,/)
      WRITE (2,17) TOTOR,TOPAD,TOPLB
```

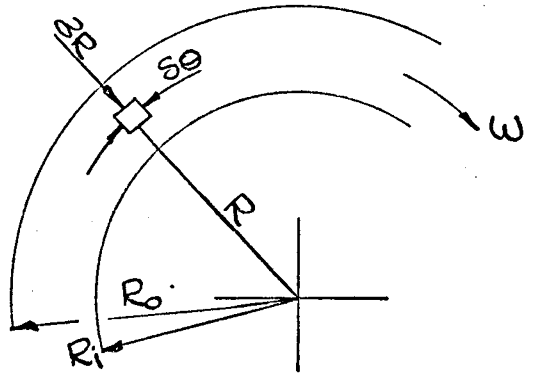
```
 17 FORMAT(6X,16H TOTAL TORQUE = ,F10.5,8H LBF IN,/6X,
   130H TORQUE DUE TO PRESSURE PAD = ,F13.9,8H LBF IN,/6X,
   231H TORQUE DUE TO PLAIN BEARING = ,F13.9,8H LBF IN,//)
 90 CONTINUE
    IF (M6 .EQ. 2) GO TO 102
    M6 = 2
    GO TO 50
C         TO CALCULATE THE FRICTIONAL TORQUE DUE TO THE END COVERS
102 RAD1 = DIA/2.0
    RAD2 = RAD1 *0.333
    DIA2 = 2.0*RAD2
    WRITE (2,19) DIA,DIA2
 19 FORMAT(6X,27H OUTSIDE DIAMETER OF PAD = ,F10.5,5H INS,6X,
   126H INSIDE DIAMETER OF PAD = ,F10.5,5H INS,//)
    DO 94 N = 50,1000,50
    REVO = N
    ANGVEL = ( REVO /60.0)* 2.0*PI
    TOREND = PI*ANGVEL*VIS* (RAD1**4 -RAD2**4)/( 2.0* EHO)
    TOTEND = 2.0* TOREND
    WRITE (2,28) REVO
    WRITE (2,18) TOREND ,TOTEND
 18 FORMAT(6X,25H TORQUE DUE TO END PAD = ,F10.5,8H LBF IN,//,
   16X,37H TOTAL TORQUE DUE TO BOTH END PADS = ,F10.5,8H LBF IN,//)
    HP = ANGVEL* 1.0 /(550.0 *12.0)
    WRITE (2,42) HP
 42 FORMAT(6X,44H THE HORSE POWER PER UNIT TORQUE (LBF IN) = ,F10.5,/)
 94 CONTINUE
    STOP
    END
```

A P P E N D I X V

The frictional torque due to an annulus running on a flat surface, with film thickness h.

Let the relative angular velocity be ω and assume that the circumferential component of velocity of the fluid varies linearly across the film.

Consider the elementary section shown



Viscous shear stress = $\eta \dfrac{\omega R}{h}$ ____________ (24)

And Area = $R\, d\theta\, dR$

Therefore the shear force on the element

$$= \eta \frac{\omega R^2}{h} dR\, d\theta \qquad \text{\_\_\_\_\_\_\_\_\_\_ (25)}$$

Therefore the total frictional torque

$$= \eta \frac{\omega}{h} \int_0^{2\pi} \int_{R_i}^{R_o} R^3\, dR\, d\theta$$

$$= \eta \frac{\omega}{h} \int_0^{2\pi} \frac{(R_o^4 - R_i^4)}{4} d\theta$$

∴ Frictional torque = $\dfrac{\pi \eta \omega}{2h} (R_o^4 - R_i^4)$ ____________ (26)

PART 4

THE DESIGN OF A MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

7. A MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

7.1. INTRODUCTION

7.1.1. Areas of Further Research

7.2. SPECIFICATION FOR A MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

7.2.1. Design Considerations for the Hydraulic System

7.2.2. The Collar Valve

7.2.3. Design of the Collar Valve

7.2.4. Other Components

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PART 4

THE DESIGN OF A MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

7. A MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE

7.1. Introduction

Previous work on the first Hydraulic Circular Weft Knitting Machine can be seen in Part 1 Chapter 3. This work has demonstrated that a knitted fabric can be produced on a camless knitting machine by using individual hydraulic actuators to move a latch needle through the desired time displacement profile.

7.1.1. Areas of Further Research.

One of the most significant factors arising from this research was that each needle transposed position individually. This gave two main advantages:-

i) Only one needle was knitting at any particular instance.

ii) The number of needles required for a knitting station could be greatly reduced.

The first advantage (i) was inherent in the hydraulic motion and should subsequently lead to higher needle speeds. The second factor (ii) will enable more feeders to be introduced round a machine so ultimately producing a greater volume of fabric per revolution of the machine. The number of needles required per knitting station could be as few as four thus giving:-

(iii) one needle in the knit position

(iv) one needle in the tuck position, and

(v) two needles in the miss position.

A second feature worthy of further development was the integration of the rotary valve and actuator block. The actuator block could be made into the rotor of the valve, and a collar placed round it to produce high and low pressure annulii. These would be positioned radially and axially around the actuator block. On rotation, the system would produce a series of pressure and exhaust pulses to drive the actuators through the pre-described time-displacement profile. This technique of using a collar valve would enable a large number of feeders to be introduced round a machine and also eliminate the complex pipework required between an actuator block and a separate rotary valve, see figure 17.

A proposal to build a multi-feeder machine was accepted by Courtaulds Educational Trust and the money for a fourth year granted.

7.2. Specification for a Multi-feeder Hydraulic Circular Weft Knitting Machine

The objective of this second prototype machine was to assess the feasibility of using a collar valve to give an integrated actuator-valve system and to prove that a multi-feeder machine could be made to knit successfully.

After consultation with Mr. F. Carrotte (Technical Director of Kirkland Engineering Ltd) it was decided that considerable time both in design and manufacture could be saved by using the framework of an existing circular weft knitting machine. The advantages to be gained by using an existing machine were:-

i) The fabric tensioning and take-down mechanism could be used without modification thus removing the limitations imposed by the previous fabric tensioning device.

ii) The actuator block would be rotated so the creel would be kept stationary again conforming to the existing machine

iii) A drive mechanism already existed for driving the needle cylinder and this could readily be adapted for powering the actuator block.

iv) The structure of the machine provided a solid base upon which to build the hydraulic system,

A redundant experimental knitting machine was donated to the Department by Mr. Carrotte and its framework was to be used as the basis for the second hydraulic circular weft knitting machine.

7.2.1. Design Considerations for the Hydraulic System.

The actuator block and trix mechanism built for the first prototype machine (see Chapter 3) was examined with the view to retaining the actuators for the second machine. The actuator block could be modified by re-machining the outside diameter, so removing the pipe connections and producing the rotor for a rotary collar valve. The salvaging of the first actuator block would save on manufacturing time and also enable the proven knitting aspect to be retained.

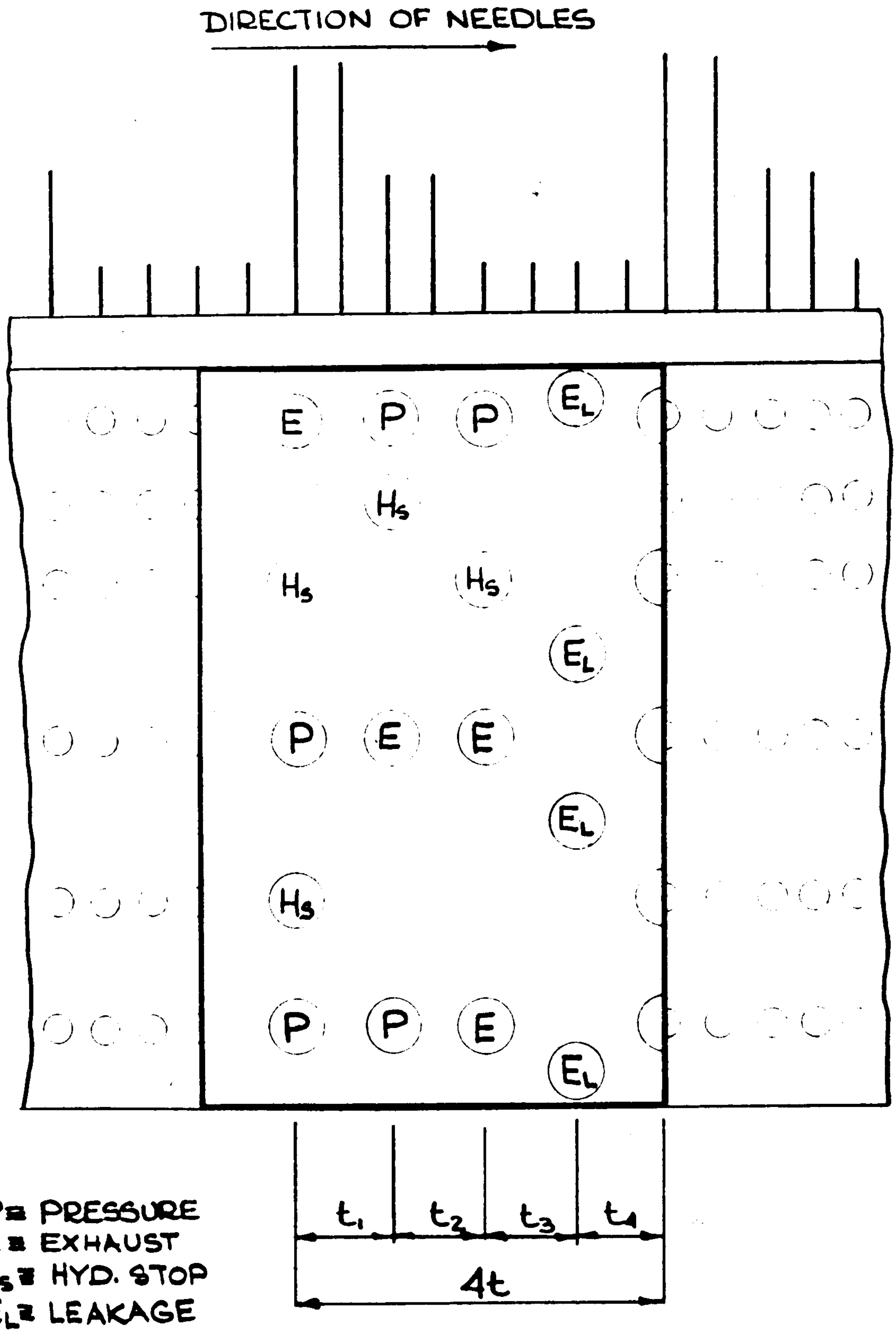
Therefore, the only major component to be considered was the collar of the rotary collar valve. This collar had to be designed to accommodate the existing actuator block which imposed several restrictions:- one being that the overall length of the collar was limited by the length of the existing actuator block.

However, after a detailed investigation it was found to be feasible to use the existing actuator block and was thus adopted. The only outstanding decision to be taken for the multi-feeder machine was to specify the number of feeders.

7.2.2. The Collar Valve.

For ease of machining and symmetry, the number of feeders had to be a multiple of ninety-six. The time displacement profile had been previously considered as occupying four equal time intervals. (see figure 1), so making the number of needles per knitting station a multiple of four, (if the same number of needles were to occupy the same state at any particular instance). Consequently for the purpose of the first multi-feeder machine a general compromise was made thus giving a twelve feeder machine with eight needles per knitting station. This configuration would prove if hydraulic knitting techniques could be applied to multi-feeder machines and still allow space to accommodate the yarn carriers round the trix. It also would produce a symmetrical needle formation with two needles in the knit position, two in the tuck position and four in the miss position. The pulses required to drive the actuators through the time displacement profile have been outlined in Table 1 and the collar valve technique consists of creating a series of pressure and exhaust ports so that as the actuator block rotates, the actuators receive the correct sequence of pulses. A diagrammatical representation of the pressure and exhaust positions for a knitting station can be seen in figure 96. As the actuators pass a

A DIAGRAM TO SHOW THE PORT GEOMETRY OF AN EIGHT NEEDLE KNITTING STATION ON THE COLLAR VALVE.



P = PRESSURE
E = EXHAUST
H_s = HYD. STOP
E_L = LEAKAGE

FIG. 96

different row of ports corresponding to a time interval t, the pulses described in Table 1 will be generated.

Hence the specification for the Multi-feeder Circular Hydraulic Knitting Machine:-

(v) Circular configuration with ninety-six needles spaced at four needles to the inch.

(vi) Twelve equally spaced feeders.

(vii) The actuator block to rotate anticlockwise to conform with the existing knitting machine components.

7.2.3. Design of the Collar Valve.

Twelve sets of ports (as shown in figure 96) were positioned radially round the annular block. The shape of the ports was not regarded as critical provided that sufficient pressurised oil could be delivered to the actuator in the time available. Consequently for ease of manufacture, all the supply ports in the collar valve were made circular. The designated positions of all the pressure and exhaust ports were used as data to check the out-of-balance forces due to the pressure distribution using the computer programme MASTER PRESSURE DISTRIBUTION OF SLOTS (see Appendix 1).

This indicated that using twelve equally spaced supply areas, the system was inherently balanced, although physical dimensions of the collar necessitated using a large grid size, which tended to reduce the overall accuracy of the calculation.

The remaining features to be resolved were the fixing of the end seals on the actuator block and the positioning of the exhaust passages allowing the rotor leakage oil to pass back to tank. The sealing problem was solved by the use of standard rotary shaft seals with a back-up ring to support the seal when subjected to pressures greater than 8 lbf/in^2. This pressure in the clearance at the back of the seal was regulated by a series of exhaust passages taking the leakage oil back to tank. The rotor leakage oil was also exhausted at two other points on the actuator block so providing specific low pressure areas round the valve. These exhaust points were grouped at the position where the actuators were in the fourth part of the cycle, that is being held in the miss state prior to the start of the next cycle.

The design for the collar valve can be seen in figures 97 and 98 which show the detail drawings J.D.G. 150 and J.D.G. 151 respectively. This component was to be manufactured in Meehanite cast iron, the same material as the actuator block. The actuator block was modified to drawing J.D.G. 153 (see figure 99) and these two components then formed the integrated actuator-rotary collar valve. Photographs of the collar valve, actuator block and the integrated actuator-rotary collar valve can be seen in figures 100, 101 and 102 respectively.

7.2.4. Other Components

The donated circular weft knitting machine was dismantled to the basic framework. The only features to be retained were the fabric tensioning mechanism and the circular driving gear for the needle cylinder. The electric motor previously used to drive the machine was replaced by a hydraulic motor so that the driving speed could be varied for test purposes. A Vickers M2-200 Vane

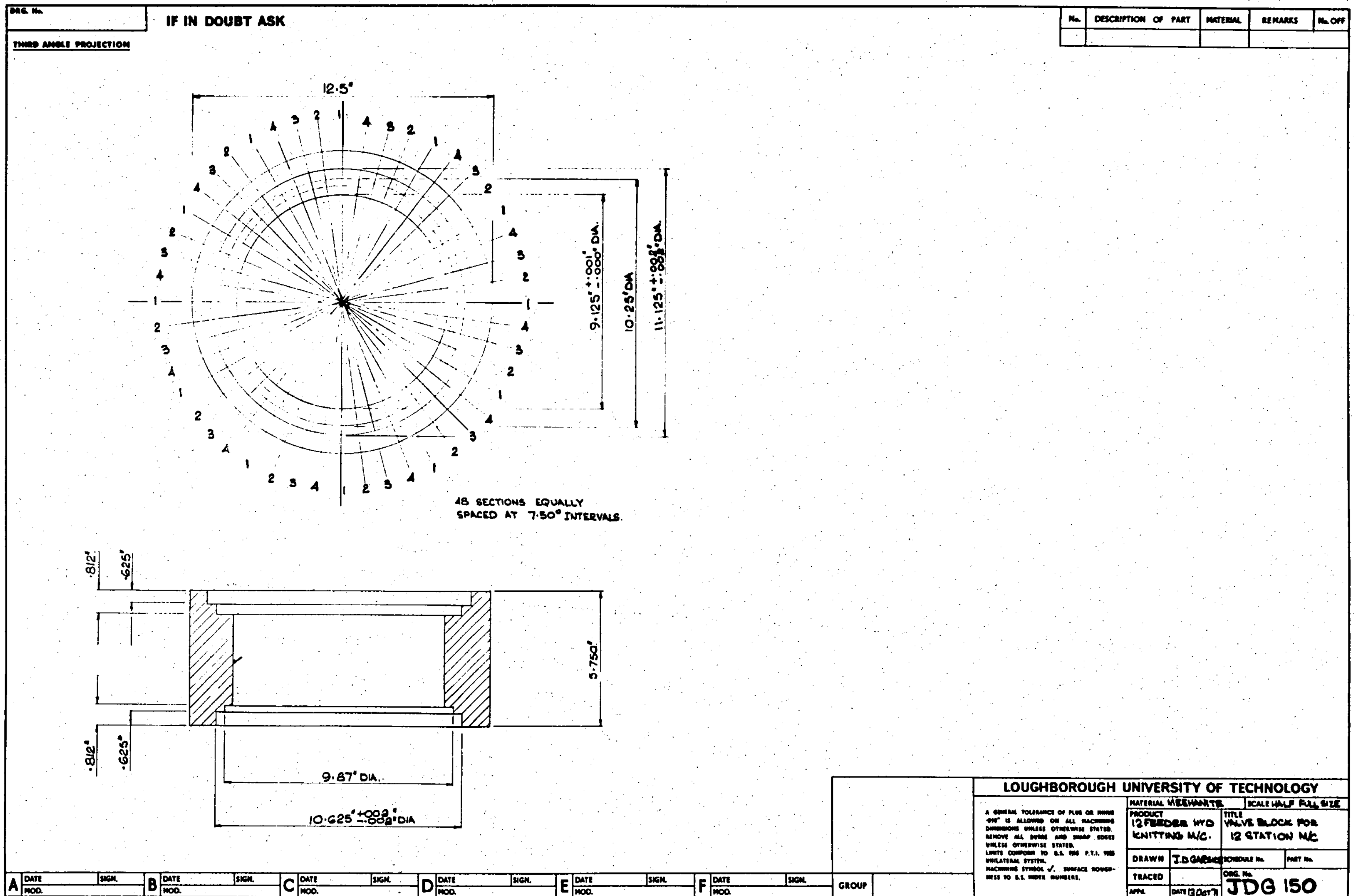
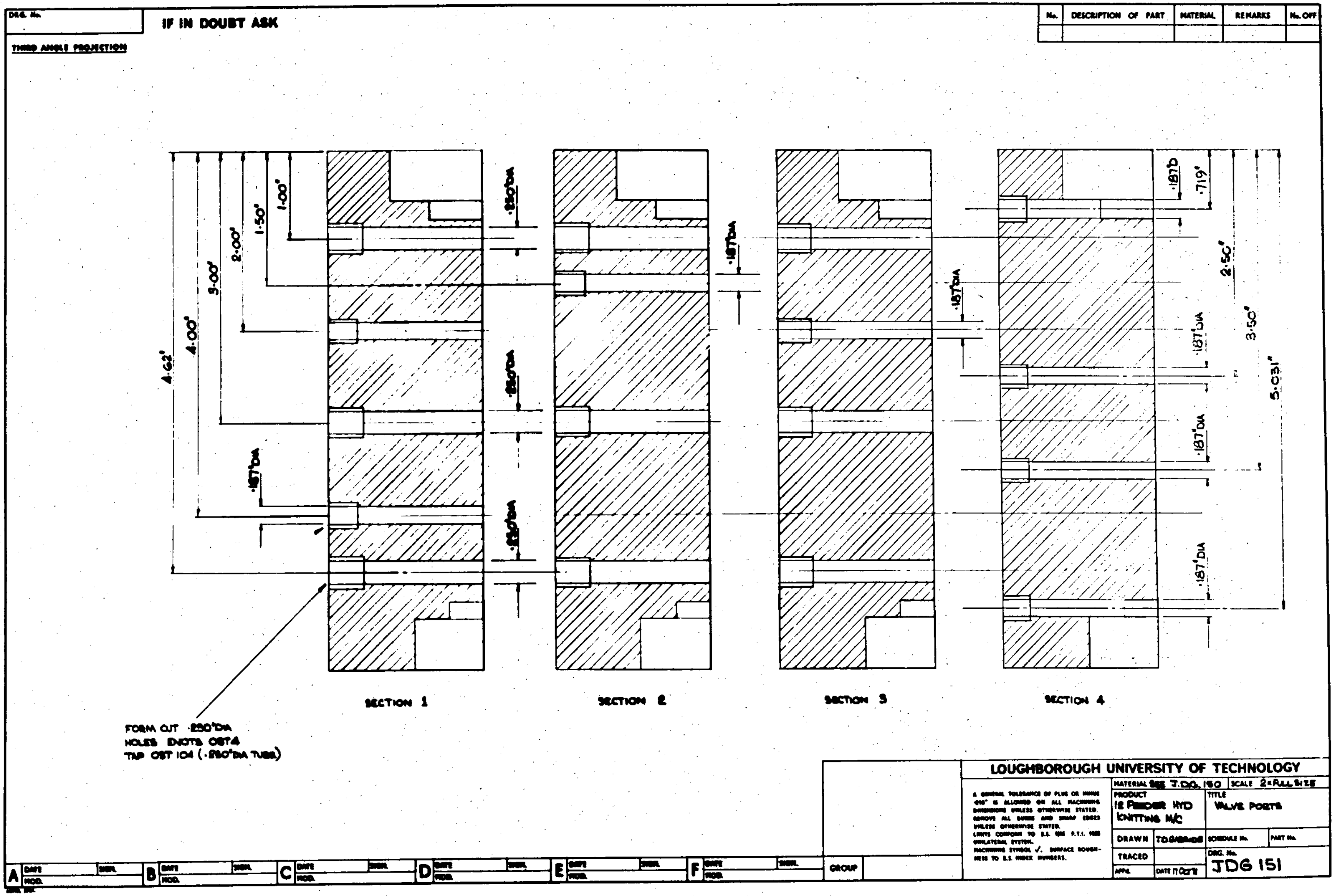

DRG. No.
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No. | DESCRIPTION OF PART | MATERIAL | REMARKS | No. OFF
12.5"
9.125" +.001" -.000" DIA.
10.25" DIA
11.125" +.002" -.002" DIA.
48 SECTIONS EQUALLY SPACED AT 7.50° INTERVALS.
.812"
.625"
5.750"
9.87" DIA.
10.625" +.002" -.002" DIA
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL MEEHANITE
SCALE HALF FULL SIZE
PRODUCT 12 FEEDER HYD KNITTING M/C.
TITLE VALVE BLOCK FOR 12 STATION M/C
DRAWN T.D.GARSIDE
TRACED
DATE 13 OCT 71
DRG. No. JDG 150
GROUP
A DATE SIGN. MOD.
B DATE SIGN. MOD.
C DATE SIGN. MOD.
D DATE SIGN. MOD.
E DATE SIGN. MOD.
F DATE SIGN. MOD.

FIG. 97



DRG. No.
IF IN DOUBT ASK
THIRD ANGLE PROJECTION
No.
DESCRIPTION OF PART
MATERIAL
REMARKS
No. OFF
4·62"
4·00"
3·00"
2·00"
1·50"
1·00"
·250"DIA
·187"DIA
·187 D
·719"
2·50"
3·50"
5·031"
SECTION 1
SECTION 2
SECTION 3
SECTION 4
FORM OUT ·250"DIA
HOLES ENOTS OST4
TAP OST 104 (·250"DIA TUBE)
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL SEE J.D.G. 150
SCALE 2×FULL SIZE
PRODUCT
12 FEEDER HYD KNITTING M/C
TITLE
VALVE PORTS
DRAWN
TRACED
DRG. No.
JDG 151
GROUP

FIG. 98

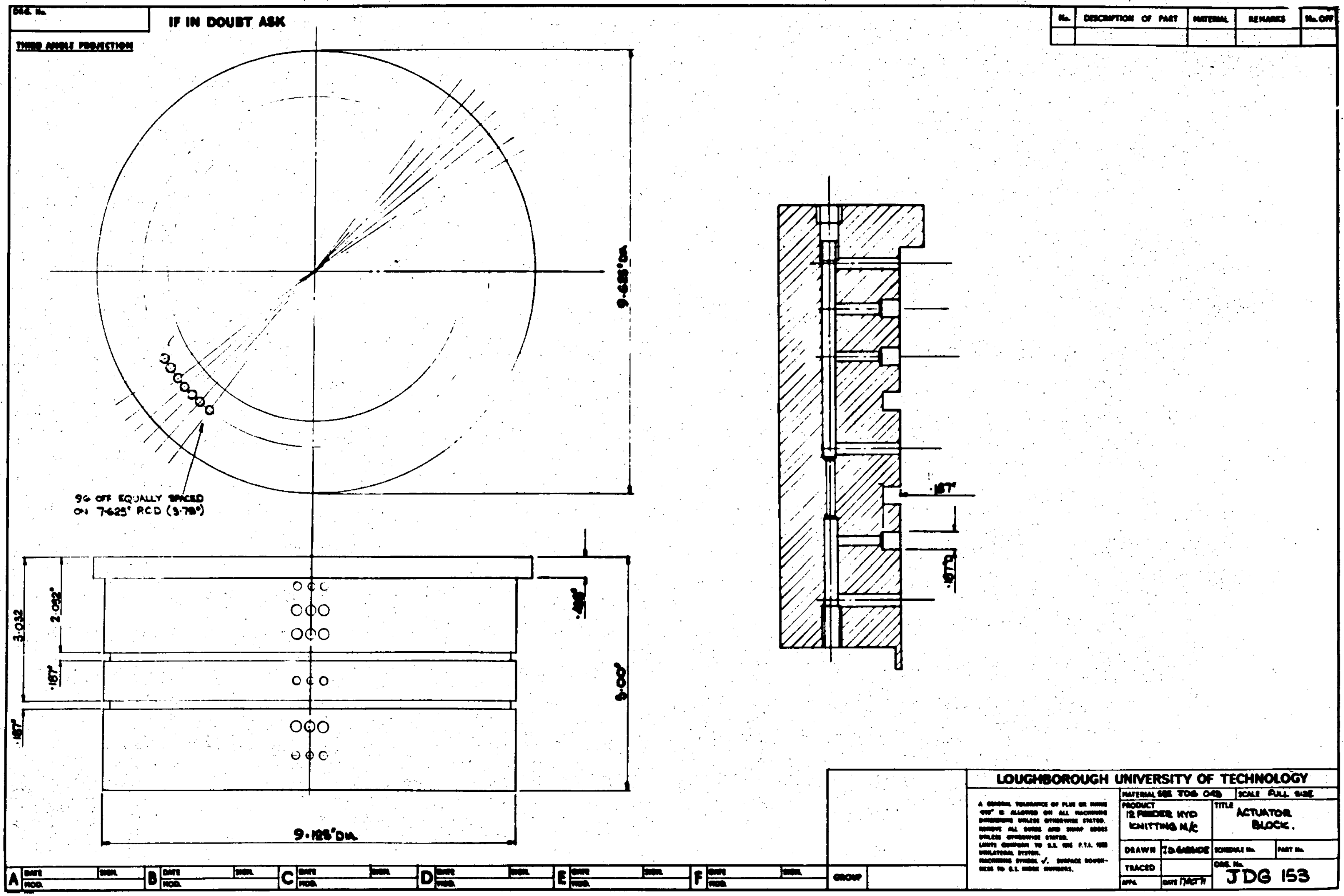

IF IN DOUBT ASK
THIRD ANGLE PROJECTION
9·625" DIA.
96 OFF EQUALLY SPACED ON 7·625" P.C.D (3·75°)
3·032
2·032"
·187"
·187"
5·00"
9·125" DIA.
·187"
LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
MATERIAL SEE TDG 043
SCALE FULL SIZE
PRODUCT 12 FEEDER HYD KNITTING M/C
TITLE ACTUATOR BLOCK.
DRAWN
TRACED
DRG. No. JDG 153
GROUP

FIG. 99

281

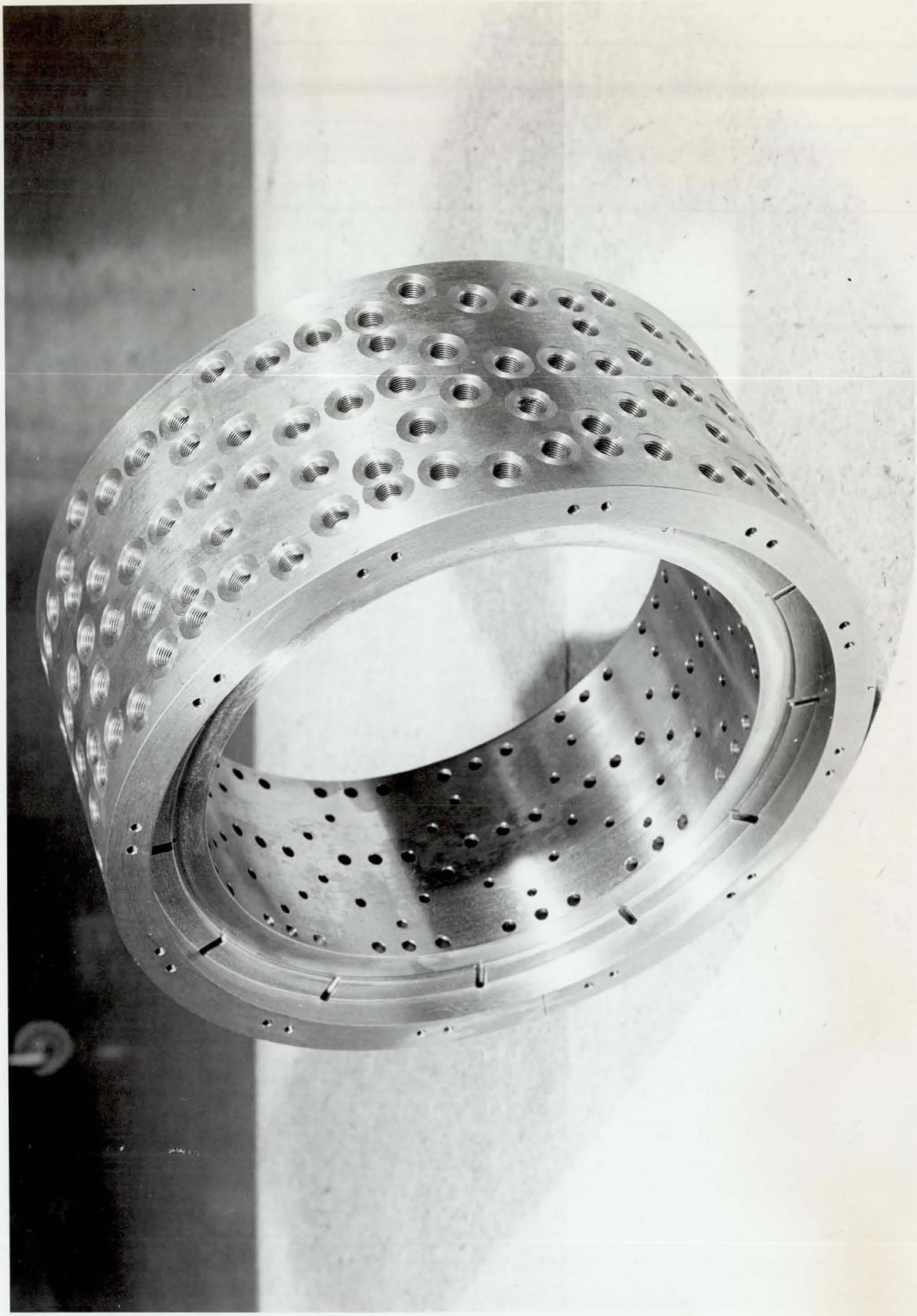


FIG. 100 COLLAR VALVE

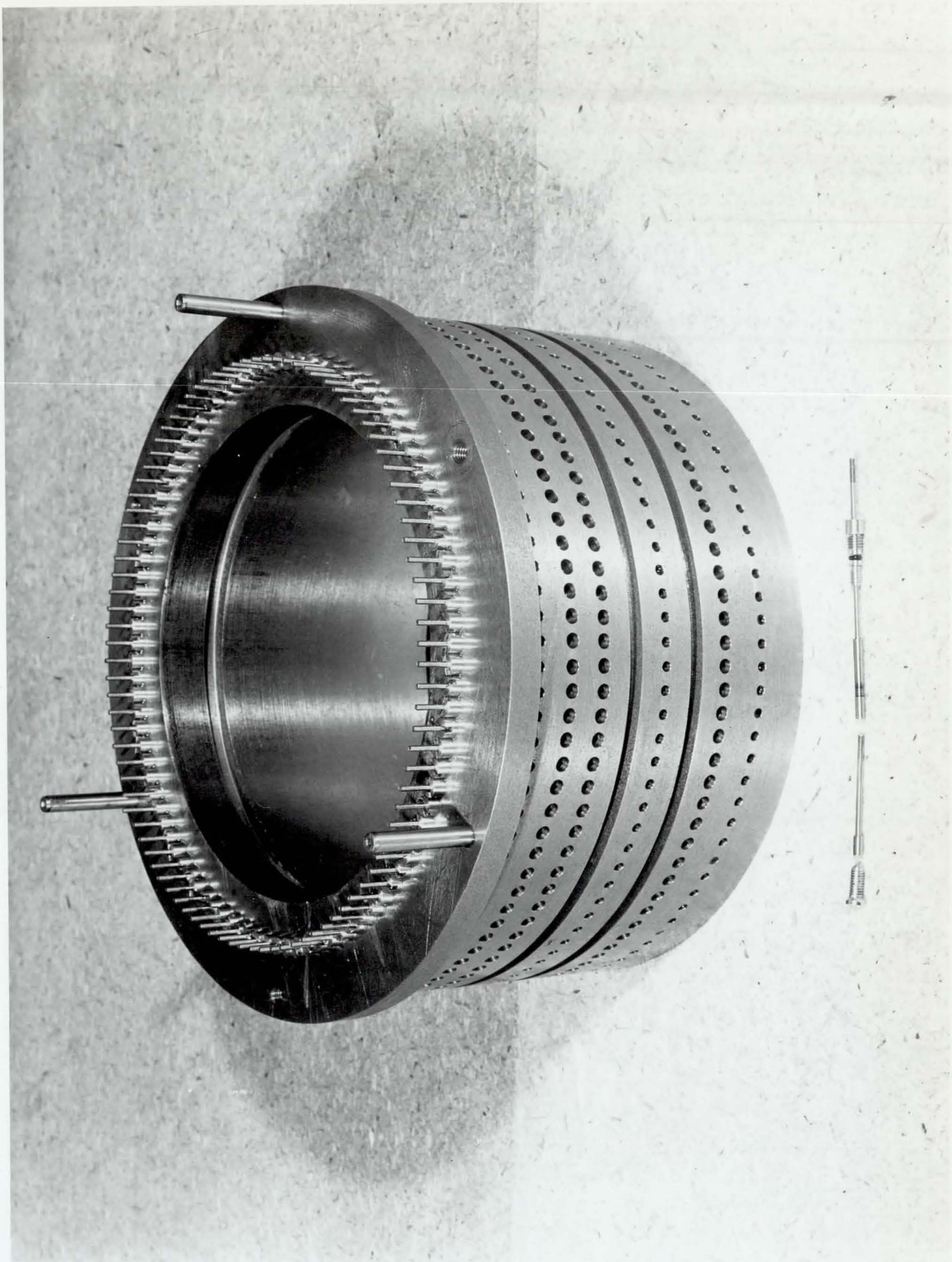


FIG. 101 ACTUATOR BLOCK

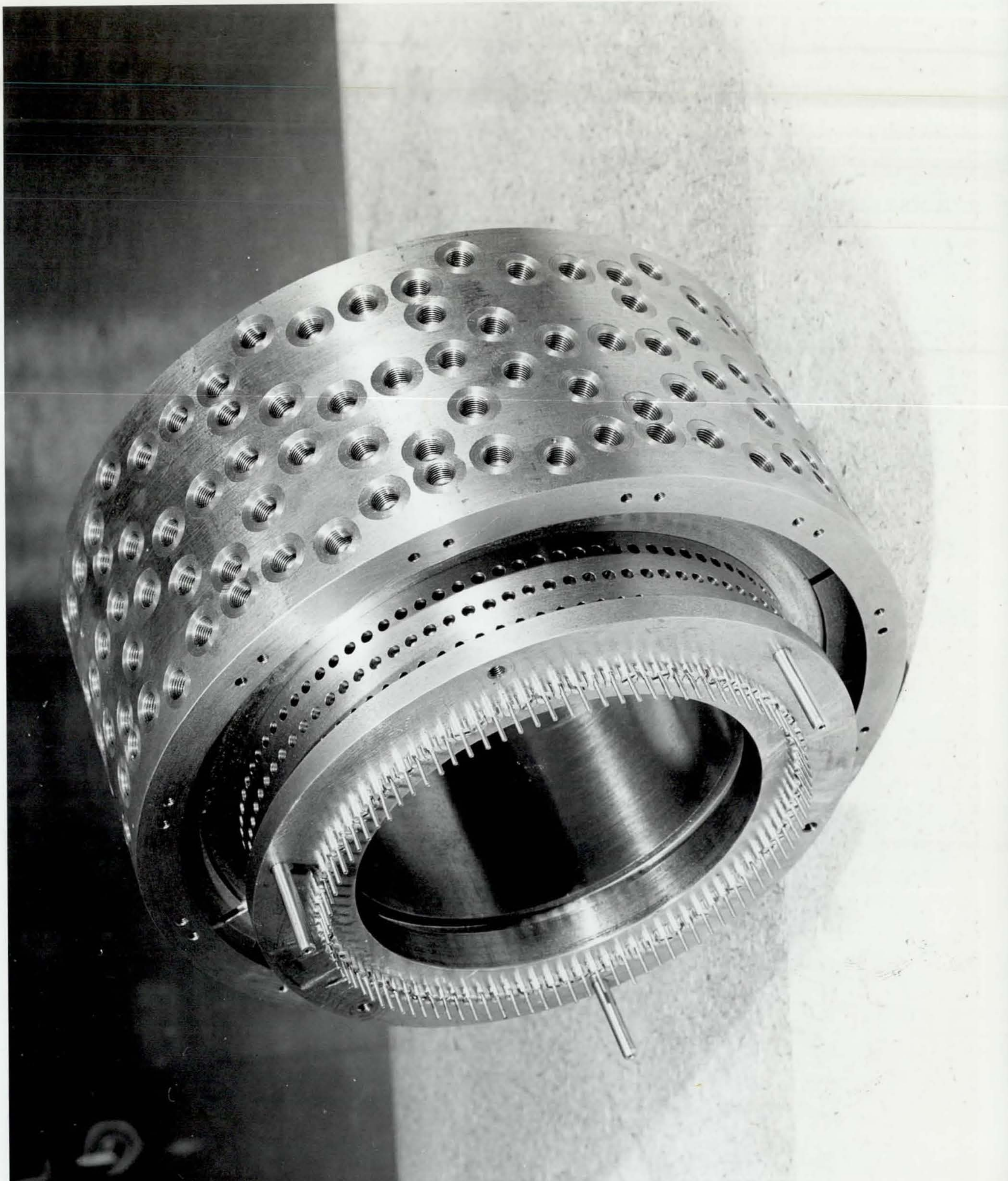


FIG. 102 INTEGRATED ACTUATOR ROTARY COLLAR VALVE

was mounted on to the input gearbox, giving a direct drive to the cylinder rack. This motor was to be coupled to a Vickers PBV 6 variable delivery piston pump so that the knitting speed could be manually controlled.

The only other components to be designed were adapting discs (these were manufactured from steel plate). The discs enabled the framework to accommodate the integrated actuator collar valve and provide a link to the existing needle cylinder drive. Figure 103 shows the knitting machine with the adaptor discs and actuator block in position. These were all securely bolted to the existing framework after having been checked for concentricity using a clock dial gauge.

7.2.5. Hydraulic Circuits.

For each knitting station the collar valve required five pressure supplies and twelve exhausts. These could all be taken from a common manifold at each knitting station, so a series of pressure manifolds (each manifold to supply two knitting stations) were positioned round the machine with six exhaust manifold blocks interspaced between the pressure supplies. The system pressure for driving the actuators was expected to be less than 400 lbf/in^2 so nylon flexible hoses could be used to link the collar valve to the manifold blocks. A colour code was used to enable the supply to the various parts of the actuator to be distinguishable. The main supply to the pressure manifolds was taken via a continuous copper ring thus linking the system. At a point on both sides of the first pressure block, two shut-off valves were introduced to enable the machine to operate as a

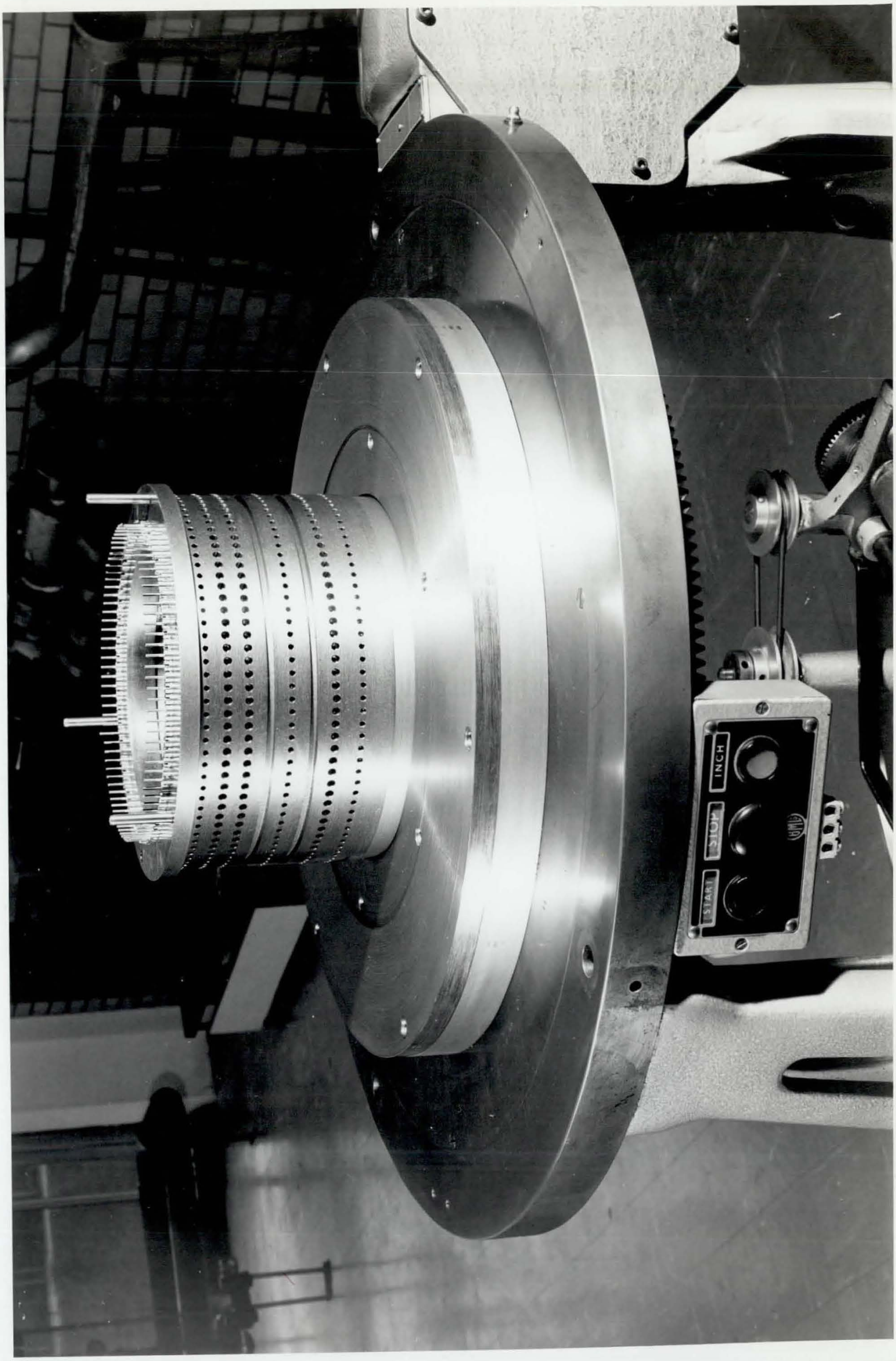


FIG. 103 ACTUATOR BLOCK IN POSITION

twin feeder machine on commencement of knitting. The exhaust manifolds were again linked to the collar valve with nylon pipe, and some lengths of smaller bore pipe were introduced to act as hydraulic stops in both mid and tuck positions of actuator displacement. The exhaust from each manifold which served two knitting stations, was taken back to tank via an independent path. Each return line was formed in copper pipe, from the manifold block to a common point under the driving motor on the machine. This group of six pipes was then linked to the power pack reservoir with .75 inch diameter nylon tube. The pipework for this multi-feeder machine was much simplified and required only half the number of connections than the first machine.

7.3. Power Requirement for the Actuators

The Power requirement to move the actuators through the time displacement profile cannot be rigorously calculated at this stage in the project due to the internal leakage flows within the collar valve. This leakage flow will run partially into the exhaust grooves in an axial direction, whilst the remainder will run both radially and axially across the radial clearance that links two adjacent collar valve ports. The method of calculating this leakage flow would be again by using hydro-dynamic bearing theory to calculate the pressure distribution. This pressure pattern could be used to estimate the flow of oil. The mathematical study required to estimate the leakage flow would be relatively complex and not until the technique of using a rotary collar valve has been fully established and proved to function effectively can such analysis be deemed worthwhile.

Practical tests on the first hydraulic knitting machine established that a knitted structure could be produced using a supply pressure of 300 lbf/in^2 on a .125 inch diameter piston. This pressure produced sufficient force to overcome the frictional resistance due to the 'O' seal; the knitting needle friction on the trix; the inertia of the moving components; plus the force required to pull the new yarn through the old loop. Therefore a nominal working pressure for the system could be regarded as 300 lbf/in^2.

The flow of oil required per actuator can be considered to comprise of three elements:-

i) The volumetric displacement of the pistons

ii) The leakage past the pistons

iii) The leakage through the hydraulic stop at the end of each piston movement.

The volumetric flow per cycle is easily calculated from the physical dimensions of the actuator and is found to be .0272 in^3. The theoretical leakage flow in the annular clearance between the piston and the cylinder is .004266 in^3/sec. Both these values have been calculated in the author's previous thesis (see chapter 4 reference 18). It is also shown that the leakage through the hydraulic stop at the end of each piston movement is responsible for controlling the volume of oil required for each actuator. The volume of oil required for powering ninety-six actuators in sequence is a function of the length, diameter and pressure drop across the hydraulic stop, plus the transient time of each piston movement which governs the length of time that each hydraulic stop is in operation.

Prior to carrying out exhaustive tests on the rotary collar valve actuator system, estimates for the power requirements can only be gauged from the performance of the previous hydraulic knitting machine. This machine had 25% of the needles supplied with pressure oil at any particular instance and required approximately eleven gallons of oil per minute. The new collar valve system has main supply ports spaced radially at .5 inch intervals. To supply actuators at .25 inch pitch means that 50% of the actuators are being supplied with oil at any particular time. Consequently the estimated volume of oil required per minute (allowing for the leakage of the collar valve) is approximately 25 galls/min. This figure has been selected for the size of pump required, but once the system has been found to knit satisfactorily extensive tests will be required to optimise the length of the hydraulic stops. The shape of the time-displacement profile could be measured together with the pressure readings at the hydraulic stops so that a real estimate of power loss could be obtained. An alternative to the hydraulic stop may be small reservoir which would absorb the initial surge of pressure but control the volume of fluid pumped directly back to tank. The power losses within the system are of prime importance and will have a direct influence on the ultimate success of the hydraulic knitting machine and it is hoped that this first multi-feeder hydraulic knitting machine will provide useful data upon which to base future decisions.

7.4. The Knitting Aspect

The basic knitting mechanism for the multi-feeder machine were the needles, and the existing knitting trix. Prior to dismantling the first machine, tests were performed to assess methods of attaching the needles to the actuators. One method that functioned satisfactorily was to leave on part of the needle butt to form a hook at the bottom of the needle. This hook could then be attached to an eye in a slug mounted onto the end of the actuator. This technique removed the soldered joint and produced a flexible coupling which would allow compensation for any slight mis-alignment. The slug was attached to the actuator rod by means of a screw thread which enabled linear adjustment to be made to each needle independently. In order to accommodate this modification, the knitting trix had to be re-machined to allow for the slight mis-alignment between the actuator rod and the back of the needle. This technique necessitated restraining the needle from moving forward out of the trix; this was achieved by machining a groove round the top of the trix to accept a spring retaining hoop.

7.4.1. Yarn Carrier.

The yarn carrier developed for the previous machine provided the basic pattern for the new yarn carrier. The overall length was reduced to suit the more compact time-displacement profile, but these basic mechanisms were all adopted at the outset:-

i) a brush to open any closed latches

ii) a guard to keep the latches open while travelling from the knit to the tuck position and,

iii) a chamfered leading edge to control the latches as the needle travelled from the miss to the knit position.

The twelve yarn carriers were mounted onto a detachable ring that provided an adjustment both radially and axially, so enabling each yarn carrier to be adjusted to suit a particular knitting station.

7.4.2. The Creel

Using the configuration of rotating the needles, the creel remains stationary so enabling the existing framework for the bobbin holders to be used. New bobbin holders were supplied by Kirklands, and since the knitting periods would be relatively short the introduction of stop motions was not warranted at this time. The yarn was also supplied by Courtaulds, but two ends of yarn per knitting station were required to make standard yarn more suitable for the coarse gauge machine. This had an advantage in the fact that if one yarn broke then the knitting would still be retained on the needles. Therefore, provision for twenty four cones of yarn was made on the creel and the yarn brought down to the carriers through a series of pot eyes.

7.5. Hydraulic Power Pack

A special power pack was built by Hymid Hydraulics Ltd to meet the estimated power requirements of the multi-feeder knitting machine. This power pack comprised of:-

i) a thirty gallon reservoir

ii) a twenty eight gallon per minute submerged fixed displacement vane pump, driven by a 10 H.P. electric motor, for powering the actuators.

iii) a five gallon per minute (maximum) variable delivery pump driven by a 7.5 H.P. electric motor for powering the hydraulic motor to drive actuator block in the rotary collar valve.

The hydraulic circuit for the actuators comprised of a pressure relief valve and a pressure gauge. The pressure relief valve was fitted with a solenoid-operated dumping valve for off-loading the system while not actually knitting. A pressure relief valve and a solenoid-operated directional control valve, made up the hydraulic circuit for the motor. This control valve enabled the speed of rotation to be switched from a slow speed for setting up purposes to the normal running speed required for knitting. All the return pipes terminated at a common manifold and the oil passed through a filter and cooler before returning to tank. (See Appendix VI for Hydraulic Circuit Diagram).

7.6. Commissioning the Multi-feeder Hydraulic Knitting Machine

The hydraulic knitting machine was connected to the hydraulic power pack. The type of pipe fittings required had been previously established so providing compatible connections. The electrical switch gear was mounted onto the power pack and appropriate electrical connections made to the two motors and the solenoid valves.

The system was switched on and allowed to cycle with both the pressure relief valves open. This allowed the pipes to be flushed and remove any contamination that remained in the hydraulic pipework. Gradually the pressure to the actuators was increased and the drive applied to the actuator block. When the supply pressure to the actuators reached approximately 200 lbf/in^2, the actuators began to cycle through the desired time-displacement profile and twelve cam-like shapes appeared round the actuator block. Unlike the previous single feeder machine, these profiles appeared stationary though were similar in form with only one needle transposing position at each knitting station at any given instance. The cycling rate of the actuators was difficult to judge because the knitting profile appeared stationary, regardless of rotational speed. A method of gauging the cycling rate was to take a reference from a fixed member on the actuator block i.e. a trix support, or thetake-down mechanism.
(It had to be remembered that each actuator made twelve cycles per revolution). After an initial test run, minor oil leaks had to be eliminated at the manifold blocks and at the bottom of the actuators. The machine was run intermittently to establish that all the actuators were functioning correctly, prior to the commencement of knitting. At the end of this initial testing period, three actuators tended to stick in the knit position but could easily be released by rotating the actuator rod. These actuators have given some trouble on the previous machine.and the problem was traced to the alignment of the actuator bores.

7.6.1. Testing Actuator Performance.

The integrated actuator rotary collar valve was run at various speeds ranging from 10 to 90 revolutions per minute, (the maximum driving speed for the hydrostatic transmission). At the highest speed, the actuators were operating at 18 cycles per second and still producing the desired time-displacement profile. Examination of the profile showed a slight tendency to overshoot the tuck position. This was remedied by introducing a larger hydraulic stop at the tuck port to enable the pressure oil driving the actuator from the knit to the tuck position to be more quickly exhausted.

7.6.2. Setting the Machine up in preparation for Knitting.

The final task prior to attempting to knit was to complete assembly of the knitting mechanism. Each actuator was fitted with a needle holder and each holder set to a common level in the not-knit position. This was to ensure that each actuator would pull the same length of loop when actually knitting. The needles were then attached to the actuators and the knitting trix introduced onto the actuator block. The creel was fitted with twenty-four cones of yarn and twelve ends brought down through the creel mechanism to the twelve yarn carriers mounted round the top of the trix. A piece of fabric knitted on the first prototype machine was then hooked onto the needles in preparation for knitting. The machine at this stage of development can be seen in figures 104, 105 and 106.

Figure 104 shows a general view of the machine with the yarn and fabric in position. Figure 105 shows detail of the

FIG. 104 GENERAL VIEW OF MULTI-FEEDER HYDRAULIC CIRCULAR WEFT KNITTING MACHINE.

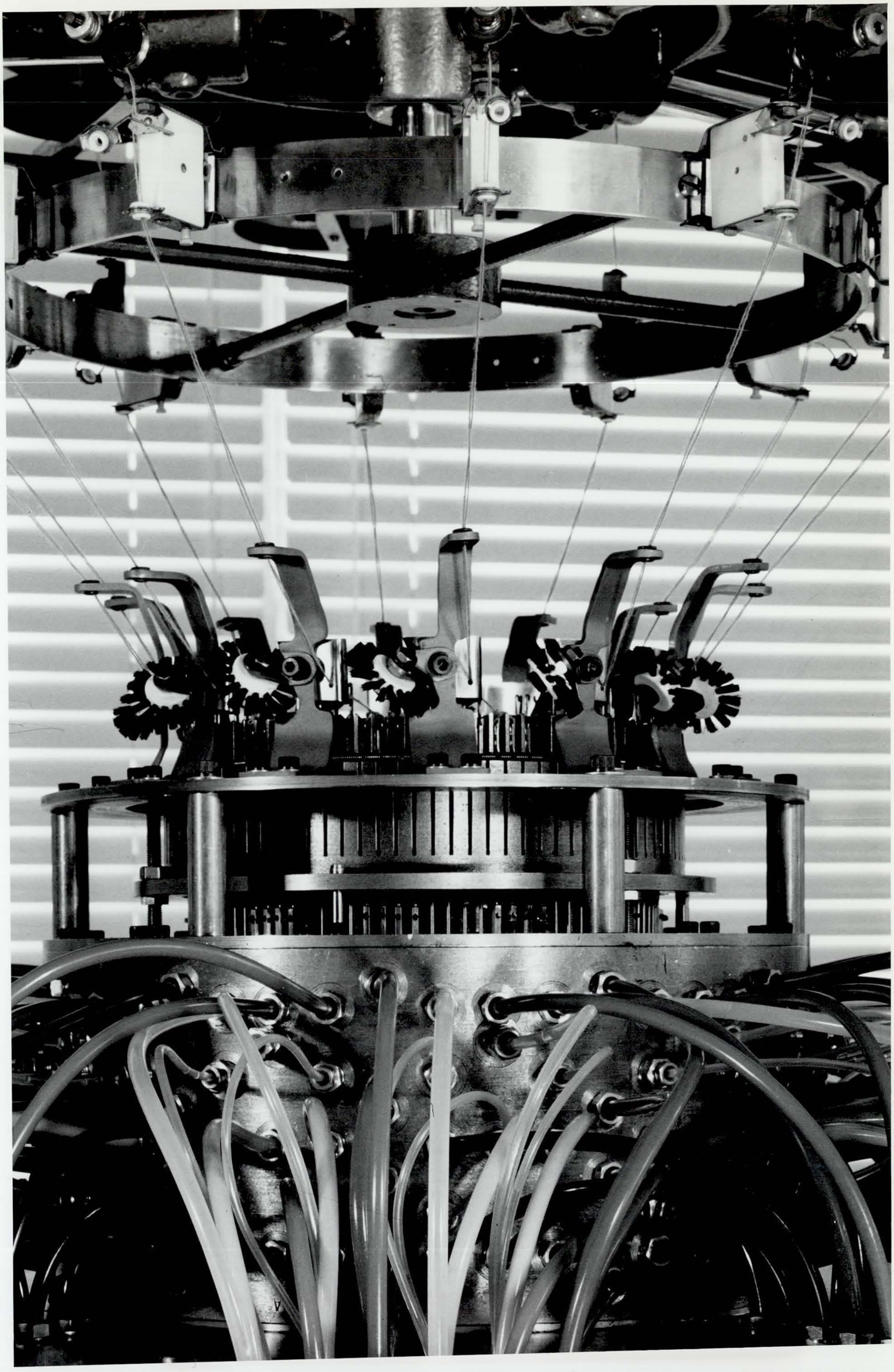


FIG. 105 ACTUATOR BLOCK AND KNITTING MECHANISM

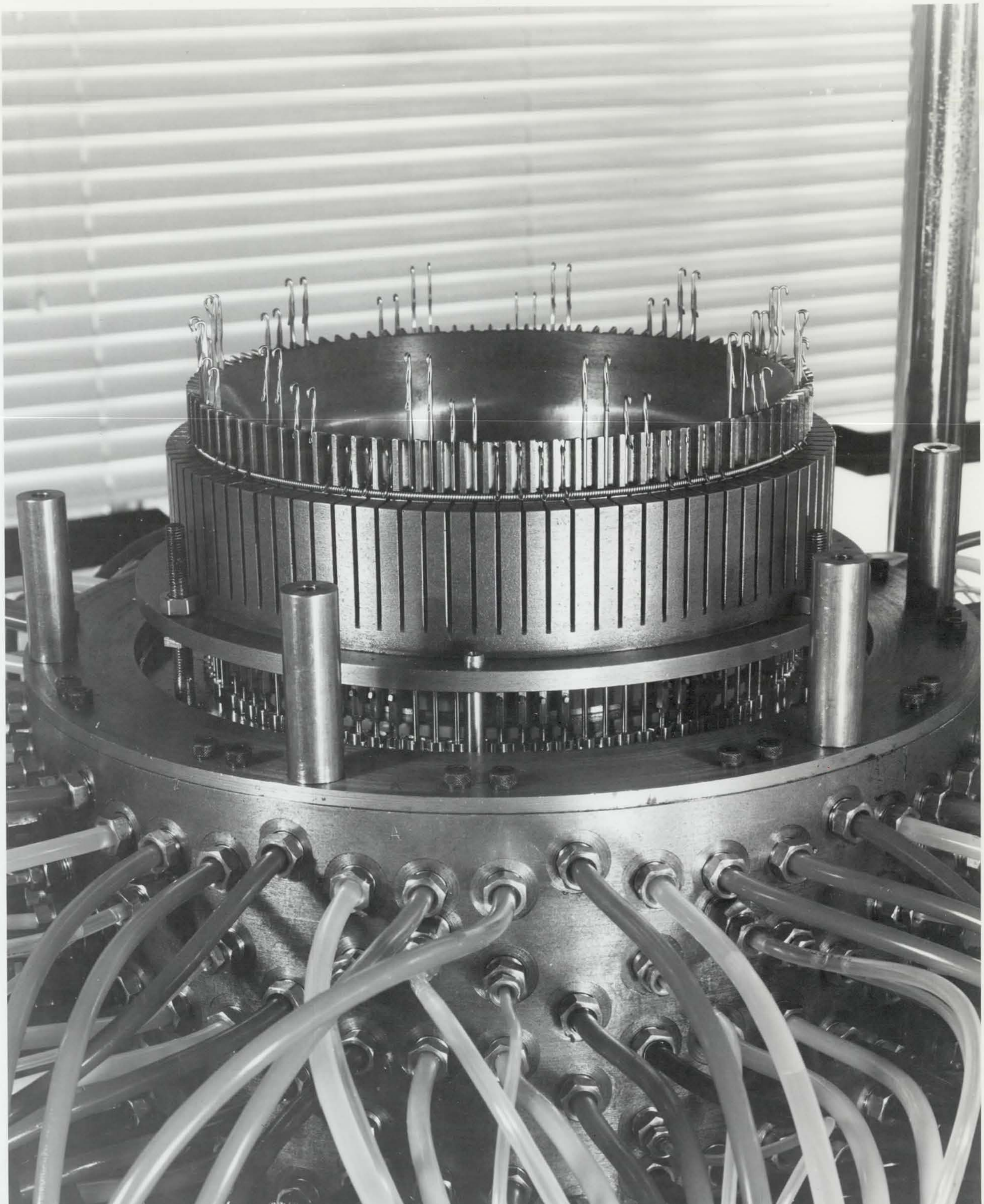


FIG. 106 VIEW OF THE TRIX SHOWING THE KNITTING PROFILE OF THE NEEDLES

hydraulic aspect of the machine, (how the manifold blocks are connected to the collar valve). The yarn carrier mechanism is also clearly visible. The last figure, 106 shows the knitting profile of the needles round the trix, without the yarn carriers in position. This needle profile round the trix mechanism remains static regardless of the rotational speed of the actuator block, so creating a hydraulic actuator system that should be capable of producing a knitted fabric in the near future.

7.6.3. Knitting with the Multi-feeder Hydraulic Circular Weft Knitting Machine.

The scheduled programme for this research will continue for a further six months. Prior to submitting this thesis, time has not been available for studying the new knitting technology necessary to make the hydraulic actuator-rotary collar valve system knit. The techniques involved will be similar to those required for the previous prototype machine, but will be more complex due to the multi-feeder system. A premature initial attempt to knit was made after the photographs had been taken, but (as could well be expected) an error resulted in the running on procedure. Several needles were damaged and no further attempts were possible in the time available.

7.7. Conclusions and Recommendations for Further Research

The integrated actuator rotary-collar valve from a hydraulic aspect functioned as predicted and produced the time displacement profile required for a multi-feeder knitting machine. The overall shape of this profile should be capable of producing a plain knitted stitch using latch needles.

The areas for future research are initially concerned with the testing of the multi-feeder knitting machine:-

i) Devise and investigate the new knitting technology required to make the hydraulic actuator system knit.

This will involve examining the time displacement profile in detail and acquiring a technique for "running on" the new knitting, together with a start-up procedure for commencing knitting.

ii) Experiment with needles and yarn characteristics in order to improve the quality of the knitted fabric.

iii) Test the actuator performance over a long period in order to determine the reliability of the system, and assess the limitations of hydraulic actuation techniques.

iv) Conduct dynamic performance tests on the overall system and measure the power consumption. These tests could be used to improve the overall efficiency of the system and enable the hydraulic stops to be optimised. They would also provide data regarding the oil volume flow rate required by individual actuators.

v) Experiment with the time-displacement profile and attempt to obtain a yarn feeder that would enable plain fabric to be knitted using a rectangular needle profile. (This would eliminate the tuck probe, hence simplifying the actuator system).

Having drawn conclusions from a full test programme on the second prototype hydraulic knitting machine a series of basic policy decisions would have to be taken. These would determine the immediate future of the hydraulic knitting machine. Two probable courses could be:-

vi) Develop a programming technique for producing a tuck and miss stitch. This would enable the range of knitted fabric to be extended from the basic plain knitted structure.

vii) Design and test a further multi-feeder circular hydraulic knitting machine with even less needles per knitting station. This development would also incorporate advances in actuator design and enable preliminary production techniques to be studied.

The techniques evolved in this research have established a new approach for generating a regular time-displacement profile and could well provide the basis for a future generation of textile machines.

APPENDIX VI

CIRCUIT DIAGRAM FOR THE HYDRAULIC POWER PACK