



COLLEGE OF AERONAUTICS
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DESIGN PROJECT 1968.

REPORT OF THE DESIGN COMMITTEE



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SUMMARY

This report outlines the design of a versatile low cost 'Precision Turning Machine with a Limited Milling Facility' which automatically produces work accurately to the required form at a high production rate.

The moving headstock and cross-slide have independent hydrostatic slideways. The headstock spindle and hydraulic motor have integral hydrostatic bearings and the cross-slide toolpost can present both static and rotating tools to the work. A sequencing circuit enables any one of sixteen tools to be selected from a magazine.

General assembly drawings are included of every machine feature together with proving calculations testing the validity of the design.

The aspects of work-loading, cutting lubricant and swarf removal are discussed in the report which concludes with recommendations for the future.

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D.1. INTRODUCTION.

Recommendations put forward by both the Configuration Committee and the Technical Survey Committee were used as a guide in formulating the design.

It was necessary to divide the Design Committee into five sub-committees in order that the design work was adequately covered. Each sub-committee consisted of at least one member of the Design Committee who acted as chairman and two or three project members. This enabled the Design Committee to co-ordinate the work of the sub-committees.

The task of the Committee was to design a machine whose function is to produce work which is performed on a centre lathe but with the added feature that it will also perform other operations which would normally require the use of a milling machine. This necessitated a departure from conventional design, which resulted in a moving headstock cradled between the slideways of a machine frame of a box-type structure, a tool-post which moves at right angles to the work axis in the vertical plane but positioned in the lower part of the machine, and in order to cater for the many different machining operations, a tool magazine to store the tools and to present them to the tool-post at the start of each machining operation.

With all the machine movements being controlled by a punched paper tape various mechanisms had to be designed in order that the machine could respond to these signals.

Other features considered in the design were floor space taken up by the machine, the services that the machine would require, and initial cost together with running cost.

D2. PRELIMINARY CALCULATIONS.

It is necessary to look at the cutting forces in order that the power required at the tool point can be calculated and therefore the machine efficiency determined before commencing the detailed design study.

D2.1. Cutting Forces.

Consider the forces acting on a tungsten carbide knife tool taking the following cut in EN 1A mild steel:-

Depth of cut	'd'	0.250 inches (6.35 mm.)
Feed/rev	'f'	0.010 inches (0.25 mm.)
Cutting speed	'CS'	350 ft/min (108 m/min)
Specific cutting Pressure	'P' for EN 1A	200,000 lb.f/inch ² (1.379 GN/m ²)

Tangential Cutting Force	$F_c = d \times f \times p$
	$F_c = 0.250 \times 0.010 \times 200,000$
	$F_c = \underline{500 \text{ lbf}} (2.224 \text{ kN})$

$$\begin{aligned} \text{Axial Cutting Force } F_T &= 40\% F_c \text{ Ref 1} \\ F_T &= 200 \text{ lbf} (0.89 \text{ kN}) \end{aligned}$$

Radial Cutting Force R' Ref. 2 for single point operations lies between $0.08 F_c$ and $0.45 F_c$ but in the case of ceramic tooling R' lies between $0.7 F_c$ and $1.1 F_c$ it is therefore assumed that R' is equal to F_c

$$R' = 500 \text{ lbf} (2.224 \text{ kN})$$

$$\text{Resultant Cutting Force } R = 735 \text{ lbf} (3.27 \text{ kN})$$

D2.2. Headstock Motor Rating

$$\begin{aligned} \text{HP required at tool} &= \frac{CS \times F_c}{33,000} \\ &= \frac{350 \times 500}{33,000} \end{aligned}$$

$$\text{HP required at tool} = 5.3 (3.95 \text{ kW})$$

The efficiency of a conventional lathe with a geared drive decreases as speed increases, only 50% of the input power being available at the tool at a 1,000 rev/min. A headstock motor and bearings have been selected for the machine which improves efficiency considerably at high speeds Ref. Section D3. Therefore a 5.0 HP (3.72 kW) headstock motor has been chosen for the machine.

It is necessary to check what cuts can be taken when machining materials which have either

- (a) High 'P' values
- (b) High 'CS'

to ensure that sufficient power is

available at the tool.

The cutting speeds are those recommended for use with tungsten carbide tooling.

'f' was held constant at 0.010 inches/rev
(0.25 mm/rev)

- (a) Consider Ni Cr Steel En 36 :-
'P' = 500,000 lb/inch² (3.45 GN/m²)
'CS' = 200 ft/min (61 m/min)

For these conditions

$$\underline{d'} = 0.165 \text{ inches} (4.2 \text{ mm})$$

- (b) Take an aluminum alloy:-
'CS' = 1,000 ft/min (305 m/min)
'P' = 90,000 lb/inch² (0.62 GN/m²)

For these conditions

$$\underline{d'} = 0.185 \text{ inches} (4.7 \text{ mm})$$

D2.3 Specific Cutting Capacity.

The value is calculated for machining EN 1A with tungsten carbide tools at 75% efficiency.

$$\begin{aligned} \text{S.C.C.} &= \frac{d \times f \times CS \times 12}{\text{HP}} \times \frac{75}{100} \\ &= \frac{0.25 \times 0.01 \times 350 \times 12}{5} \times \frac{75}{100} \end{aligned}$$

$$\text{S.C.C.} = \underline{1.57 \text{ inches}^3/\text{hp min. (46 k.mm}^3/\text{kW min)}}$$

D2.4 Machine Capacity

$$\text{Machine Capacity} = \text{HP} \times \text{S.C.C. (at 75\% efficiency)}$$

$$= 5 \times 1.57$$

$$\underline{\text{Machine Capacity} = 7.85 \text{ in}^3/\text{min (172 k.mm}^3/\text{min)}}$$

D3. REPORT OF THE HEADSTOCK SUB-COMMITTEE.

Prepared by :- A.V. Terry
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D3.1 Introduction.

An important feature of the configuration proposed for the precision turning machine with a limited milling facility was a moving headstock. This implied that the headstock would need to be a compact, self-contained unit capable of moving with respect to the machine frame along a slideway with a very low coefficient of friction. To accommodate both precision turning and milling operations the headstock drive will need to provide a speed variable over a wide range.

This report describes the design procedure leading to the final headstock arrangement. A headstock assembly drawing is presented together with detail drawings of all major parts of the assembly. Also included in the report are calculations necessary for the headstock design concerning the hydrostatic spindle journal bearings and hydrostatic slideway bearings.

It was decided by the group board to offer a work-loading mechanism as an accessory to the machine. The basic requirements and an economic survey of this facility were investigated. The design is included in this report of a cheap but effective way of providing automatic work-loading on the machine.

D3.2. Basic Requirements.

Headstock Drive.

The headstock drive was to be capable of providing rotation at a controlled rate to facilitate, for example, milling a helix, and it was desirable for it to have a continuously variable speed within the desired speed range. It had to be capable of indexing with high accuracy to any position in 360°, with provision for positive locking in this position.

Chuck.

On automatic machines with high initial capital costs it is essential to aim for the maximum machine utilisation, which can be aided by power-work holding. Further, to take full advantage of the high metal removal rates possible with modern tool technology, it becomes desirable to use a power-operated chuck to obtain the higher horsepowers needed on lathe spindles. The requirement, therefore, was for a fully automatic chuck controlled by a numerical control system but with provision for manual override.

The gripping force of the chuck had to be controllable to accommodate the wide variety of components that the machine would be able to handle.

Thereby, in the case of thin components which were liable to be distorted by the high gripping forces necessary for heavy roughing cuts, the gripping force could be reduced to enable a finishing cut to be taken without removing the component from the chuck. Also, the consistency of gripping force possible with power-operated chucks would promote accuracy.

It was decided to aim for a chuck with a brief specification as follows:-

(a) A 4 inch (100m.m.) diameter work capacity. This value was derived directly from the Technical Survey results, but to accommodate small castings and up to 8 inches (200m.m.) diameter short work, it was envisaged to fix these items to a "face plate pallet" fitted with a central boss which could be gripped in the chuck.

(b) A minimum overall diameter preferably less than 12 inches (300 m.m.). This was considered to be the minimum dimension possible from an initial survey of chucks capable of holding 4 in. (100 m.m.) diameter workpieces.

(c) Capable of holding the workpiece concentric with the axis of the machine to within 0.0005 in. (0.012 m.m) maximum, which was considered from past experience to be the order of magnitude for precision work.

It was essential that the design of the headstock resulted in a compact unit of minimum size in order to keep overall machine size to a minimum, and to be aesthetically acceptable.

Spindle.

It would be necessary to have a hole down the centre of the spindle to accommodate the draw bar for power-chucking, but provided the diameter of hole was of the order of half the diameter of the spindle the reduction in spindle stiffness could be ignored, see Fig D3/2/1.

To determine the range of spindle speed required, examples of what were felt to be the extreme conditions likely to be encountered by the machine were examined.

Maximum rev/min: Consider taking a light single point finishing cut on 0.5 in. (12.7 m.m) o.d. Aluminium at a cutting speed of 2000 ft/min. (610 m/min with a Tungsten Carbide cutting tool.

$$\text{Rotational Speed} = \frac{\text{cutting speed}}{\text{circumference}} = \frac{2000 \times 12}{0.5 \times \pi} = 15000 \text{ rev/min.}$$

This was an extremely high speed and it was thought that some compromise on this figure would be necessary.

Minimum rev/min: This must tend to zero for the cutting of a helix with a milling cutter, for example, since the rotation of the chuck will be linked with the feed rate. Consider, however, screwing an 8 in. (200 m.m.) o.d. cast iron casting (Brinell 230) at a cutting speed of 6 ft/min (1.8 m/min) with a high speed steel cutting tool.

$$\begin{array}{l} \text{Rotational} \\ \text{Speed} \end{array} = \frac{6 \times 12}{8 \times} \quad 3 \text{ rev/min.}$$

From the above calculations it appeared that a continuously variable speed from 0 - 15000 rev/min was required. A calculation based on mean work diameter of a mild steel workpiece using carbide cutting tools resulted in a spindle speed as follows:

$$\begin{array}{l} \text{Rotational} \\ \text{Speed} \end{array} = \frac{400 \times 12}{4 \times} \quad 400 \text{ rev/min.}$$

A frequency distribution diagram would result in the form shown in Fig D3/2/2. Therefore, summarising, a continuously variable speed range of 0 - 3000 rev/min would enter for most combinations of work and tool material.

Work Loading.

The requirement was for an attachment which would load and unload the machine automatically and would perform the operation as quickly as possible. The attachment should be operated by one command from the tape input and then continue by means of an internal actuation system. This could be in the form of a cam, rotating drum with stops, or a rotary hydraulic valve. The attachment should be as compact and versatile as possible accepting workpieces up to 8 in. (200 m.m.) long and a 4 in. (100 m.m.) diameter to give the machine good job capacity.

Since the headstock already had a controlled movement, the work-loading attachment should be stationary, hence requiring the chuck to move up to accept the workpiece.

D3.3 Survey.

Headstock Drive.

It was considered that change-speed gearbox and cone pulley drives would not be adequate and it was therefore proposed to investigate drives eliminating the disadvantages of these types. In addition following Configuration Committee Report PE/CF/1/9 recommending a moving headstock, a compact headstock which did not involve direct mechanical link with the driving motor was required. These requirements could be met by hydraulic motors.

Hydraulic power in the operation of machine tools offers continuously variable speed and considerable power. A difficulty arises in obtaining a chuck capable of holding work with sufficient grip to be able to take advantage of the power which is available. Disadvantages appear with hydraulic motors in that as the efficiency is the combined one of a pump and motor it is generally less than a mechanical drive, and if the hydraulic motor is expected to give full power over a wide speed range it may become

very heavy and costly. Further, oil viscosity varies with temperature and may cause fluctuations in speed, although compensating devices can be introduced to deal with this problem. Rapid and shockless reversing is possible and relief valves provide a safety factor. Changes in spindle speed can be effected either manually or by numerical tape control and the superiority of the drive is enhanced by the fact that speed variation can be made while cutting. Thus while using cemented-carbide tools a cut could be started at a speed which would not damage the tool, and then with the tool in full cut the speed could be set to a maximum.

An alternative to a hydraulic drive was an electro-magnetic induction coupling such as the TASC (Torque and Speed Control) unit. This could be used to provide the lathe requirement for falling torque-constant horsepower as speed rises. A TASC torque system allows manual control of the torque from the TASC unit output shaft. The speed will rise until the torque demanded by the load rises to that set manually on by the tape. The rated output torque is typically quoted in the range 0-2500 rpm, and higher speeds would involve using a gearbox.

Spindle Design.

In the design of machine tool spindles, the requirements for high stiffness, smoothness of sliding and thermal distortion conflict with each other.

Deformation of the spindle contributes typically about two-thirds of the flexibility at the chuck even with the best layout. The stiffness of the chuck is related to the fourth power of the diameter of the spindle between the bearings and it is therefore important to select a spindle diameter which strikes the right balance between the requirements for high stiffness and the avoidance of thermal distortion. The diameter of the spindle is usually limited by the diameter of the bearings which must be kept within certain limits to avoid high temperatures.

The choice of bearings for the spindle lies between:-

- (1) rolling element
- (2) hydrostatic

Rolling element bearings are readily available and offer relative cheapness but are limited in performance in comparison with hydrostatic bearings.

Hydrostatic bearings have very low friction properties. The use of oil hydrostatic bearings is growing rapidly. In these bearings a finite thickness of oil exists round the journal at all times even when the spindle is not rotating. Wider speed ranges are attainable with these bearings than with rolling element bearings within given temperature limits. Air hydrostatic (aerostatic) bearings run very cool compared with oil hydrostatic bearings but are generally used for light loads and very high speeds - up to 10,000 - 20,000 rev/min. An advantage of aerostatic bearings is that the working fluid is clear and it need not be recirculated, so that the problem of excluding abrasive dust is much simpler. They are not suitable, however, for general use in larger bearings.

In the choice of bearings for the spindle, hydrostatic were preferred to rolling element for their greater performance, and oil hydrostatic bearings appeared more practicable than aerostatic bearings in this application.

The Machine Tool Industries Research Association (M.T.I.R.A.) have produced non-dimensional graphs in order to simplify the calculations for the optimum spacing of bearings for machine tool spindles. Analysis shows that it is very difficult to improve by more than 1/6 upon the stiffness of an optimum two-bearing system by the addition of a third bearing. The spindle arrangement would therefore be a two-bearing system and it was envisaged providing a motor whose bearings were integral with the main spindle bearings of the machine.

Types of Chucks.

The vast majority of power operated chucks in use are operated by a direct mechanical method, but with the greater demands of numerically controlled machines the operation of more recently developed power chucks can be classified into three basic types:

- (1) pneumatic.
- (2) hydraulic
- (3) electric

Of these three the most common are pneumatic and hydraulic, owing to the greater use nowadays of pneumatics and hydraulics for general power on machine tools. Assuming both sources of power are available on the machine, the decision to use pneumatic or hydraulic operation normally depends on a combination of several factors. From the cost viewpoint, pneumatic operation appears to be cheaper than hydraulic operation with chucks under 16 in. (400 mm) in diameter.

The most popular arrangement consists of a cylinder mounted on the tail of the lathe spindle and rotating with it, pneumatic or hydraulic power being supplied to the cylinder from a distributor. Cylinders for pneumatic operation are usually made of aluminium to reduce the moment of inertia, while those for hydraulic operation are generally made of steel to cater for the higher pressures involved. Normally the cylinder piston is connected to the chuck by an operating draw-bar.

Pneumatic or hydraulic operation can also be provided by a "front-end chuck", which incorporates a cylinder built into the back of the chuck. This arrangement has the advantage of a simpler installation without the need for a draw-bar, but increases overhang and loads on the front spindle bearing. The most commonly employed method of supplying power with this arrangement is to use an annular ring either on the rear face of the chuck or on its periphery.

Electrical operation usually consists of a reversible electrical motor which drives, through a reduction gearbox and a preset friction clutch, a nut which operates a draw-bar. The motor and reduction gear are mounted on and rotate with the spindle, power being brought to the motor by means of conventional slip rings.

The mode of operation of power chucks is either by a system of levers or by wedge action to close the jaws concentrically.

Lever-type chucks provide the best compromise between a high mechanical advantage to provide a good gripping force, and adequate jaw movement to accommodate tolerances on the gripping face of the component. Chucks operating on this principle can be provided with different mechanical advantages, depending on the proportions of the lever.

The wedge-type of chuck is much simpler than the lever-type as there are fewer moving parts, but has a slightly lower efficiency and a mechanical advantage somewhere in the middle of the lever-type chuck range. However, its loss of efficiency is more than balanced by the increased ruggedness inherent in its construction, allowing higher draw-bar pulls to be utilised, with a consequent increase in life.

To obtain high speeds it is essential to minimise the moment of inertia of the complete chuck thereby reducing the power required to accelerate and decelerate the spindle and workpiece, and to minimise the effect of centrifugal forces on the jaws, to overcome consequent loss of gripping power. Composite chucks, in which the stiffness of the body is provided by lightweight aluminium and only the load-bearing jaw-ways are in steel, meet these requirements. With these lightweight high speed chucks which generally have wedge action, power requirements and loading on brakes, clutches and transmission are decreased.

Apart from the mechanical properties and dimensions of the components concerned there were obviously many factors which affected the choice of the correct chuck, so that it became difficult to specify a particular type from the wide range of power-operated chucks available, as being most suitable for our needs. It was felt that expert advice was essential and therefore the choice was made following a visit and consultation with F. Pratt and Co. Ltd., of Halifax, who are specialist chuck manufacturers.

Work Loading.

The study of group technology for components has caused the classification of work pieces. The use of a row of digits to relate to the overall dimensions and machining sequence means, that for a machine such as the one being considered, a standard range of blanks can be used. This eases the problem of work loading since a standard range of clamps can be used and matched to the batches to be produced.

The automatic work loading of the machine being designed in this project was considered to be desirable by the group board, but the work-loading attachment should be included as an accessory and not as an integral part of the machine.

The economies of work loading fall into several categories. For small components automatic work loading will speed up production by reducing idle time as compared with manually fed machines.

For a co-ordinated production line where one part is passed successively from machine to machine, all machines have to be timed to run to the same time cycle as the slowest machine. Thus by using work loading to speed up the slowest machine the production rate could be increased.

On a hydraulically operated machine the work loading device can be operated from the main circuit as it will only be used when the rest of the machine is at rest. This eliminates the need for a separate power supply to drive the work loading equipment. However, although the work loading is of a sequential operation type and is hence self contained, an initiating control from the main control system would be required to start the sequence.

D3.4. Design Solution.

Headstock Assembly.

The survey into headstock drives revealed that a hydraulic motor meeting the design requirements almost exactly had been developed by the National Engineering Laboratory (N.E.L.) for application to machine tool drives. This motor was of the axial piston variable displacement type. Speed control was achieved by varying the swash plate angle and in conjunction with a variable displacement pump a speed capability up to 2000 rev/min. was available. The headstock motor was based on this design, Ref. Headstock Assembly Drawing No. D3/4/3, and by sealing techniques it was considered that this smaller motor would be capable of operation up to 3000 rev/min to meet the project specification. The motor shaft has been extended to form the machine spindle and is supported by two oil hydrostatic journal bearings. The front bearing is of double conical design and will take both radial and axial loads, while the rear bearing is of parallel design and will resist radial loads only. The conical bearing consists of two rows of six equally spaced pads of recessed design set at 30° to the spindle axis, and the parallel bearing consists of one row of six equally spaced recessed pads. A six pad configuration was selected since with a smaller number the load capacity of the bearing would have been dependent to some extent on the direction of load application. Calculations were carried out to evaluate the design parameters of the spindle bearings, Ref. D3.5 Appendix.

A hydraulic motor of the piston type has the ability to act as a very effective brake and can stop a machine spindle from high speed almost instantaneously. Therefore the braking of the spindle and chuck is carried out by the hydraulic motor and the function of the disc brake is to hold the spindle rigid against the cutting tongue applied by an end mill or drill.

Following information given by N.E.L., no temperature problems are envisaged with the headstock since a similar headstock motor running for an 8 hour period at N.E.L. experienced only an external 3°C rise above an ambient temperature of 28°C . It was therefore apparent that an efficient air cooler, Ref. Control Systems Final Report, would keep the headstock at a reasonable temperature.

The headstock motor casing has been designed to incorporate the hydrostatic slideways. So that it will be sufficiently rigid this casting will be cast from Grade 17 Cast Iron. Calculations for the hydrostatic slideway bearings have been evaluated for a three pad configuration to give good resistance to tilt by opposing movement forces, Ref. D3.5 Appendix. Provision on the casing has been made for securing the actuation cylinders which move the headstock, the linear measuring system and the digital revs which control the motor speed.

Fixed to the rear of the motor casing by a flange and bolts is a casting also cast from Grade 17 Cast Iron housing the rotary encoder, Ref. Control System Final Report, and the disc brake, Ref. Drawing No. D3/4/4. This casing incorporates an outrigger roller bearing housing, the function of the bearing being to prevent 'whip' in the spindle caused by the overhang from the headstock motor bearings. A removable cover in the casing gives easy access to the rotary encoder for assembly and inspection purposes and to the disc brake for maintenance. In addition the cover provides a sealed exit for the oil supply lines from the motor and the disc brake, and the air supply lines from the encoder.

The chucking requirements were met by an 8 in. (200 m.m.) dia. three-jaw high-speed power chuck, Ref. Drawing No. D3/4/5, manufactured by F. Pratt & Co. Ltd., (Halifax). This is a lightweight composite chuck having an aluminium body fitted with steel jaw-ways carrying hard top jaws. The gripping operation is by wedge action effected by a draw bar having a stroke of 0.937 in. (23.8 m.m.) and a maximum pull of 6000 lbf (2721 kg) provided by a 4 in (100 m.m) bore rotating hydraulic cylinder mounted on the rear end of the spindle. Oil is supplied to the cylinder by means of a two way rotating hydraulic distributor, Ref. Drawing No. D3/4/6. A maximum static grip per jaw of 3680 lbf (1670 k.g.) is available at 6000 lbf (2721)kg draw bar pull. The arrangement operates up to a maximum recommended speed of 3100 rev/min. with a consequent 15 - 20% loss of maximum static grip.

Work Loading Mechanism.

The work-loading mechanism, Ref. Drawing No. D3/4/7, will be offered as an accessory to the machine. It consists essentially of the following three units:-

- 1) Loading drum
- 2) Transfer channel
- 3) Transfer arm

The loading drum, Ref. Drawing No. D3/4/8, has been designed to be compact and capable of accepting any diameter or length of workpiece within the specification of the machine. It is a simple sequential unit, operated pneumatically from the main air line, advancing the drum through a ratchet and pawl mechanism. It will be constructed from welded steel plate to minimise cost. The unit accommodates 20 components, but can be lengthened by

approximately 4 in (100 mm.) for each additional two components. A capacity of 20 components was chosen so that the unit could be free standing vertically alongside the machine. In addition the type of work for which the machine is designed will have a fairly long floor to floor time and consequently the work loading drum should only have to be refilled once every hour on average. Provision is made for reloading through the open side of the drum before it is completely empty. Maintenance will be limited to routine greasing of the bearings and pivot pins.

The transfer channel has a simple vee-configuration accepting billets 1 - 4 in (25 - 100 mm.) diameter. It is attached to the loading drum and transports the components to the transfer arm.

The transfer arm, Ref. Drawing No. D3/4/9, consists of a transfer ram, adjustable transfer channel, and incorporates an overload device in the head. The transfer ram is a pneumatic type having a stroke of 15 in. (380 mm). This will be attached to the machine and will offer the workpiece up to the chuck through an aperture in the machine side. The transfer channel is provided with adjustment so that each time the batch is changed the unit can be quickly readjusted. A retractable stop compensates for changes in workpiece length, whilst there is an adjustment vertically to allow different workpiece diameters to be positioned exactly on the chuck centre-line with the transfer ram extended. The overload device in the head provides a means of switching off the headstock motion when the component has been properly located in the chuck. It is a simple compression spring type unit with a pneumatic transducer giving the cut-off point.

D3.5. Appendix - Headstock Design Calculations.

Nomenclature.

- AcW = Load capacity of headstock conical bearing in an axial plane
lbf (k.N)
- Ae = effective area of hydrostatic pads inch^2 (mm^2)
- b = pad oil escape perimeter length inch (mm)
- D = diameter of capillary or restrictor inch (mm)
- F = Force exerted by disc brake piston lbf (k.N.)
- h = actual gap between pad and face inch (mm)
- h_o = equivalent gap between pad and face at which
R = r inch (mm)
- l = length of capillary or restrictor inch (mm)
- l₁ = Pad land width inches (c.m)
- LcW = Radial load on Headstock conical bearing lbf (kN)
- LpW = Radial load on Headstock parallel bearing lbf (kN)
- Pc = Cell Pressure lb/inch² (M.N/m²)
- Ps = Supply Pressure lb/inch² (M.N/m²)
- Q = Oil flow inches³/s (mm³/s)
- Q_H = Oil flow from Headstock Journal Bearings inches³/s (mm³/s)
- R = resistance to oil flow of pad lands lbs/inch⁵ (N.S/mm⁵)
- r = resistance to oil flow of capillary or restrictor
lb s/inch⁵ (N.S/mm⁵)
- RcW = Load capacity of headstock conical bearing in a radial plane
lbf (k.N)
- Ri = Cell Radius inch (m.m)
- Ro = Pad outside radius inch (mm)
- RpW = Load capacity of headstock parallel bearing in radial plane
lbf (k.N)

- R_R = Radial Resultant cutting force lbf (k.N)
- S = Stiffness of Pad or Bearing lb/inch (N/m.m)
- T = Thrust from Pad or Bearing lbf (k.N)
- W = Maximum load which can be applied to headstock journal bearings
lbf (k.N)
- μ = viscosity of hydrostatic bearings oil reyns lbs/inch²
(N.s/m²)
- f_o = coefficient of friction between disc brake pad and disc

Hydrostatic Spindle Journal Bearings.

Oil is supplied to the bearings at $1000 \text{ lb/in}^2 (6.9 \text{ MN/m}^2)$

Consider each bearing row as 6 circular pads of recessed design equally spaced around a circumference. Each pad is separated from the next by recess in line with the bearing axis Ref. Fig D3/5/10. These recesses have been found to increase the stiffness of the bearing.

Consider a Single Pad:-

$$\begin{aligned} \text{Let } h_o &= 0.0006 \text{ in } (15.2 \mu\text{m}) \\ \therefore h_o &= 0.0003 \text{ in } (7.6 \mu\text{m}) \\ \text{Let } R_o &= 0.9 \text{ in } (22.8 \text{ mm}) \\ \text{and } R_i &= 0.6 \text{ in } (15.2 \text{ mm}) \end{aligned}$$

$$\begin{aligned} \text{then } A_c &= \frac{(R_o^2 - R_i^2)}{2 \ln R_o/R_i} \\ A_c &= \frac{(0.9^2 - 0.6^2)}{2 \ln 1.5} \\ A_c &= \underline{1.74 \text{ in}^2 (1120 \text{ mm}^2)} \end{aligned}$$

Stiffness of one Pad:

$$h = h_o = 0.0003 \text{ inches } (7.6 \mu\text{m}) \text{ for best operating gap}$$

$$S = \frac{3 A_c P_s (h^2/h_o^3)}{(1 + h^3/h_o^3)^2}$$

$$S = \frac{3 \times 1.74 \times 1000 \times 10^4}{4 \times 3}$$

$$S = \underline{4.35 \times 10^6 \text{ lb/in } (0.77 \text{ N/mm})}$$

The maximum stiffness of a pad occurs when $h_o^3 = 2h^3$

$$\therefore h \text{ for maximum stiffness} = 0.000238 \text{ inches } (6.0 \mu\text{m})$$

$$\therefore \underline{S_{\text{MAX}}} = 5 \times 10^6 \text{ lb/in } (0.885 \text{ N/mm})$$

Maximum Stiffness of Double Conical Bearing in Radial Plane:

The conical bearing consists of 2 rows of 6 equally spaced pads set at 30° to the spindle axis.

By taking components in plane of load W Ref. Fig D3/5/10

$$\text{Radial } (S_1 + S_6) = 2 \times 2 \times 4.35 \times 10^6 \times \text{Cos } 30 \text{ deg.}$$

$$\text{Radial } (S_4 + S_3 + S_2 + S_5) = 4 \times 2 \times 4.35 \times 10^6 \times \text{Cos } 30 \text{ deg} \times \text{Sin } 30 \text{ deg.}$$

$$\begin{aligned} \text{Maximum Radial Stiffness} &= \text{Sum of Radial Stiffness Components} \\ &= \underline{30.2 \times 10^6 \text{ lb/in. (5.35 N/mm.)}} \end{aligned}$$

Maximum Stiffness of Double Conical Bearing in Axial Plane;

12 Pads will contribute to Axial Stiffness in either direction.

$$\begin{aligned} \therefore \text{Maximum Axial Stiffness} &= 12 \times 4.35 \times 10^6 \times \text{Sin } 30 \text{ Deg.} \\ &= \underline{26 \times 10^6 \text{ lb/in. (4.6 N/mm)}} \end{aligned}$$

Maximum Stiffness of Parallel Bearing in Radial Plane:

The parallel bearing consists of 1 row of 6 equally spaced pads.

$$\begin{aligned} \text{Maximum Radial Stiffness} &= 2 (S_1 + S_2 \text{ Sin } 30 \text{ deg.} + S_4 \text{ Sin } 30 \text{ deg.}) \\ &= \underline{17.4 \times 10^6 \text{ lb/in. (3.0 N/mm.)}} \end{aligned}$$

Maximum Load Capacity of Hydrostatic Spindle Bearings:

The maximum load permissible will be that load (W) which occurs just before metal to metal contact in the bearing. Ref. FIG D3/5/10.

$$T = \frac{Ac Ps}{1 + \left(\frac{h}{h_0}\right)^3}$$

$$W = (T_5 - T_1) + (T_6 - T_2) \text{Cos } 60 \text{ deg.} + (T_4 - T_3) \text{Cos } 60 \text{ deg.}$$

$$W = Ac Ps \left[\frac{1}{\left\{ \frac{1 + (h_0 - x)}{h_0} \right\}^3} - \frac{1}{1 + \left(\frac{h_0 - x}{h_0}\right)^3} + 2 \text{Cos } 60 \text{ deg.} \right] X$$

$$X \frac{1}{1 + \left(\frac{h_o - x/2}{h_o}\right)^3} = \frac{1}{1 + \left(\frac{h_o - x/2}{h_o}\right)^3}$$

At point of metal to metal contact in bearing $h_o = x$

$$\therefore \underline{W = A_c P_s(1.549)}$$

Maximum Load Capacity of Conical Bearing in Radial Plane:

$$R_c W_{MAX} = \cos 30 \text{ deg. } A_c P_s(1.549) \times 2 \quad \text{since two rows of conical bearings.}$$

$$R_c W_{MAX} = \cos 30 \text{ deg. } \times 1.74 \times 1000 \times 1.549 \times 2$$

$$\underline{R_c W_{MAX} = 4,660 \text{ lbf (20.7 kN)}}$$

Maximum Load capacity of Conical Bearing in Axial Plane:

$$A_c W_{MAX} = \cos 60 \text{ deg. } A_c P_s (1.549)$$

$$\underline{A_c W_{MAX} = 1,350 \text{ lbf (6.0 k.N)}}$$

Maximum Load Capacity of Parallel Bearing in Radial Plane:

$$R_p W_{MAX} = A_c P_s(1.549)$$

$$\underline{R_p W_{MAX} = 2,700 \text{ lbf (12.0 k.N)}}$$

Maximum Load to which Headstock Spindle Bearings will be submitted :
Ref. FIG. D3/5/11

Calculations are carried out neglecting any load carrying capacity of the outrigger bearing which is present only to prevent whip in the spindle.

Turning forces are considered in the calculations as they are much larger than the milling or drilling forces.

The 30 deg slope of the machine bed is not taken into account because the resulting weight component of the headstock will oppose the axial cutting force.

For evaluation the turning cutting forces have been split into an axial force of 200 lbf (0.89 kN) and a force R_T of 707 lbf (3.14 kN) which combines the tangential and radial cutting forces Ref. Section D2.

For the worst condition the assumption is made that these forces are acting at the end of an 8 in (203 mm) length workpiece of 4 in. (101 mm) diameter. The tailstock centre is taken not to be in use to support any of the load.

The weight of the units comprising the headstock was estimated and includes oil and weight of spindle for that portion of its length.

Radial Reactions at the Parallel and Conical Spindle Bearings:

$$(500 \times 26) + (3 \times 12) + (20 \times 6) + (11 \times 45) - (14 \times 20) - (22 \times 25) - (200 \times 5) = 10L_c W$$

$$\underline{L_c W = 1,182 \text{ lbf (5.25 kN)}}$$

$$\underline{L_p W = 1,004 \text{ lbf (4.47 kN)}}$$

Consider force R_R acting on spindle bearings:-

$$707 \times 26 = 10 L_c W$$

$$\therefore \underline{L_c W = 1,838 \text{ lbf (8.18 kN)}}$$

$$\underline{L_p W = 1,131 \text{ lbf (5.03 kN)}}$$

Axial Reactions at the Conical Spindle Bearing:-

The maximum axial force due to cutting will be 200 lbf (0.89 kN) Ref. Section D2. However if the tailstock centre is in use it could apply a force of 400 lbf (1.78 kN) Ref. Appendix D4. When giving maximum support to a workpiece. Therefore the maximum total axial force was considered to be:

$$\underline{600 \text{ lbf (2.67 kN)}}$$

TABLE I.

Comparison of maximum force which can be applied under the worst conditions and the load capacity of the headstock spindle bearings.

	Load Capacity of Bearing lbf (k.N)	Maximum load applied lbf (k.N)
Double Conical Bearing in radial plane	4,660 (20.7)	1,838 (8.18)
Double Conical Bearing in Axial Plane	1,350 (6.0)	600 (2.67)
Parallel Bearing in Radial Plane	2,700 (12.0)	1,131 (5.03)

In each case the bearings have an operating safety factor of at least 2. This will allow for a drop in the 1000 lb/inch² (6.9 MN/m²) oil supply due to say a partially blocked filter. Also it will protect the bearings against overload conditions or misuse.

Hydrostatic Slideway Bearings:-

Each bearing face has hydrostatic pads as shown in FIG D3/5/12

Maximum Loads to which Headstock Slideway Bearings will be subjected:-

The cutting tongue will be opposed at faces A, B and E. Taking minimum rev/min. when cutting as 100

$$\begin{aligned} \text{Max Torque} &= \frac{HP \times 33,000}{2 \pi N_{\min}} \\ &= \frac{5 \times 33,000}{2 \pi \times 100} \\ &= 262 \text{ lbf. ft (355 N.m)} \end{aligned}$$

Now faces A, B and E are at a mean radius of 6.5 inches (16.5 cm) so,

Face A must oppose a force of 124 lbf (167 Nm)
 Face B " " " " " " "
 Face E " " " " " 24.7 lbf (335 N.m)

Consider force R_R which caused maximum reactions in headstock spindle bearings,

$$\begin{aligned} LcW &= 1,838 \text{ lbf (8.18 k.N)} \\ LpW &= 1,131 \text{ lbf (5.03 k.N)} \end{aligned}$$

Now LcW and LpW act in opposite directions therefore the worst condition is taken and it is assumed that the headstock tips about the middle pad. It was therefore assumed that the middle pad does not contribute to either the stiffness or load capacity of the slideway bearings. Calculations were then referred to the centres of the two end pads.

Maximum loads on slideway bearings will be, LcW and LpW.
 These loads will be :

+	loading due to cutting torque of 124 lbf(167 Nm) at face A
+	" " " " " " " " " " " B
+	" " " " " " " " 24.7 lbf(335 Nm) " " E
-	" " " " " " " " " " " C
-	" " " " " " " " " " " D

Maximum possible loading LcW will be opposed by faces A,B and D.

$$\begin{aligned} \text{Faces A and B will oppose } & \frac{8}{13} LcW + 247 \\ & = \underline{1,379 \text{ lbf}} \quad (6.12 \text{ kN}) \end{aligned}$$

$$\begin{aligned} \text{Face D will oppose } & \frac{5}{13} LcW - 247 \\ & = \underline{459 \text{ lbf}} \quad (2.04 \text{ kN}) \end{aligned}$$

Consider load LpW , this will be opposed by face C and E

$$\begin{aligned} \text{Face C opposes } & \frac{8}{13} LpW - 247 \\ & = \underline{453 \text{ lbf}} \quad (2.03 \text{ kN}) \end{aligned}$$

$$\begin{aligned} \text{Face E opposes } & \frac{5}{13} LpW + 247 \\ & = \underline{678 \text{ lbf}} \quad (3.01 \text{ kN}) \end{aligned}$$

Maximum load capacity of headstock slideway bearings:

The maximum load permissible will be that load which occurs just before metal to metal contact as for the hydrostatic spindle bearings.

$$\text{oil supply } 1000 \text{ lb/in}^2 \quad (6.9 \text{ MN/m}^2)$$

$$h_o = 0.0005 \text{ in} \quad (12.7 \mu\text{m})$$

Minimum gap possible at centre of end pads Ref. FIG D3/5/13

$$\begin{aligned} h_1 &= \frac{0.001 \times 2.25}{12} \\ &= \underline{0.00019 \text{ in}} \quad (5.0 \mu\text{m}) \end{aligned}$$

$$\begin{aligned} h_2 &= \frac{0.001 \times 9.75}{12} \\ &= \underline{0.00081 \text{ in}} \quad (20.3 \mu\text{m}) \end{aligned}$$

A_c for faces A,B, D and E

$$= \underline{3.655 \text{ in.}^2} \quad (2,360 \text{ mm}^2)$$

As for face C

$$= \underline{6.2 \text{ in.}^2} (4,000 \text{ mm}^2)$$

$$\begin{aligned} \text{For faces A,B,D and E } (T_1 - T_2) &= 3,360 - 415 \\ &= \underline{2,945 \text{ lbf}} (13.1 \text{ kN}) \end{aligned}$$

$$\begin{aligned} \text{For face C } (T_1 - T_2) &= 5,770 - 710 \\ &= \underline{5,060 \text{ lbf}} (22.5 \text{ k.N}) \end{aligned}$$

Taking force components in the plane of maximum loading will determine if the slideway bearings have sufficient capacity.

Maximum load which can be carried by faces A and B

$$\begin{aligned} &= \sin 75 \text{ deg } (TA) + \sin 45 \text{ deg } (TB) \\ &= \sin 75 \text{ deg } (2945) + \sin 45 \text{ deg } (2945) \\ &= \underline{5,115 \text{ lbf}} (22.7 \text{ kN}) \end{aligned}$$

Maximum load which can be carried by face D

$$\begin{aligned} &= \sin 75 \text{ deg } (TD) \\ &= \underline{2,845 \text{ lbf}} (12.6 \text{ kN}) \end{aligned}$$

Maximum load which can be carried by face C

$$\begin{aligned} &= \sin 75 \text{ deg } (TC) \\ &= \underline{4,900 \text{ lbf}} (21.8 \text{ kN}) \end{aligned}$$

Maximum load which can be carried by face E

$$\begin{aligned} &= \sin 75 \text{ deg } (TE) \\ &= \underline{2,845 \text{ lbf}} (12.6 \text{ kN}) \end{aligned}$$

TABLE II

Comparison of the maximum load which can be applied under the worst conditions and the load capacity of the headstock slideway bearings considered in the plane of maximum load.

	Load Capacity of Bearing		Maximum load applied	
	lbf	(k.N)	lbf	(k.N)
Face A)	5,115	(22.7)	1,379	(6.12)
Face B)				
Face C	4,900	(21.8)	453	(2.03)
Face D	2,845	(12.6)	459	(2.04)
Face E	2,845	(12.6)	678	(3.01)

It can be seen that the minimum operating safety factor is 4.

Maximum Stiffness of the Headstock Slideways Hydrostatic Bearings in the plane of maximum loading:

Stiffness of the bearing faces A B D and E

$$\frac{S}{h} = \frac{5 \times 10^6 \text{ lb/in}}{0.0005 \text{ in}} \quad (0.885 \text{ N/mm.}) \quad \text{at} \quad (12.7 \mu\text{m})$$

Maximum stiffness occurs when $h_o^3 = 2h^3$
i.e. at a gap of 0.000397 inches (9.0 μm)

$$S_{MAX} = 5.6 \times 10^6 \text{ lb/inch} \quad (0.99 \text{ N/mm})$$

Stiffness of bearing face C

$$\frac{S}{h} = \frac{7.0 \times 10^6 \text{ lb/inch}}{0.0005 \text{ inches}} \quad (1.24 \text{ N/mm}) \quad \text{at} \quad (12.7 \mu\text{m})$$

$$S_{MAX} = 7.9 \times 10^6 \text{ lb/inch} \quad (1.40 \text{ N/mm})$$

Maximum Stiffness of headstock slideway bearings.

= Sum of component stiffnesses.

$$= \sin 75 \text{ deg } S_A + \sin 45 \text{ deg } S_B + \sin 75 \text{ deg } S_D + \sin 75 \text{ deg } S_C + \sin 75 \text{ deg } S_E$$

$$= \sin 75 \text{ deg}(5.0 + 5.0 + 7.0 + 5.0) + \sin 45 \text{ deg} (5.0) \times 10^6$$

$$= 25.7 \times 10^6 \text{ lb/in} \quad (4.55 \text{ N/mm})$$

Therefore the loads to which the headstock slideway hydrostatic bearings will be subjected are well within their capabilities and a high degree of stiffness is maintained. The same situation applies to the cross-slide and tailstock hydrostatic bearings which are of similar design.

Size of Capillary or Restrictor for Headstock Spindle Bearings:

Maximum stiffness is achieved when $R = r$

$$\therefore P_c = P_s/2$$

$$\text{now } r = P_c/Q = \frac{128 \mu l}{D^4}$$

$$Q = \frac{P_c h^3}{6 \mu \ln R_o/R_i} \quad \text{_____ (1)}$$

Consider one pad and assume $l = 1'' (2.54 \text{ cm})$

$$\therefore D = \sqrt[4]{\frac{128 \mu l h^3}{6 \times 0.40547}}$$

$$D = 0.0082 \text{ inches } (0.208 \text{ mm})$$

If a standard capillary of 0.010 inches (0.254 mm) diameter is chosen then the length of the capillary = 7.0 inches (176 mm) since l varies as D^4 . It is not possible to fit a capillary 7.0 inches (176 mm) long into the bearings unless it is coiled. In practise a capillary of say 0.008" (0.203 mm) could be employed and the length adjusted to obtain the correct resistance and accommodate errors in diameter. Alternatively a larger diameter coiled capillary could be fitted which would be less susceptible to blockage.

Flow of oil from one pad in headstock spindle bearings:

$$\begin{aligned} \text{Ref equation (1)} \quad Q &= \frac{500 \times \pi \times 0.0003^3}{6 \times 38 \times 10^{-6} \times 0.40547} \\ Q &= 460 \times 10^{-6} \text{ ins}^3/\text{s} \quad (7.5 \text{ mm}^3/\text{s}) \end{aligned}$$

From the 18 pads which make up the headstock spindle bearings the flow of oil will be

$$\begin{aligned} Q_H &= 460 \times 10^{-6} \times 18 \\ Q_H &= 8.3 \times 10^{-3} \text{ ins}^3/\text{s} \quad (136 \text{ mm}^3/\text{s}) \end{aligned}$$

Flow of oil from Hydrostatic Slideway Bearings:

Consider one pad

$$Q = \frac{P_c \cdot b \cdot h^3}{12 \mu l_1}$$

$$Q = \frac{500 \times 0.0005^3 \times 7.5}{12 \times 38 \times 10^{-6} \times 0.375}$$

$$Q = 2.75 \times 10^{-3} \text{ in}^3/\text{s} \quad (45 \text{ mm}^3/\text{s})$$

From the headstock hydrostatic slideway bearings which consist of 15 pads

$$Q = 41.2 \times 10^{-3} \text{ in}^3/\text{s} \quad (675 \text{ mm}^3/\text{s})$$

For all the hydrostatic slideway bearings on the machine

$$Q = 0.125 \text{ inches}^3/\text{s} \quad (2,050 \text{ mm}^3/\text{s})$$

and for all hydrostatic bearings on the machine both journal and slideway

$$Q = 0.133 \text{ in}^3/\text{s} \quad (2,190 \text{ mm}^3/\text{s}) \quad \text{at}$$

1000 lb/inch² (6.9 M.N /in²) oil supply

Size of Capillary for Slideway Pads:

$$D = 4 \sqrt{\frac{Q \cdot 128 \mu l}{P_c \pi}}$$

Consider capillary to be 1 inch (2.54 cm) long.

$$D = 4 \sqrt{\frac{2.75 \times 10^{-3} \times 128 \times 10^{-6} \times 38}{500 \pi}}$$

$$D = 0.0096 \text{ inches} \quad (0.244 \text{ mm})$$

If a standard 0.010 inches (0.254 mm) diameter capillary is used then l becomes 1.175 inches (44.5 mm).

Disc Brake.

The function of the disc brake is to hold the spindle rigid against the torque applied by an end mill or drill. The end mill or drill is driven through a 1:1 planetary gear by a gear pump motor situated on the cross-slide of 3 HP (2.20 kW) rating with a minimum speed of 500 rev/min.

For friction pads μ_o varies between 0.30 and 0.45 and is unaffected up to 400 deg C.

Maximum Torque that can be applied to the spindle by cross-slide gear motor:-

$$\begin{aligned}\text{Torque} &= \frac{\text{HP} \times 33,000}{2\pi N_{\text{MIN}}} \\ &= \frac{3 \times 33,000}{2\pi \times 500} \\ &= \underline{31.5 \text{ lbf.ft}} \quad (42.7 \text{ N.m})\end{aligned}$$

Disc brake operating pressure is 1,000 lbf/inch² (6.9 M.N/m²)

Disc Brake acts at a radius of 2.4 inches (61 mm)

Diameter of disc brake operating cylinder is 1.25 inches (38 mm)

Assume μ_o at lowest value of 0.3

$$\text{Maximum drag at mean radius} = 2.F.\mu_o$$

Maximum torque which can be resisted by disc brake

$$\begin{aligned}&= \frac{1,000 \times \pi \times 1.25^2 \times 0.3 \times 2.4}{4 \times 12} \\ &= \underline{147 \text{ lbf.ft}} \quad (200 \text{ N.m})\end{aligned}$$

Hence the disc brake is adequate for this application with an operating factor (147/31.5) of over 4. The high operating factor will provide rigidity against fluctuating loads as the number of teeth cutting varies and also prevent torsional vibrations.

D3/6 References.

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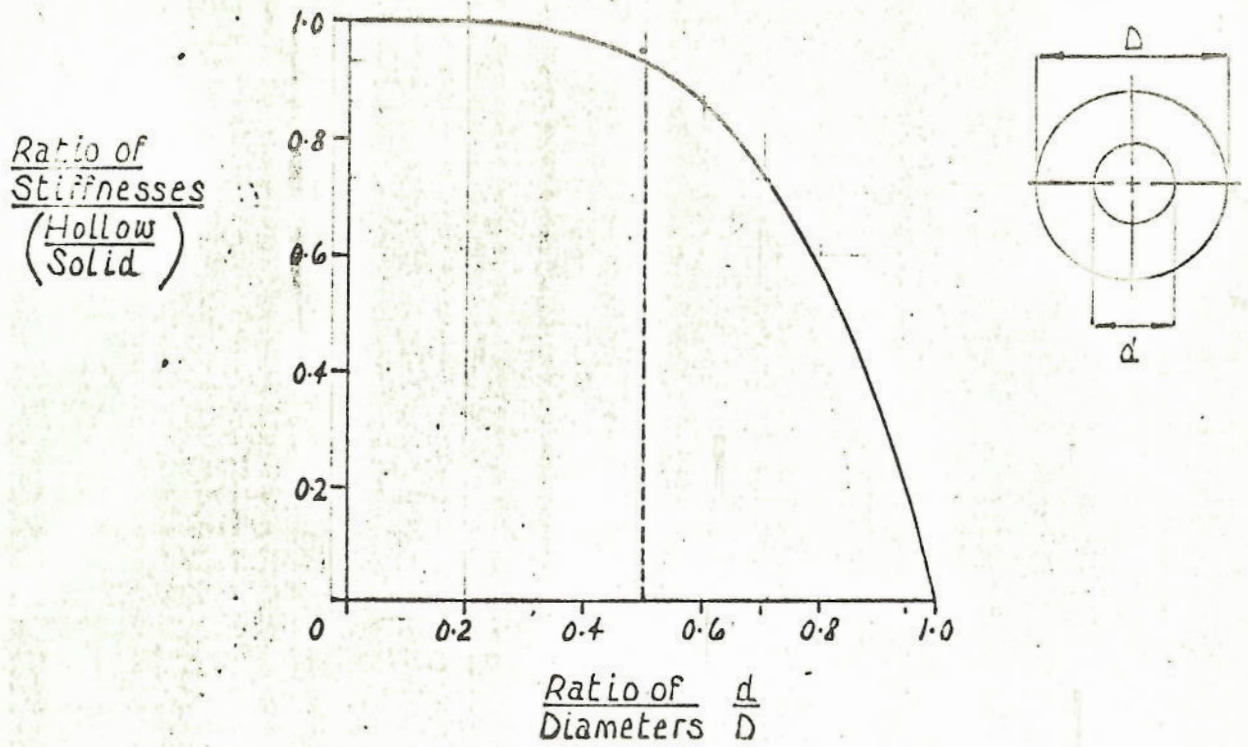


FIG. D3/2/1. THE EFFECT OF A HOLLOW CENTRE ON SPINDLE STIFFNESS.

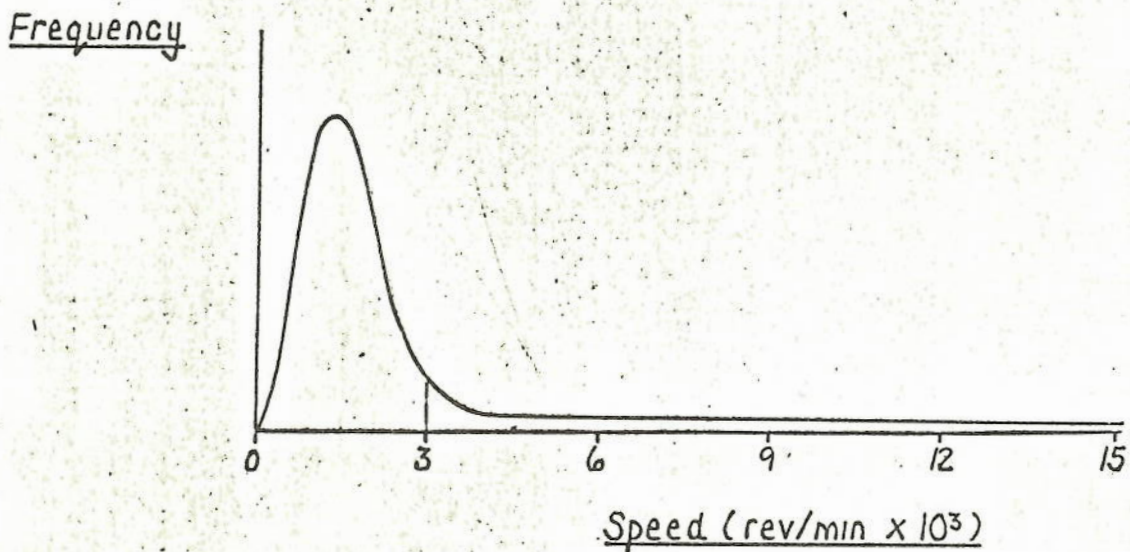
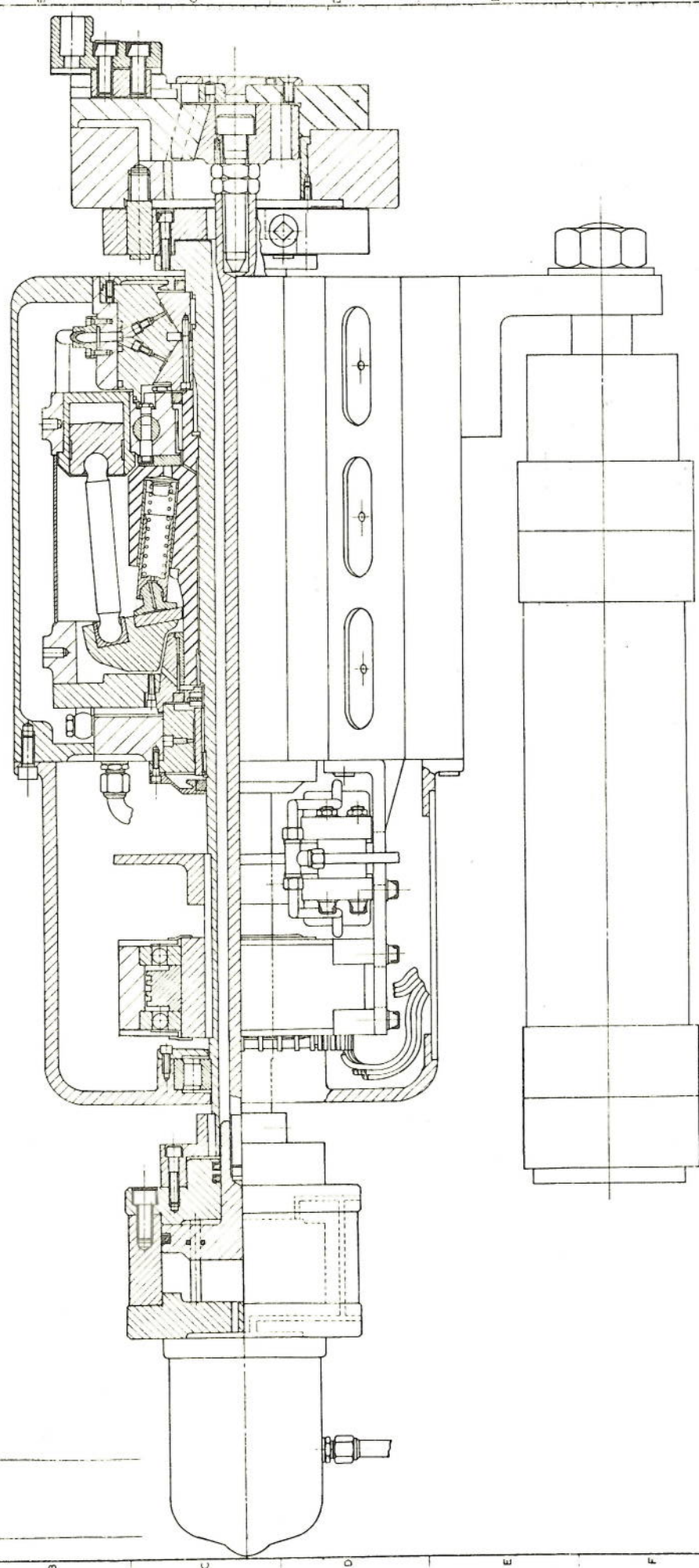


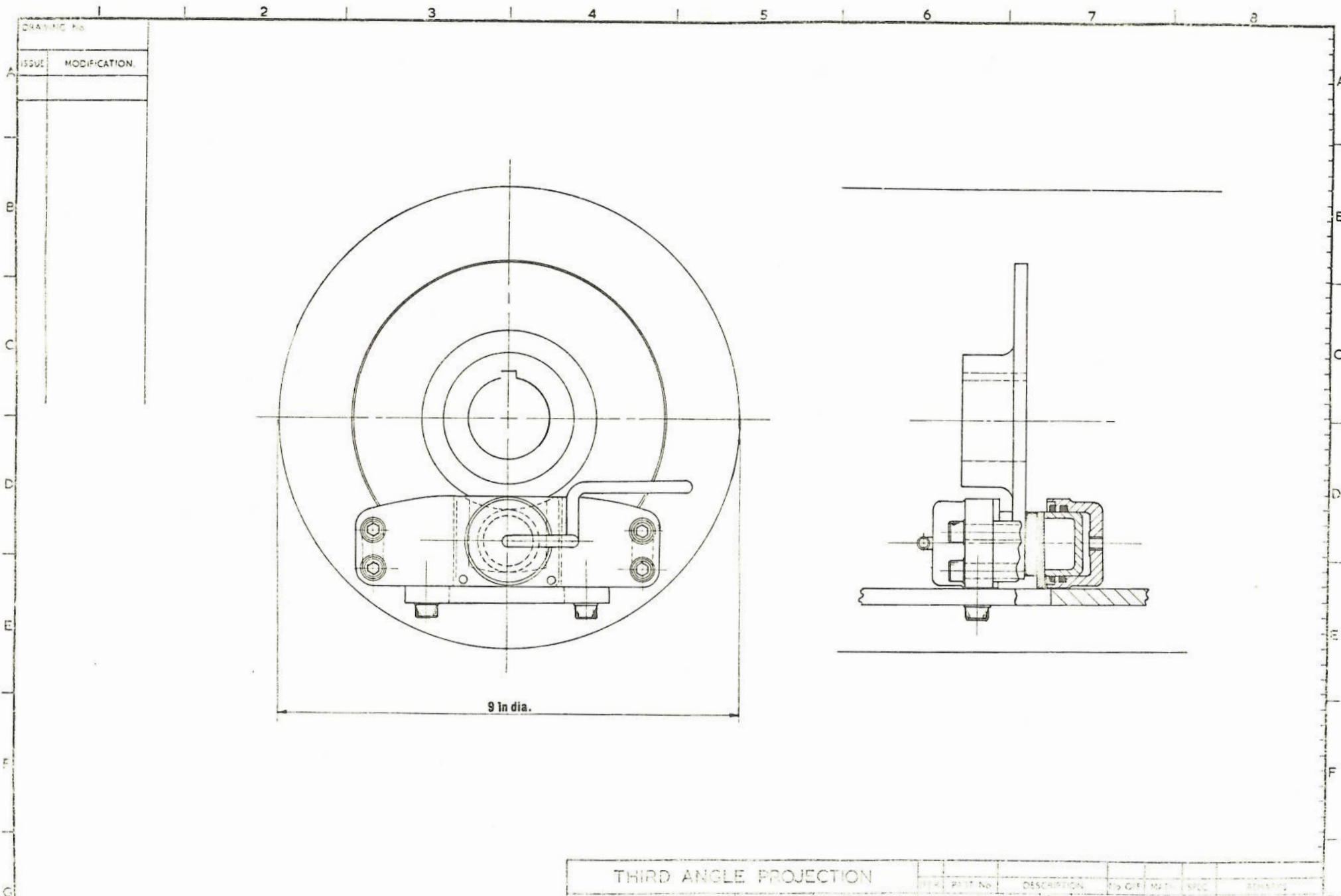
FIG. D3/2/2. DISTRIBUTION OF SPINDLE SPEED.

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 ISSUE MODIFICATION _____



6 in.
 150 mm.

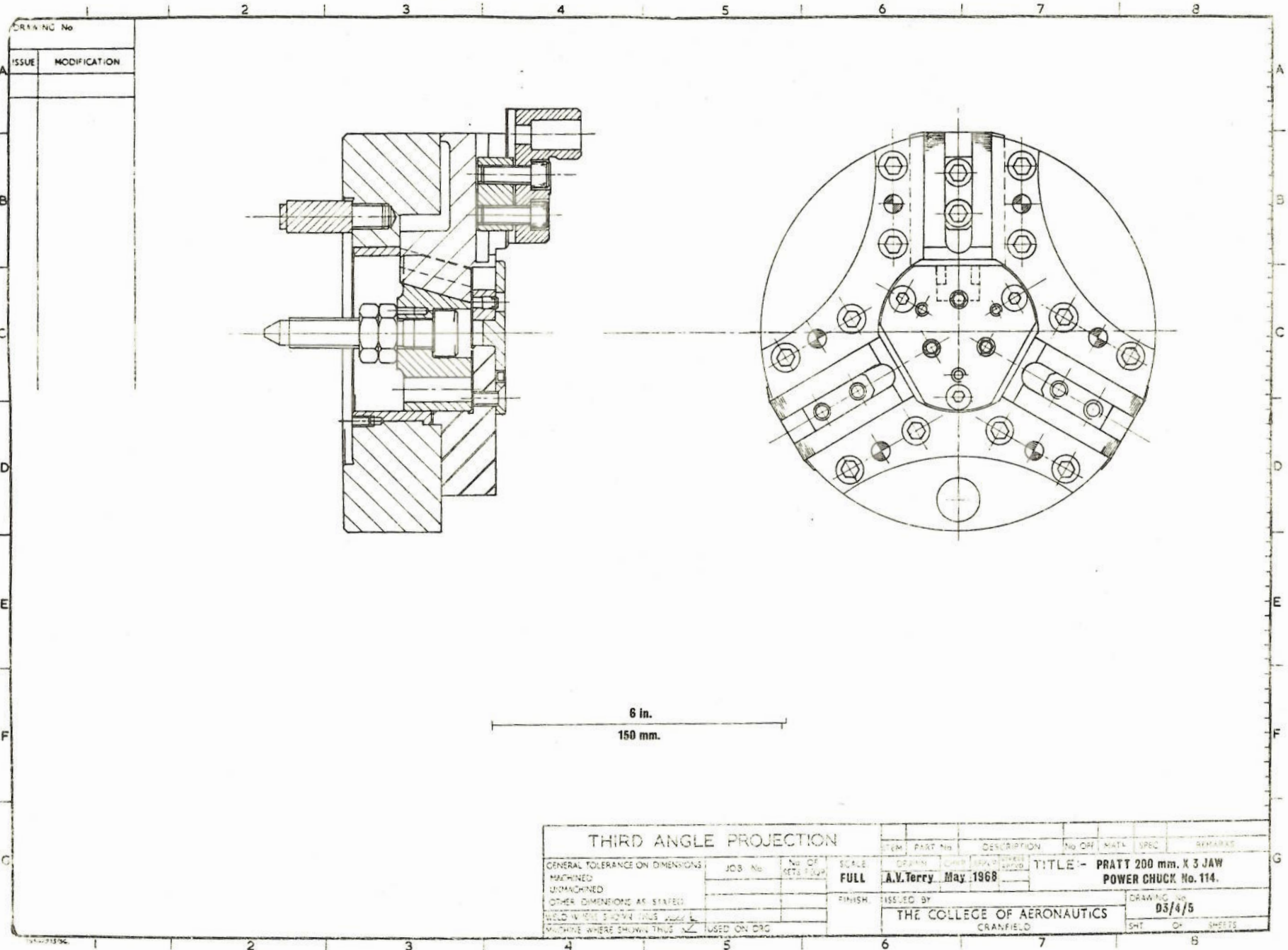
THIRD ANGLE PROJECTION		STUDENT NAME: _____ STUDENT NO.: _____ DATE: _____ TITLE: HEADSTOCK ASSEMBLY
NAME: _____ COURSE: _____ DATE: _____ A. K. JERRY May 1968	FULL FULL	THE COLLEGE OF AERONAUTICS 03/4/5



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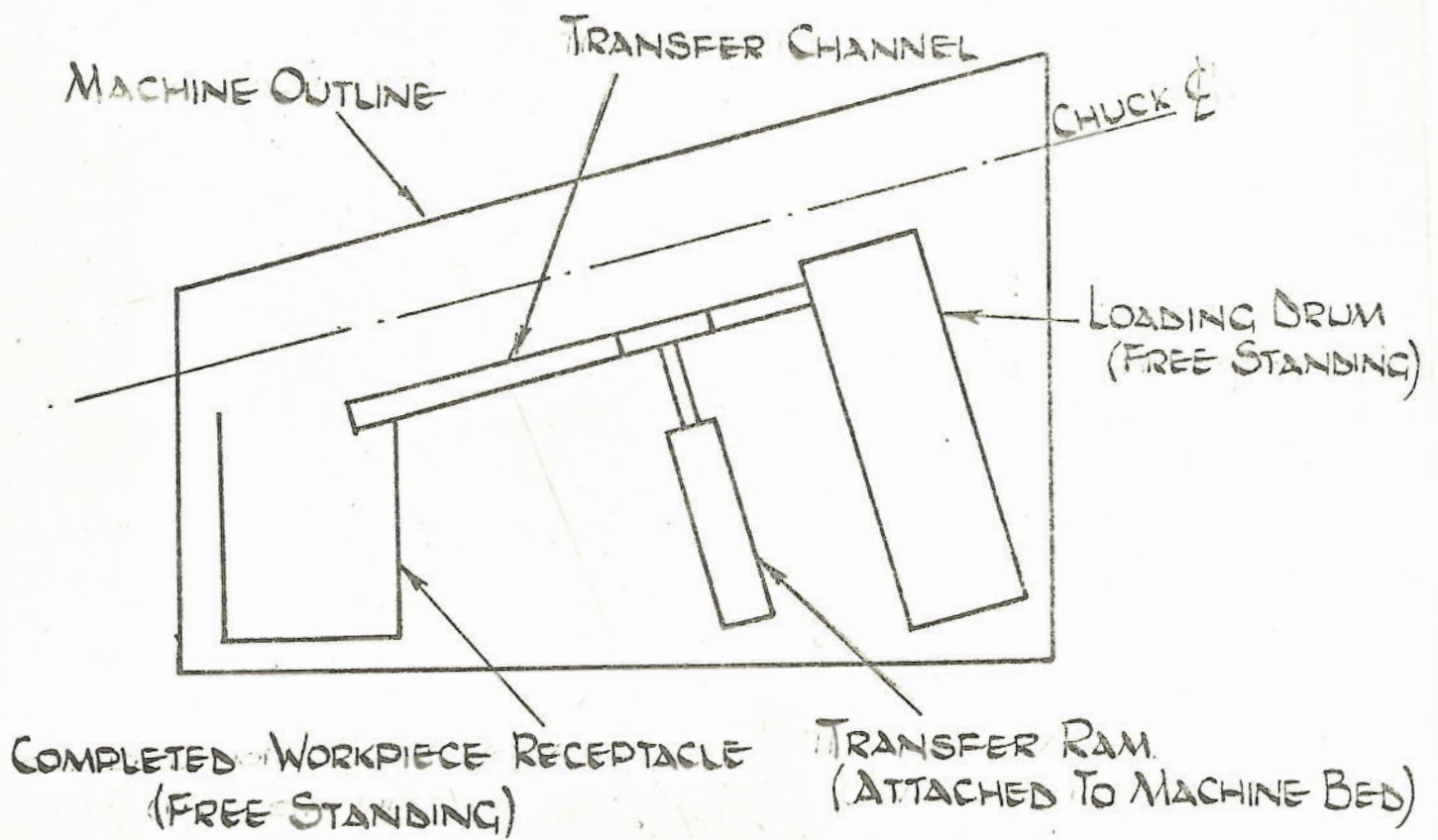
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GENERAL TO ALL FACED DIMENSIONS	JOB No.	DATE OF	SCALE	TITLE: DISC BRAKE ASSEMBLY							
FINISHED		ISSUE	FULL								
UNFINISHED			A.V. Terry May 1968								
OTHER DIMENSIONS AS STATED			FINISH	ISSUED BY	THE COLLEGE OF AERONAUTICS			DATE	05/4/4		
DRAWN BY				A.V. Terry							



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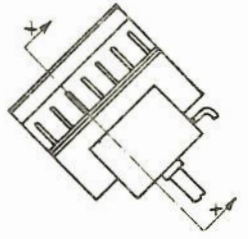
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GENERAL TOLERANCE ON DIMENSIONS	JOB No.	No. OF SETS TO USE	SCALE	DRAWN	CHECKED	DATE				
MACHINED			FULL	A.V. Terry		May 1968				TITLE - PRATT 200 mm. X 3 JAW POWER CHUCK No. 114.
UNMACHINED										
OTHER DIMENSIONS AS STATED			FINISH	ISSUED BY				DRAWING No.		
WELD WELLS SHOWN THIS WAY				THE COLLEGE OF AERONAUTICS				D3/4/5		
MACHINE WHERE SHOWN THIS WAY			USED ON DRG	CRANFIELD				SHT	OF SHEETS	



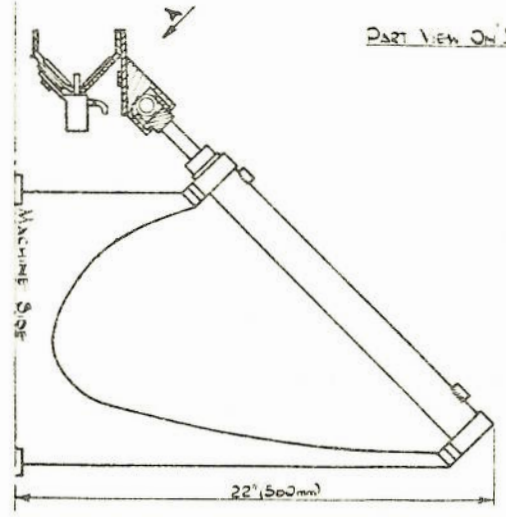
DIAGRAMMATIC SCHEME OF WORK LOADING ARRANGEMENT

D3/4/7

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PART VIEW ON A



SECTION XX

THIRD ANGLE PROJECTION				ITEM	PART No	DESCRIPTION	No OFF	MATL	SPEC	REMARKS
GENERAL TOLERANCE ON DIMENSIONS	JOB No	NO OF SETS REQ	SCALE	DRAWN	CHECK	APPROVED				
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UNMACHINED			FINISH							
OTHER DIMENSIONS AS STATED										
WELD WHERE SHOWN THIS										
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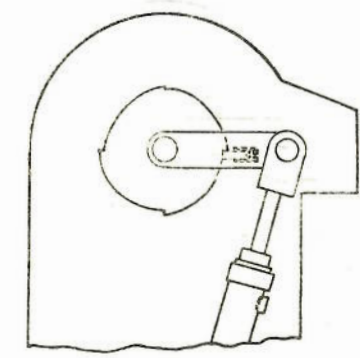
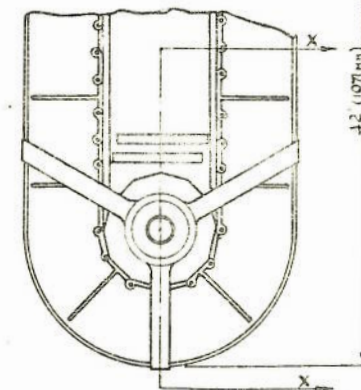
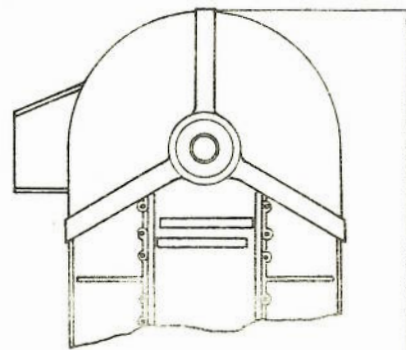
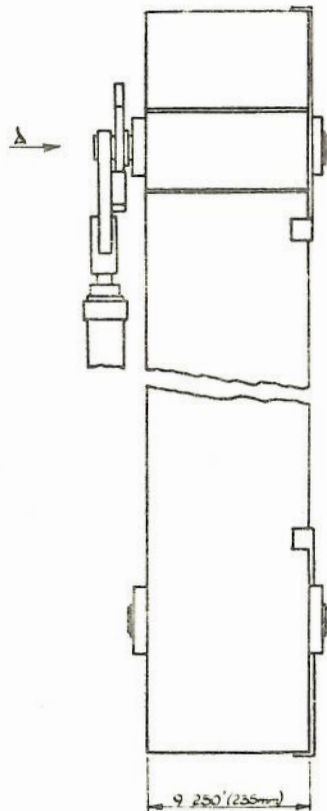
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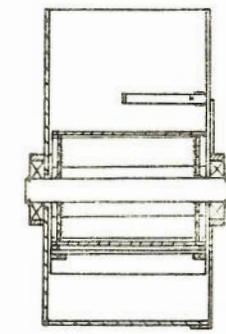
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DRAWING No

ISSUE	MODIFICATION



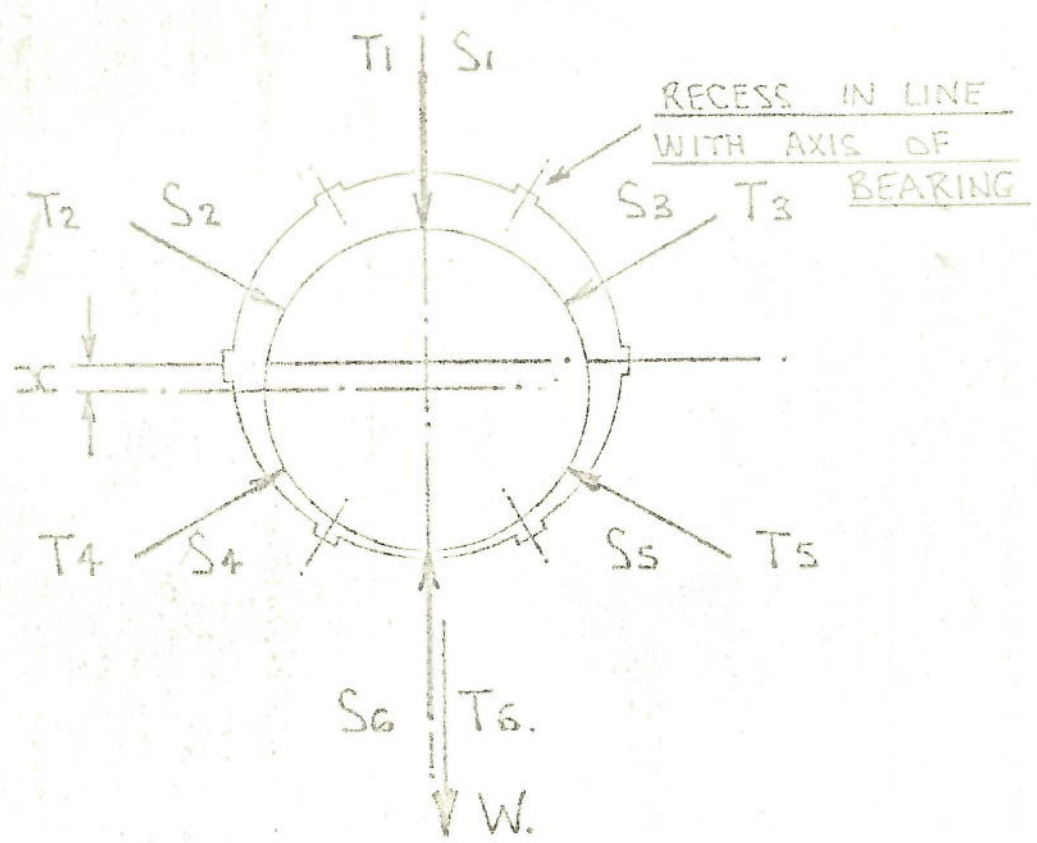
VIEW ON A



SECTION XX

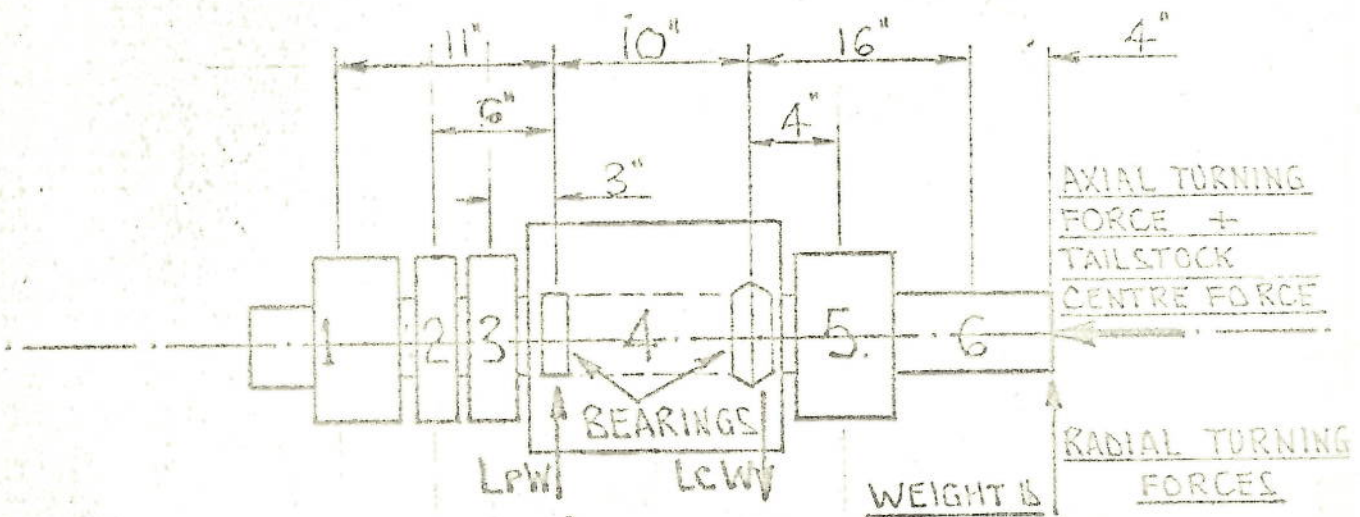
THIRD ANGLE PROJECTION

GENERAL TOLERANCE ON DIMENSIONS	JOB No	No OF SETS REQD	SCALE	ITEM	PART No	DESCRIPTION	No OFF	MATN	SPEC	REMARKS
MACHINED			1/4 FS	DRAWN	CHKD	APPROV'D				
UNMACHINED			FINISH	DATE						
OTHER DIMENSIONS AS STATED				ISSUED BY						
FIELD WORK SHOWN THIS WAY				THE COLLEGE OF AERONAUTICS						
MACHINE WHERE SHOWN THIS WAY	USED ON DFC			CRANFIELD						
						TITLE:-				DRAWING No.
						WORK LOADING DRUM				SHEET OF SHEETS



HYDROSTATIC JOURNAL BEARINGS THRUST & STIFFNESS NOTATION

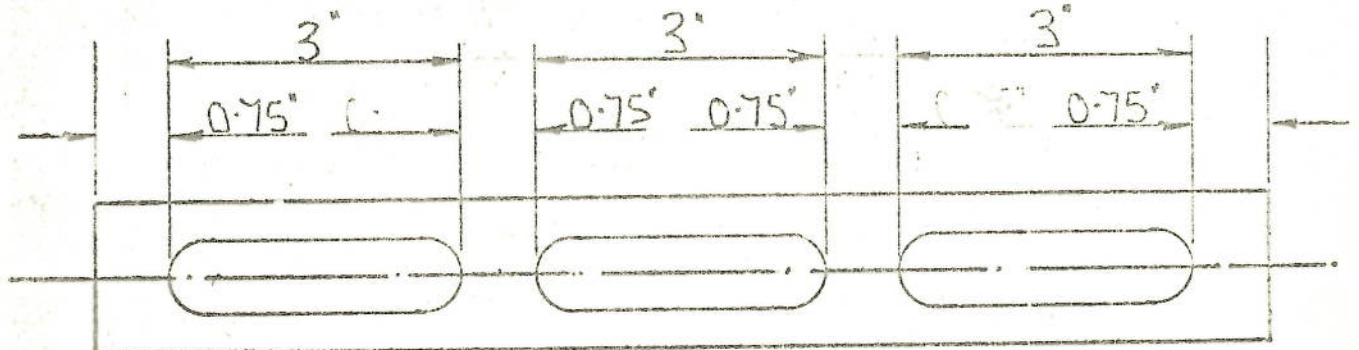
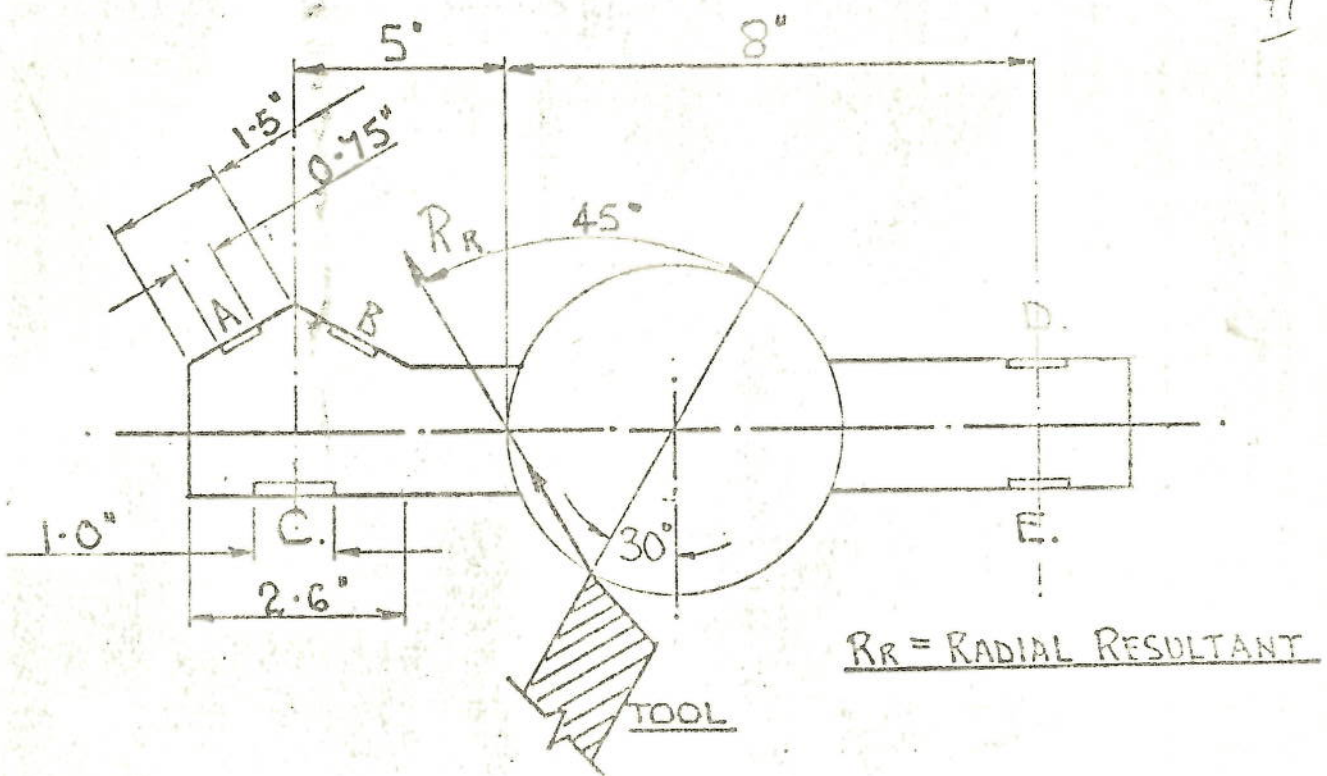
FIG D3/5/10



1. CHUCK ACTUATION CYL & DISTRIBUTOR.	45
2. ROTARY ENCODER	20
3. DISC BRAKE	12
4. HYDRAULIC MOTOR.	200
5. CHUCK	20
6. WORKPIECE.	25

LOADS ACTING ON HEADSTOCK

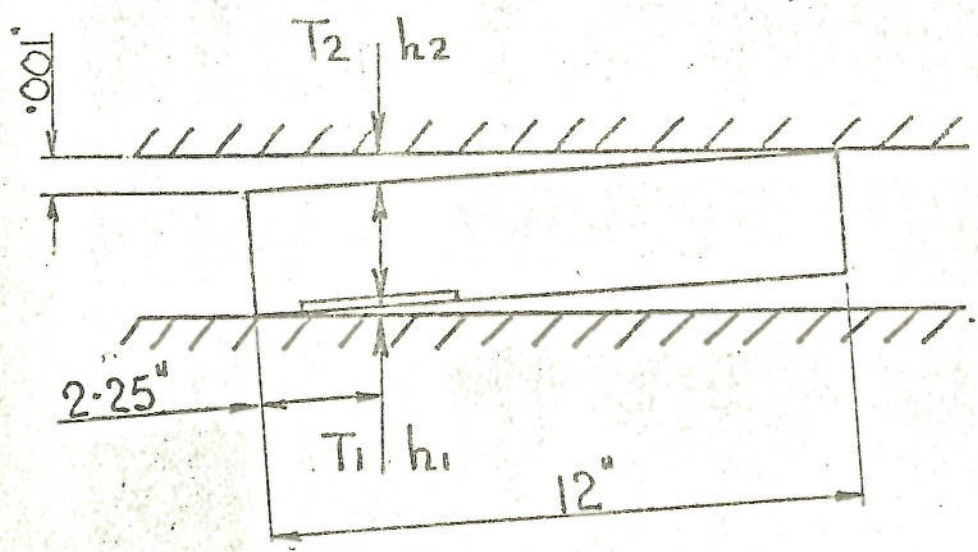
FIG D3/5/11



VIEW ON FACES A, B, C, D & E

HYDROSTATIC SLIDEWAY BEARINGS.

FIG D3/5/12



HYDROSTATIC SLIDEWAY BEARINGS
UNDER MAXIMUM LOAD CONDITIONS.

FIG D3/5/13

D4. REPORT OF THE FRAME AND SLIDWAYS SUB-COMMITTEE.

Prepared by:- N.H. Powell
M.F. Scarffe
M.L. Norton.

D4.1. Introduction.

The design of a machine frame and slideways suitable for a machine capable of precision turning and also a limited amount of milling poses two problems. The first of these is that the frame would have to be a very stiff structure and the second is that the slideways would have to provide exact alignment and a sliding surface with a low co-efficient of friction.

The report of the Configuration Committee put forward certain design proposals which have been considered and included where possible.

The box-type structure for the moving headstock and tailstock is a feature which has been included, but in the case of the tool-post a compromise has been reached. Measurement along the axis of movement has been retained for the tool-post but not for the headstock, and the actuation of the moving elements has been separated from the axis of movement.

Hydrostatic bearings have been used where possible to provide an almost frictionless movement for all sliding parts.

A survey was carried out into the various methods of slideway protection and the results and recommendations are included at the end of this report.

D4.2. Basic Requirements.

This can be divided into two sections:-

Machine Frame.

This is required to be of sufficient strength and rigidity to absorb the forces encountered in all the machining operations performed on the work, and as a result the deflection of the machine frame should be a minimum.

It should be stable in the static case withstanding the weight of the headstock, tool-post, tailstock, hydraulic actuators and all ancillary equipment which is mounted on the machine frame structure. In the dynamic case it must close the flow of the operating forces which are exerted between workpiece and cutting tools.

The design of the machine frame should enable it to be mounted kinematically with the minimum of restraints so that it functions correctly without reliance on the foundations for extra stiffness. ref. 1.

Slideways.

The machine tool slideways should provide exact alignment of all

moving parts in all positions of movement under the effect of operational forces.

The slideways should satisfy the following requirements:-

- 1) Provide the movement required with the minimum of friction and the maximum stiffness.
- 2) Provide a means of compensating for wear.
- 3) Allow ease of assembly and economy in manufacture.
- 4) Be free from restraint.
- 5) Prevent chip accumulation and provide for easy removal of chips.

D4.3. Survey.

The previous work done by the Configuration Committee in preparing a line diagram of the machine resulted in a layout in which the moving parts were housed in a box-type structure.

This arrangement is considered to be very sound in that it forms the basis for a closed loop structure which should result in an extremely stiff frame. This will therefore result in a 'T' shaped frame, the headstock and tailstock moving in one leg of the 'T' and the tool-post moving in the other, and at right angles to it. This will enable the machine to be mounted kinematically with a support under each limb of the 'T'.

A comparison of the stiffness of four cross-sections of equal height h and of equal cross-sectional area is shown in fig. D4/3/1.

Within practical limits of the ratio width-to-height (k), i.e. $k = 0.5$ to 1.5, the closed box cross section appears most favourable because, compared with the tubular section, the slightly lower torsional stiffness is more than compensated for by the increased bending stiffness. In addition, the ratio between free length and cross-sectional area is important and has to be given special consideration having regard to the properties of the material employed. This becomes particularly important when for certain reasons the stiffness and also the strength has to be considered and when, for instance, a choice has to be made between the use of cast iron or a welded steel fabrication for a machine structure. (ref.1).

In general either cast iron or welded steel can be used. Both have advantages and disadvantages which must be considered for technical and economic reasons.

The properties to be considered are:-

(a) Material:

- 1) Strength under tensile and compressive loads.
- 2) Stiffness
- 3) Vibration characteristics (damping).

(b) Manufacture:

- 1) Maintaining accurate wall thickness in castings is difficult since core displacements can occur, so a generous allowance must be made. With welded structures this problem does not arise.
- 2) Combining different wall thicknesses in a weldment is not difficult. In the case of a casting faults may arise, so the casting is made thicker, and hence heavier than need be.
- 3) Machining allowance, this is such that no trace of skin remains after machining.

A recent article in the "Production Engineer" stated that:-

Steel is much better than cast iron. It immediately gives a structure of twice the stiffness for the same weight as cast iron. Damping properties of cast iron are very slight compared with natural damping of slides, jointing faces, etc., and although steel structures may be noisy, vibration properties invariably are better overall. (Ref.2.)

(c) Economy:

- 1) The cost of patterns may be prohibitive for small quantities but this must be balanced against the cost of welding fixtures.

Broadly speaking welded construction is more suitable for simple structures whilst for complicated shapes the cast design appears preferable.

- 2) The overall weight of the machine could prove important when export markets are considered which involves transport and Customs. The possibility exists, therefore, of using light alloy castings for some parts.

Influence of swarf on frame design.

High speed turning and milling machines produce large quantities of chips per unit time, the problems of chip removal and chip disposal have to be borne in mind during the design of the machine bed. The general requirements of slideways have already been listed.

A review of existing slideway design shows that it is based on one or more of the following elements and these can be arranged in different configurations.

- 1) The Vee
- 2) The flat
- 3) The dovetail
- 4) The cylinder

A drawing of these is shown in fig D4/3/2.

Any configuration must satisfy kinematic design (Ref. 3), that is a body in free space has six degrees of freedom, three axes for displacement and three axes for rotation.

To locate the body in one axis five constraints must be applied, that is three rotational and two axial.

D4.4. Design Solution.

The solution to the design of the frame depends to a large extent upon the arrangement of the various machine elements within the frame.

One of the features that it was considered should be incorporated in the machine is the "principle of alignment", where the axis of measurement and the axis of the workpiece are coincident. Therefore a design layout was prepared which included this feature; this is shown in fig D4/4/3. In this the measuring system for the headstock movement is placed behind the headstock and in line with the axis of the work. The measuring system for the tool-post is placed on the centre line of the tool-post station and at right angles to the axis of the work.

This drawing also shows the total forward movement of the headstock and work, which is 22 inches (550 m.m) in this case. This is made up of an 8 inch. (200 m.m) movement for drilling, then a forward movement to get the work to the turning position, then finally an 8 inch. (200 m.m) movement past the turning tool. The total movement is controlled by a hydraulic ram, and it is necessary to have controlled feedrate over the entire stroke. The result is a machine of large overall length and this is considered to be excessive for a machine whose maximum work length is 8 inches. (200 m.m.).

The design layout shown in fig. D4/4/4. represents a compromise on the "principle of alignment", in that the measuring system for the headstock movement is placed underneath the headstock, whilst it is retained in the same position for the tool-post. The headstock movement is shared by two rams, one

giving an 8 inch (200 m.m.) controlled feedrate movement, and the other a 14 inch. (350 m.m.) point to point movement. In this way the measuring system need only cover the movement of the 8 inch (200 m.m.) ram. The other ram will be actuated when it is required to move the work to the turning position. This will carry the controlled feedrate ram and measuring system with it. This arrangement results in a more compact machine.

The design layout shown in fig. D₄/4/5 incorporates a fixed headstock and the total movement of the cutting tools being obtained from the saddle. In this design the saddle is actuated by two rams in the same way as the headstock in the previous design, the measuring system also behaves in a similar way. The overall size of the machine is much the same.

Fig. D₄/4/6 shows a design which eliminates the point to point ram, this is achieved by moving the tool-post on a quadrant so that a displacement of 14 inches (350 m.m.) is obtained. The work movement is obtained by actuating the headstock with a ram of 8 inch (200 m.m.) controlled movement.

The configuration in fig D₄/4/7 shows a fixed headstock with the controlled feedrate movement being given to the tool-post, together with the quadrant movement of the previous design described above.

The design shown in fig. D₄/4/8 represents a compromise over the previous designs. The headstock has an 18 inch (450 m.m.) controlled feedrate movement. The need for a point to point ram is eliminated by specifying that all drills over 3 inches (75 m.m) in length are fitted into the tailstock. The tailstock movement is utilised to position drills so that the headstock movement only is required.

Of the six designs put forward for consideration, it was decided that the design shown in fig D₄/4/4 offered the best solution. The reasons for arriving at this decision are as follows.

In consultation with Control Committee it was agreed that an 8 inch (200 m.m) controlled feedrate movement would simplify the fluidic control system. Also the use of a moving headstock and a tool-post with a single axis movement at right angles to the work axis, was considered a superior arrangement to a tool-post with a two axis movement. Separation of the slideway movements should result in a reduction of the errors associated with compound slides.

In this design the mounting of the measuring system for the tool-post should enable an absolute reading to be obtained between tool-post and machine frame.

Finally in meeting all of these conditions the design still offered a machine which was as compact as any of the others.

Fig. D₄/4/9 shows a design which incorporates all of the above features, but in addition the axis of the work is tilted through an angle of 45°, and the tool-post slide is tilted through 90° into the well of the machine, these two changes result in a machine occupying a small floor space, without sacrificing any of the previously mentioned features.

Fig. D4/4/10 shows the design layout for the headstock. In this design the headstock is cradled between the slideway bearings of the machine frame. This arrangement enables the axis of the work to be in line with the headstock supports, and as a result any deflection of the bearings supporting the headstock is along the work axis.

In the work cutting tool relationship chosen, the turning tools have been positioned so that they move in a vertical direction, and as a result contact the underside of the work, as shown on drawing No. D4/4/9

In this arrangement an analysis was made of the cutting forces between turning tool and workpiece, as this is considered to be the worst machining condition, and as a result the slideways have been inclined at an angle of 30° to the horizontal. This will give the maximum support to the headstock and the best reaction to the cutting forces.

The headstock bearings have been so designed that they are kinematically correct, in that they provide five degrees of restraint with one degree of freedom.

Hydrostatic bearings have been chosen for the slideways, as it is considered that these offer features that are unattainable with conventional slideways using cast-iron on cast-iron with oil film separation. High stiffness values together with very low values of co-efficient of friction are inherent features of hydrostatic bearings, so that these are particularly suitable for use on a numerically controlled machine tool.

The runners for the headstock are cast integral with the motor casing, and as this is the moving member, the hydrostatic pads are machined in the runners. The male portion of the vee is on the sliding element and the female portion is on the frame, it is considered that this will provide the best arrangement as well as being the easiest to manufacture.

The top half of the frame supporting the headstock is split above the line of the hydrostatic bearings, and, therefore, is removable. This will enable the bearings to be machined in each half of the frame, and then bolted together for final assembly.

The slideway design for the tool-post, shown in fig. D4/4/11, is similar to that of the headstock, the main difference is that the hydrostatic vee is inverted so as to absorb the downward forces resulting from the cutting action.

The slideway design for the tailstock is again similar to that of the headstock in this case the main difference is the span between the bearing slides. This has been reduced in order to cut down the size of the tailstock.

The frame design is shown in fig. D4/4/12. It is made up of slideway which is common to the headstock and tailstock and a slideway which supports the tool-post. Across the centre of the machine is a sloping panel which braces together two vertical panels and also assists in ducting the swarf towards the rear of the machine.

D4.5. Slideway Protection.

Three categories of protective devices are available:-

- 1) Corrugated covers and bellows of rubber, leather or terylene.
- 2) Telescopic steel covers.
- 3) Wipers.

Corrugated bellows are particularly suitable for protection of ram shafts and leadscrews. Where a length in excess of 12 inches is involved, supports should be used. These should be spaced at intervals of 9 inches to 12 inches over the required length. End fixings are available of the flange or collar type, or a combination of both. The collar fixing generally being preferred because it does not require end fixing holes.

The compression ratios available are given in reference 4, tables 1 and 2. Where bellows are protecting moving parts such as a leadscrew, FORTOX provide a yellow and black design as a safety measure, so making the operator aware of the concealed danger.

In order to achieve maximum compressibility ratio a neoprene (FORTOX) NN20 bellow is recommended for ram protection, assuming of course that this material is suitable for the coolants and lubricants to be used.

Leather bellows are available for dry applications and aluminium asbestos for high temperature conditions.

Corrugated covers provide full protection for all vertical and cross slideways and main beds of fast traversing machines. The material is a woven terylene cloth impregnated with polyurethane rubber, two layers of terylene film, bonded together by a heat setting adhesive to form a laminate. These covers are claimed to be light in weight, strongly resistant to abrasion, swarf, hot chips, grinding dust and are immune to the effects of most machine coolants and lubricants.

FORTOX can supply these covers in standard sizes or to the customer's specification. These covers are built sectionally so that in the event of local damage a new section may be readily inserted.

This type of protective cover is recommended for the slideways, so completely enclosing them.

The covers would be flat rather than folded. Fortox recommend a fabricated material of relatively low stiffness. This is in order to eliminate the lifting of the cover from the bed, when under compression due to its own rigidity. FORTOX do not supply these from the shelf but their design department facilitates for the customers individual requirements. An approximate quotation obtained was £10 for the two slideways.

The second category, Telescopic Steel covers are considered unnecessary for this application. They would normally be used on larger machines where greater protection is required for impact on the top of the slideways, i.e. where operators are likely to stand or walk along the slideways. The cost of this would be approximately £30 - £40 per slideway.

Wipers are unsuitable because they only provide protection against foreign matter entering between the saddle and slideway. The swarf is pushed along the slideway in front of the saddle. Wipers are now made from polyurethane rubber in preference to felt. The design of the wiper includes a stainless steel case, the leading edge of which clears the larger swarf from the slides. Some swarf may be hot, but insulation against a build-up in latent heat is provided, the wiper is correctly positioned by the creation of an air pocket between the leading edge and the rubber insert.

Other less significant means of achieving slideway protection being, armoured aprons and roller blinds.

D4.6 References

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- 4) Machine Protection (Fortox)
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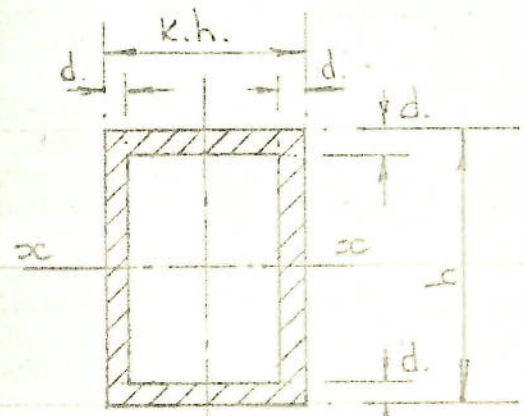
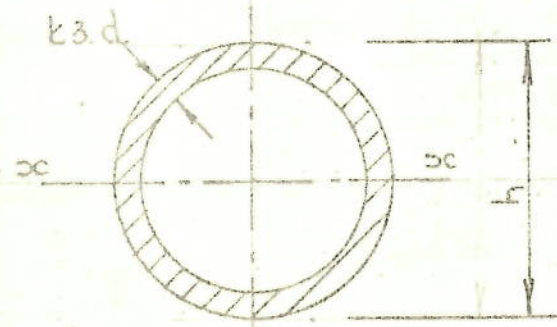
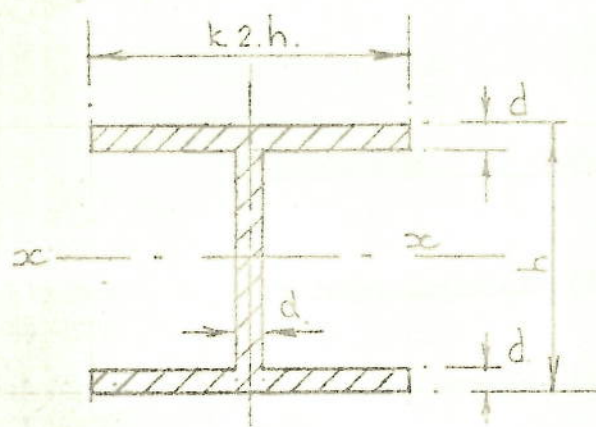
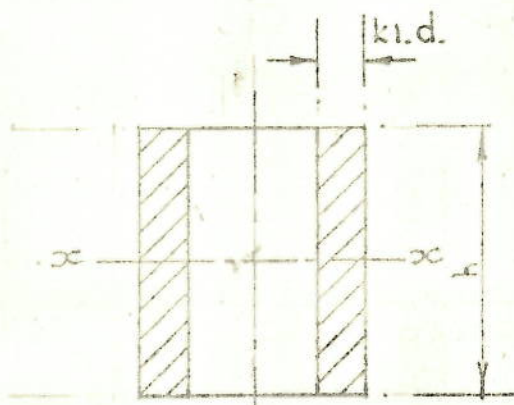


FIG. D4/3/1.
DIFFERENT CROSS SECTIONS.

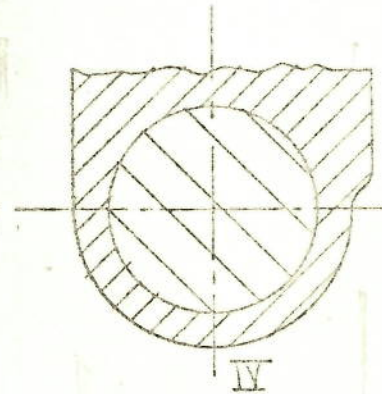
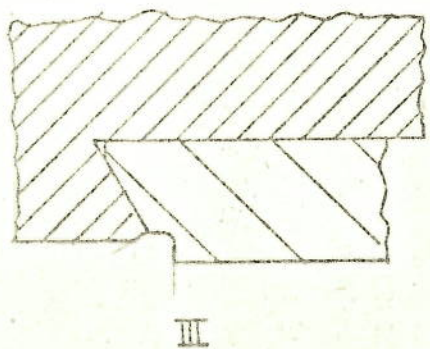
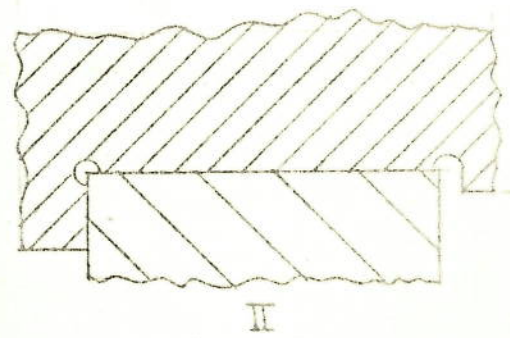
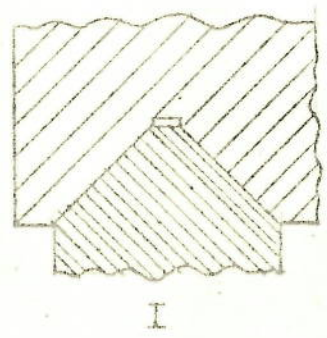
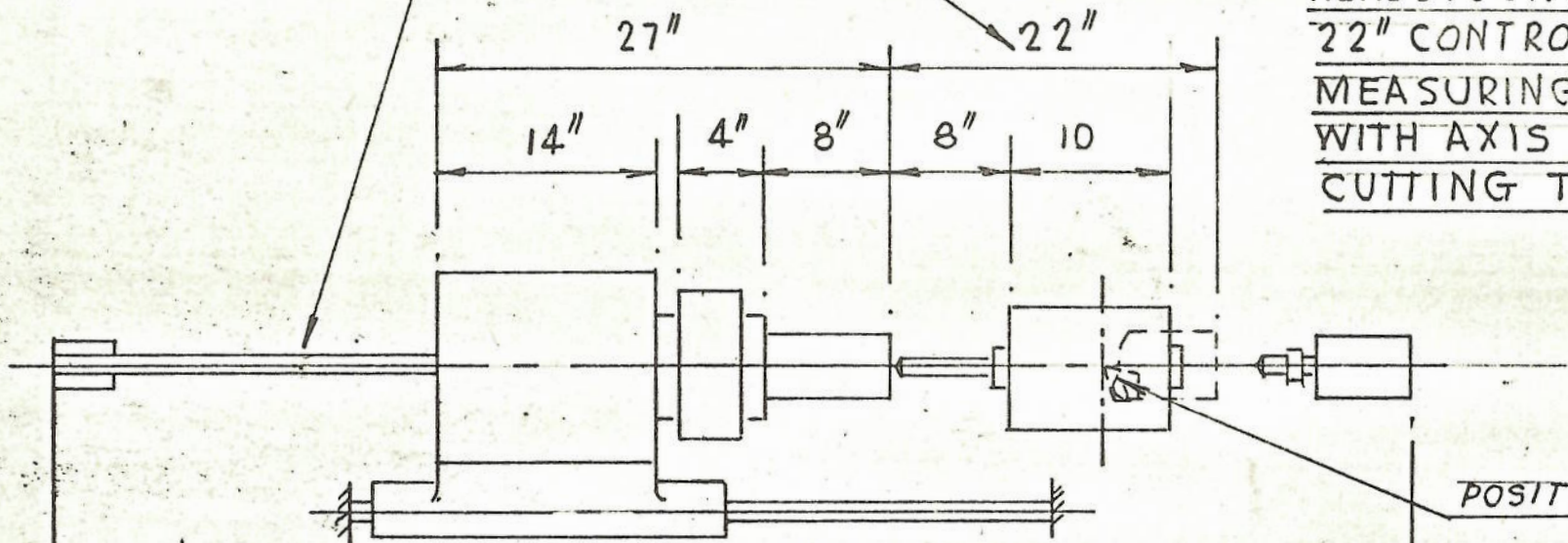


FIG. D4/3/2.
DIFFERENT SLIDEWAYS.

MEASURING SYSTEM.

TOTAL HEADSTOCK &
WORK MOVEMENT.

FEATURES —
SADDLE FIXED ON 'X' AXIS.
HEADSTOCK MOVEMENT —
22" CONTROLLED FEEDRATE.
MEASURING SYSTEMS IN LINE
WITH AXIS OF WORK &
CUTTING TOOL.



POSITION OF TURNING TOOL.

7'3"

FIG. D4/4/3.
MACHINE LAYOUT.

FEATURES -
SADDLE FIXED ON X AXIS.
HEADSTOCK MOVEMENT -
8" CONTROLLED FEEDRATE.
14" POINT TO POINT.

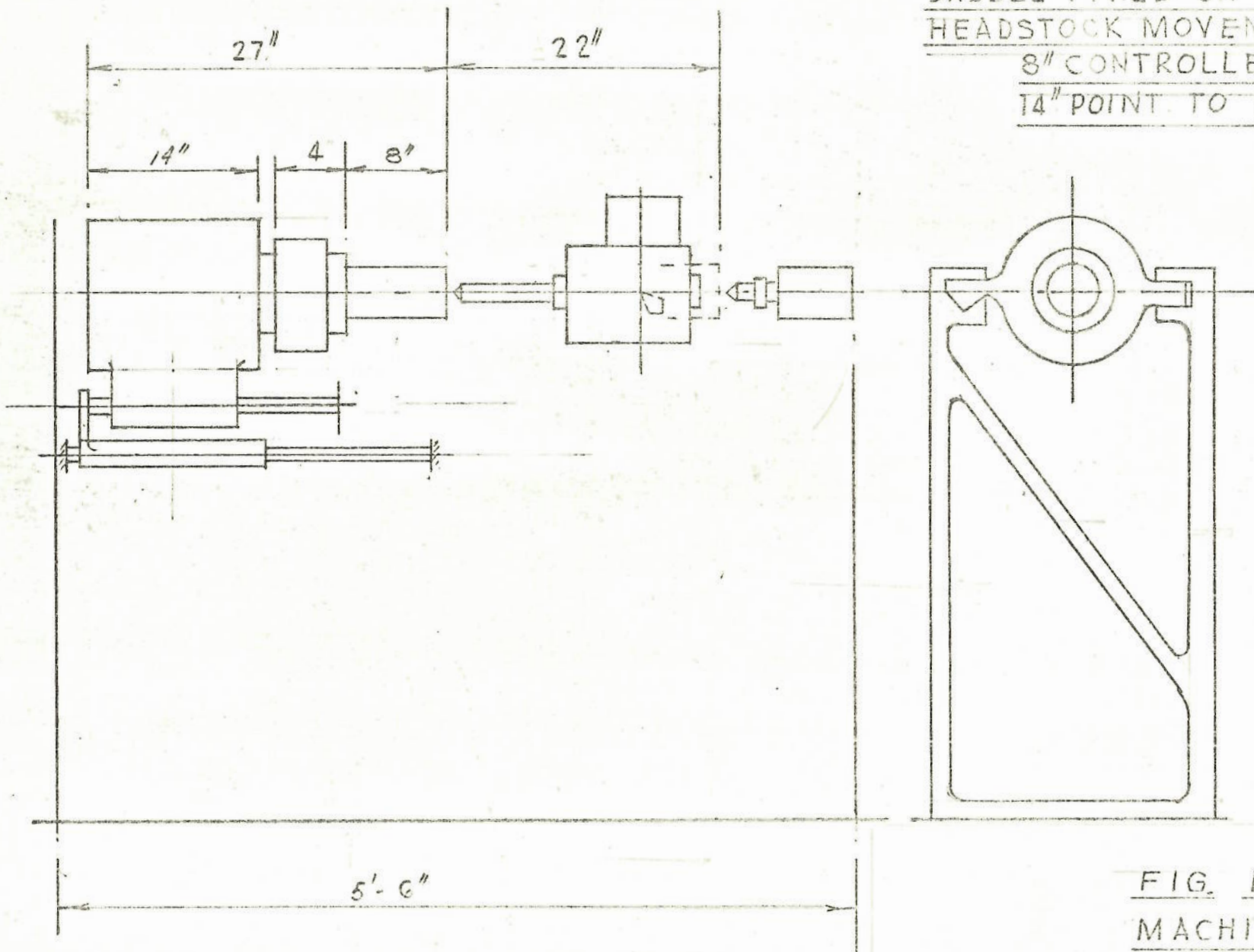
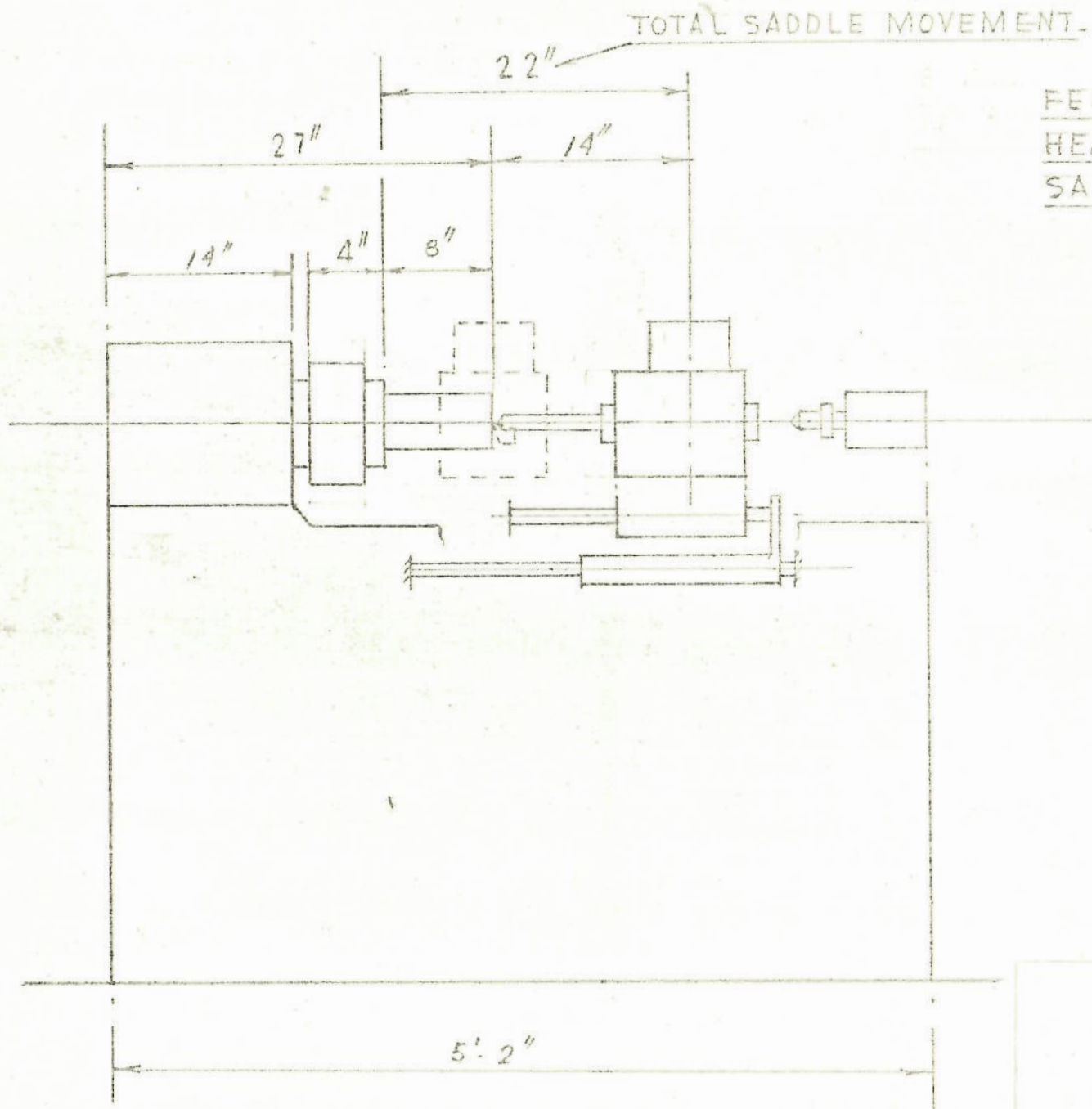


FIG. D4/4/4.
MACHINE LAYOUT.



FEATURES.

HEADSTOCK FIXED.

SADDLE MOVEMENT —

8" CONTROLLED FEEDRATE.

14" POINT TO POINT.

FIG. D4/4/5.
MACHINE LAYOUT.

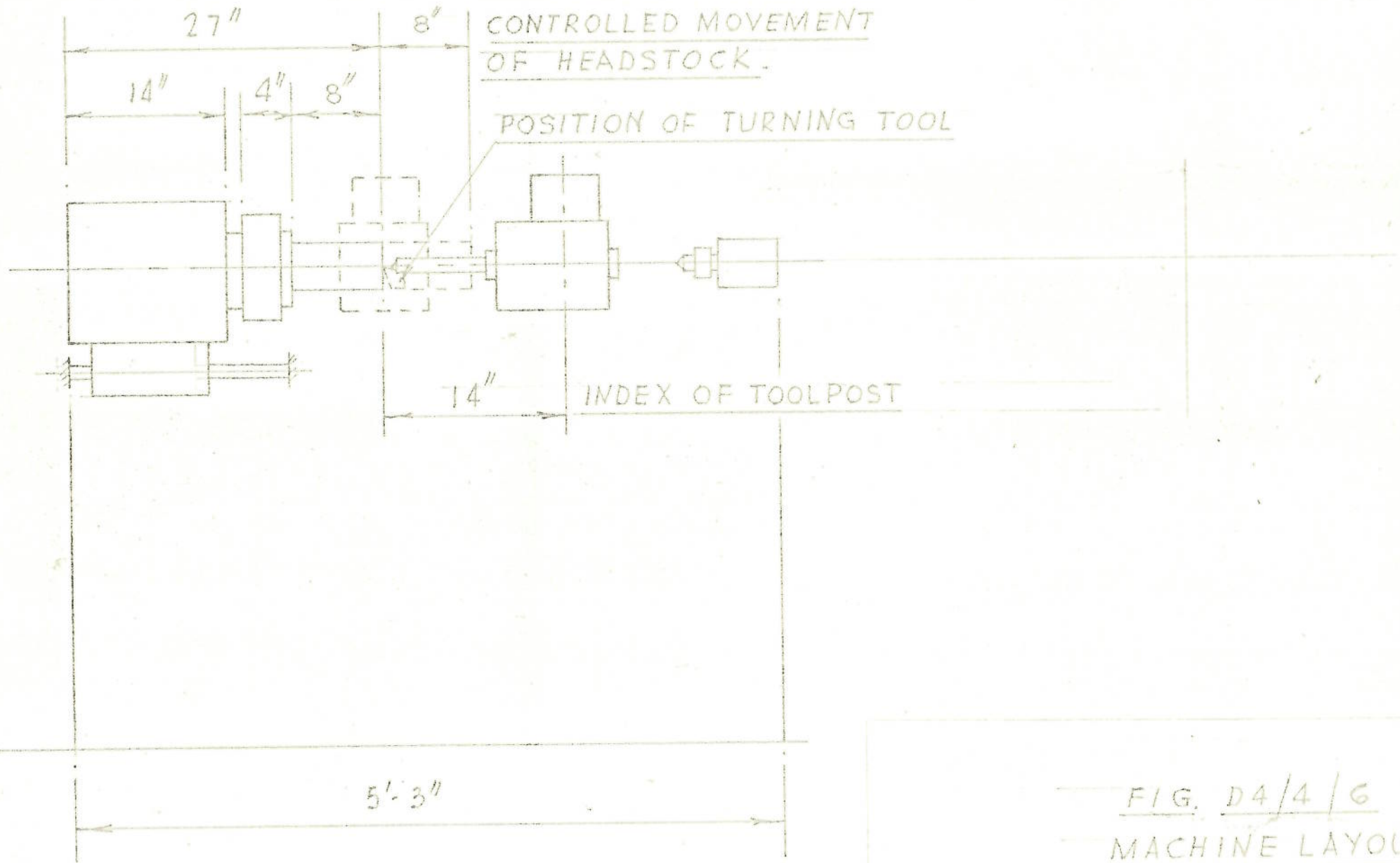
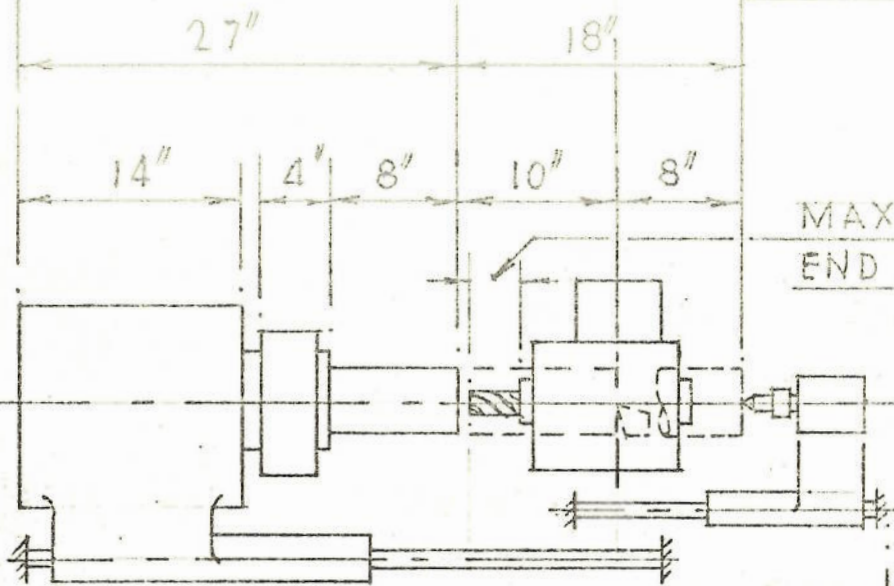


FIG. D4/4/G
 MACHINE LAYOUT.

WORK POSITION-FULLY RETRACTED.-

WORK POSITION-START OF TURNING OPERATION.-

WORK POSITION-FULLY FORWARD.-



MAX. PROJECTION OF
END MILLS - 3"

FEATURES -

SADDLE FIXED ON 'X' AXIS.

HEADSTOCK MOVEMENT -

18" CONTROLLED FEEDRATE.

TAILSTOCK MOVEMENT -

8" POINT TO POINT.

DRILLS ABOVE 3" IN LENGTH TO BE
MOUNTED IN TAILSTOCK.

4'-6"

FIG. D 4 / 4 / 8.

MACHINE LAYOUT.

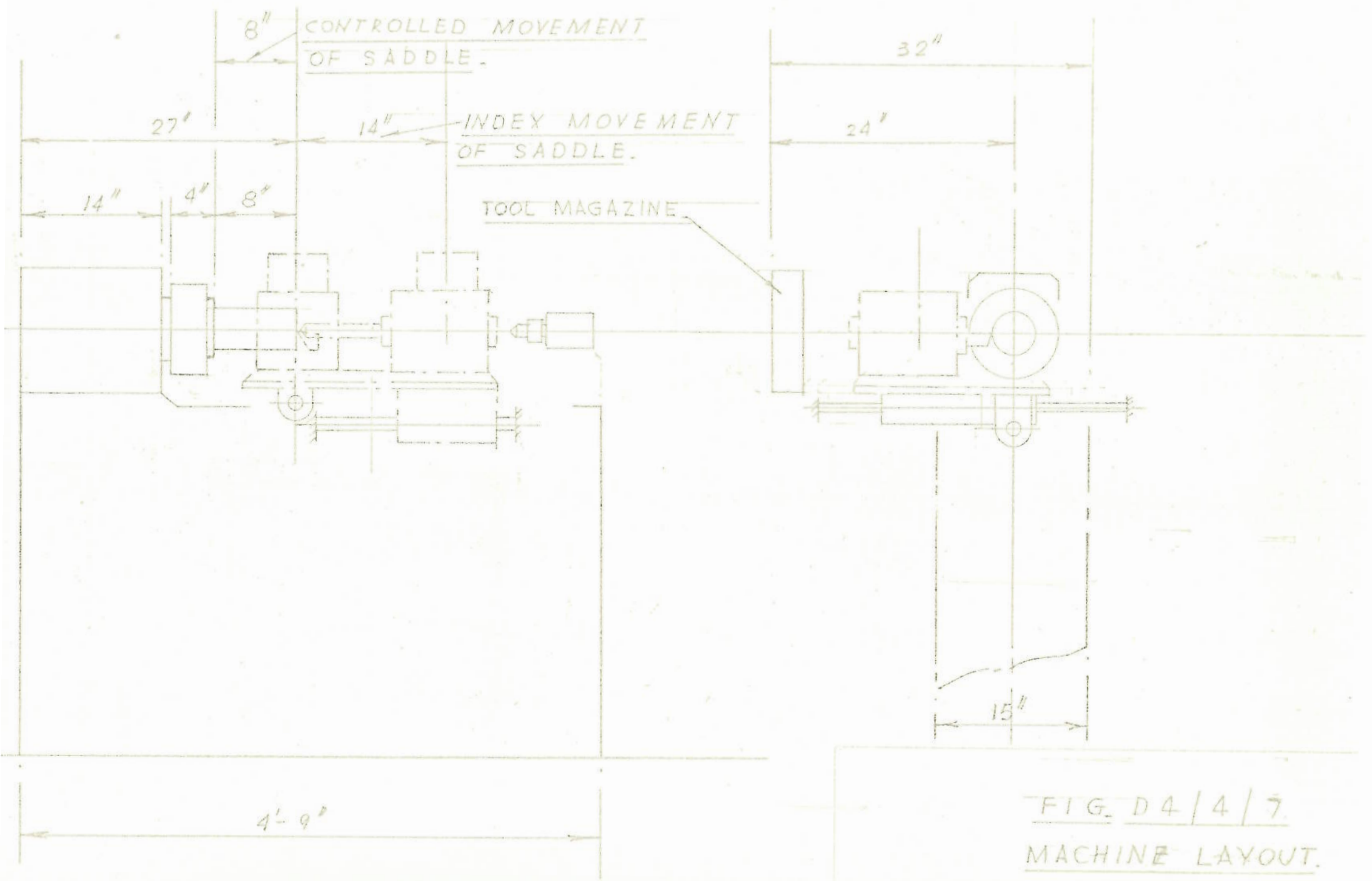


FIG. D4/4/7.
 MACHINE LAYOUT.

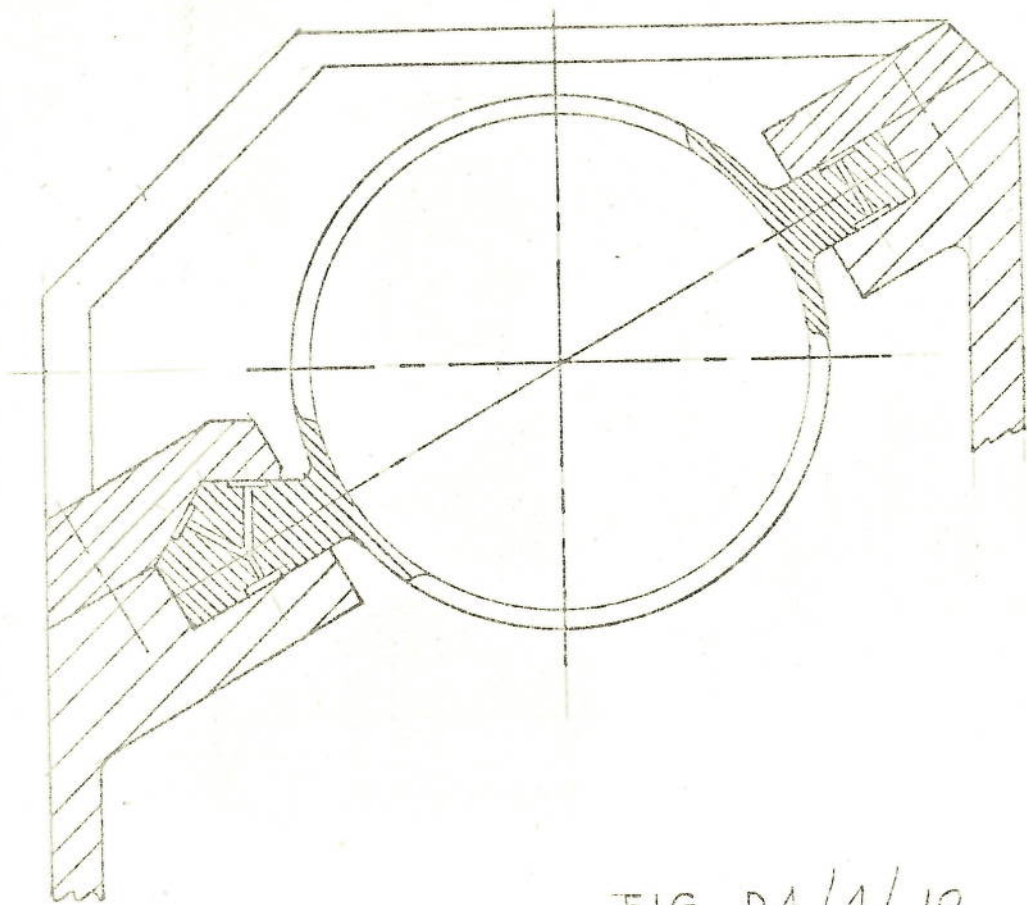


FIG. D4/4/10.
HEADSTOCK SLIDEWAY.

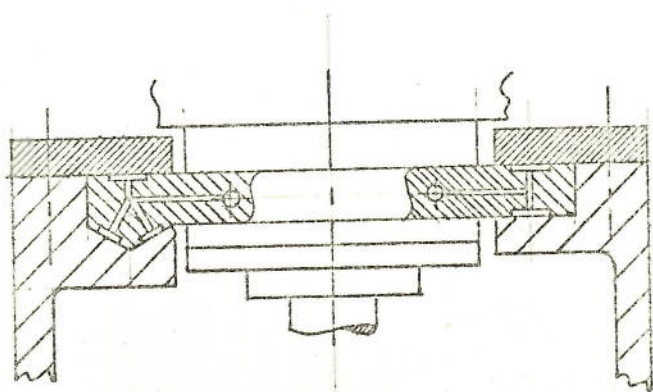
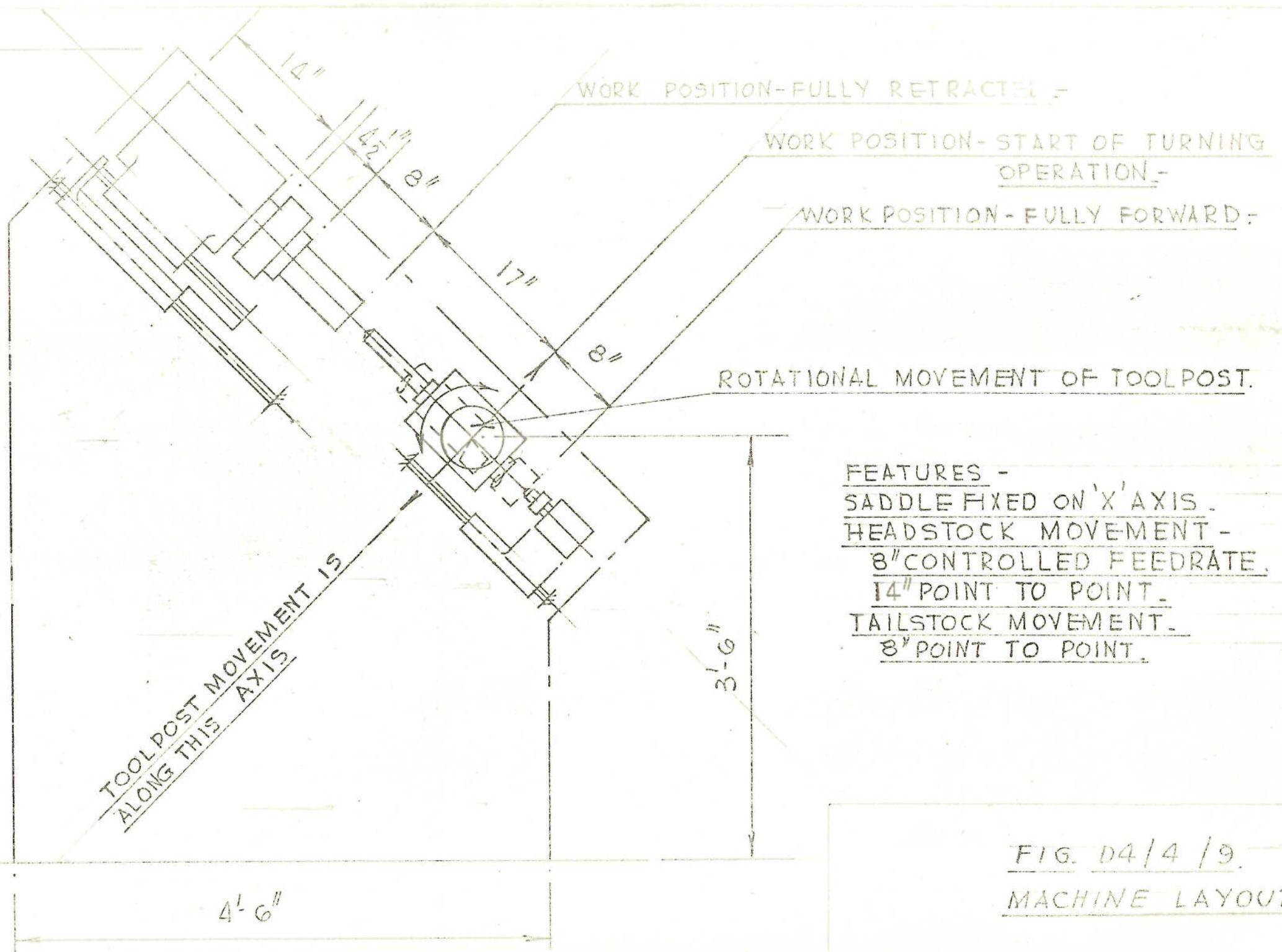


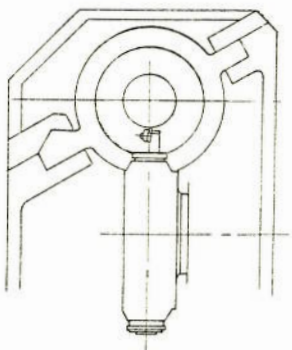
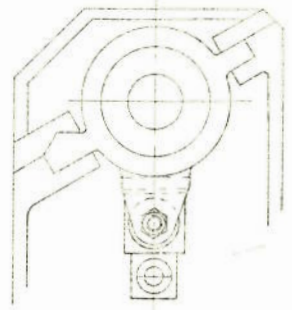
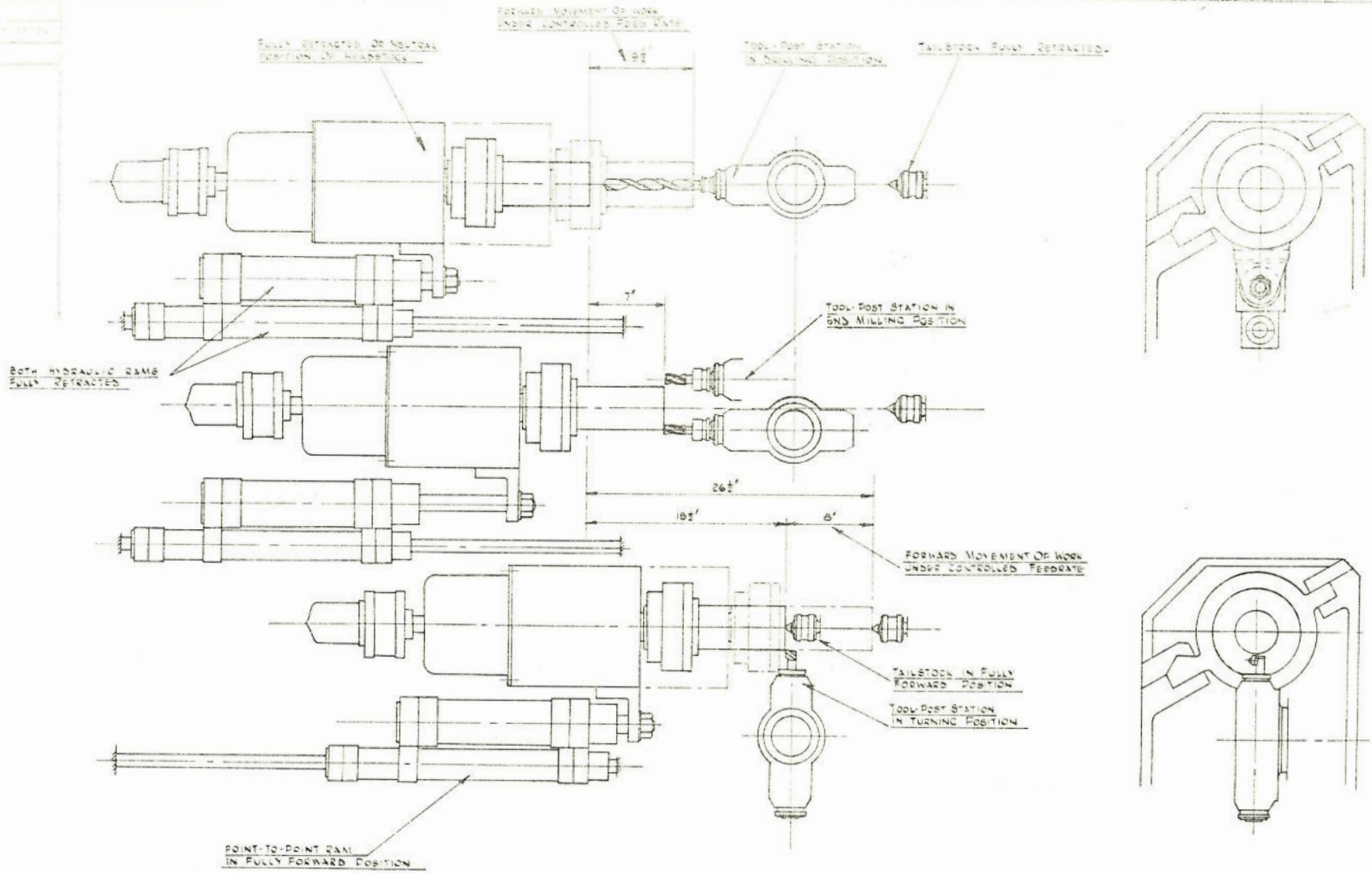
FIG. D4/4/11.
TOOL-POST SLIDEWAY.



- FEATURES -
- SADDLE FIXED ON 'X' AXIS.
 - HEADSTOCK MOVEMENT -
 - 8" CONTROLLED FEEDRATE.
 - 14" POINT TO POINT.
 - TAILSTOCK MOVEMENT.
 - 8" POINT TO POINT.

FIG. D4/4/9.

MACHINE LAYOUT.



THIRD ANGLE PROJECTION				DESIGN NO.	DATE	SCALE	TITLE
DESIGNED BY	CHECKED BY	APPROVED BY	DATE	100	1948	1:1	RELATIVE MOVEMENTS BETWEEN WORK & CUTTING TOOLS
THE COLLEGE OF AERONAUTICS				A-4-4-18			

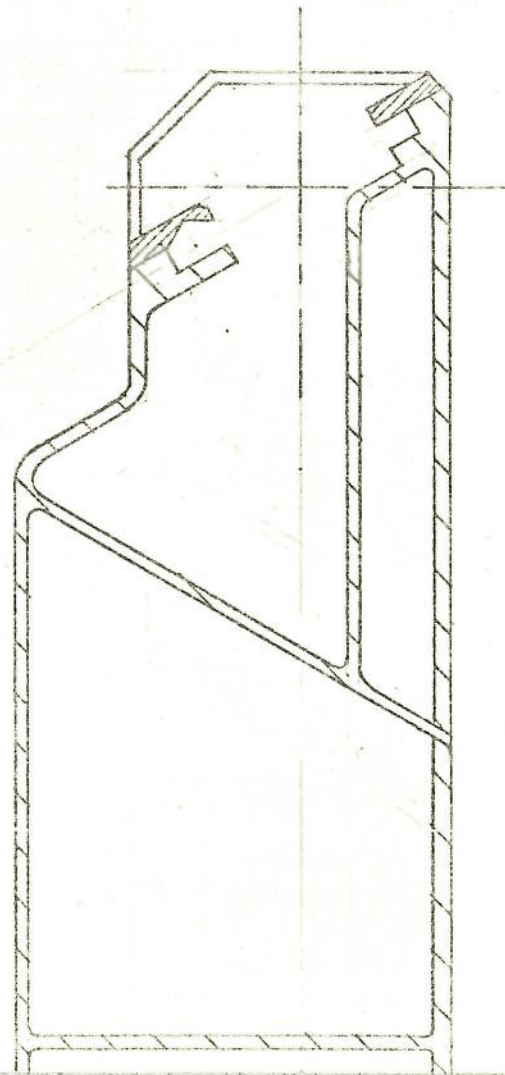
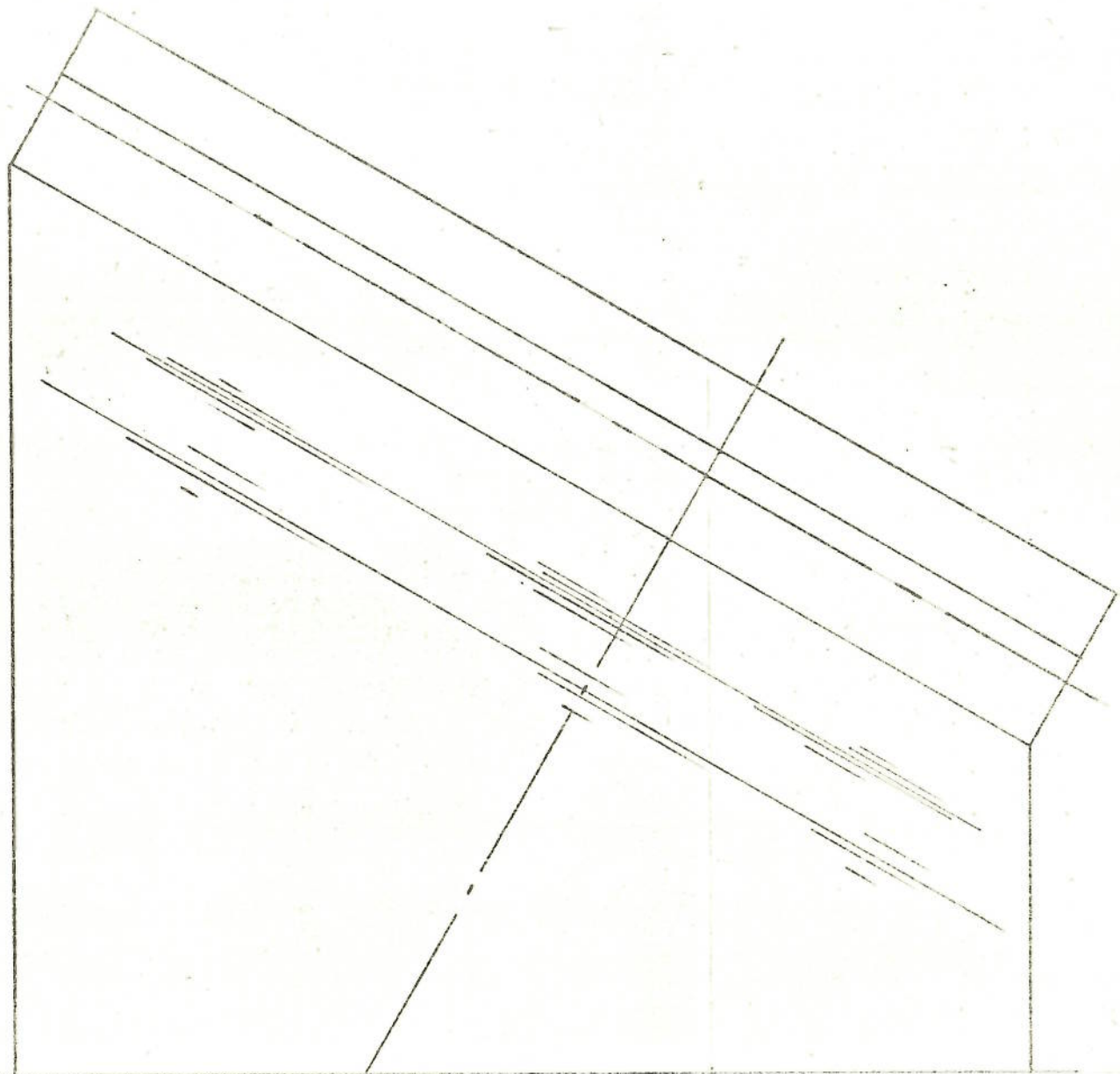


FIG. D 4 / 4 / 12.

MACHINE FRAME

D5. REPORT OF THE TOOL-POST AND TOOL-LOADING SUB-COMMITTEE.

Prepared by:- N.M. Powell
A. Mclson
G.M. Rajkumar

D5.1. Introduction.

The decision to conduct a design study on a precision turning machine with limited milling facilities, using numerical control, meant that this sub-committee would have to consider the problem of locating and clamping both turning tools and rotational tools as well as providing a drive for the latter.

From the results of the Technical Survey Report, it became clear that a large number of tools would be required for use on this machine, and it was decided that a total of eighteen tools should be catered for. These being divided into two groups, namely turning and rotational tools.

With such a large number of tools and each one being selected by command signals from the punched paper tape, it meant that either all tools would have to be contained in a multi-station turret mounted on the machine and then indexed into position when required, or stored in a magazine of some form and presented to a tool-station when required by some transfer mechanism. The latter was considered more favourable and the design study was based upon this arrangement.

The advantages to be gained by the inclusion of a copy-turning attachment to the machine were considered, both as a built-in feature and also as an accessory. The investigation showed that the advantages were small in comparison with the cost involved, and that the majority of copy-turning machines are special purpose units, and are normally used exclusively for this type of work. It is for these reasons that this committee has rejected the inclusion of a copy-turning feature on this machine.

D5.2 Objective.

To examine the requirements of the tool-post station in order that it may present to the work specified cutting tools.

To consider the best arrangement of a magazine which can store a total of eighteen cutting tools, any one of which can be presented to the tool-post, so that it can, in turn, be presented to the work for the next operation.

D5.3. Basic Requirements.

This can be divided into two sections of work,

Tool-Post Station.

This is required to accept from the magazine store single point turning tools and rotational tools and to support and present these to the work at the start of a machining operation and also to rigidly hold these tools so that they withstand the action of the cutting forces.

Tool Magazine.

As a result of information gained from the Technical Survey Report it was decided initially to design a magazine which would accept eighteen cutting tools comprizing of eight turning tools, four drills and three milling cutters with a spare position on each of the stations. At the request of the Control System Committee it was agreed to limit the number of tools to sixteen as this would enable a simpler encoder system to be used for determining the position of tools in the magazine. It was considered that with sixteen tools the machine would not be restricted in any way in its versatility to perform its range of complex machining operations.

It was considered that it should be possible to pre-load the cutting tools into the magazine away from the machine, and then fix the magazine to the machine suitably indexed so that the magazine adopts its correct orientation.

This would enable the tools required for a given component to be pre-selected and when loaded into the magazine would enable a cycle of operations on a component to be completed.

The magazine should be capable of indexing station by station, and when required present to the tool-post any tool stored in the magazine.

The position of the various tools stored in the magazine can be determined by a rotary encoder, using say a binary code.

D5.4. Survey.

Initially consideration was given to the possibility of fitting both single point turning tools and rotational tools into the same tool post socket. This was abandoned in view of the large cutting forces encountered in turning, as these forces would have to be borne by the bearings supporting the rotary cutter spindle. This would cause brinelling of the bearings and no doubt shorten their life considerably.

It was, therefore, decided to have two sockets in the tool-post station, one for tools requiring rotational movement and the other for fixed tools.

An alternative solution was also considered which provided the rotational tools with their own bearings as a complete unit. Thus the drills and mills would be fitted into individual bearing units and stored in the magazine in this way.

This would enable both rotational and turning tools to be fitted into the same socket in the tool post. In this arrangement the bearings would not be subjected to the cutting forces encountered during a turning operation.

This would have the advantage that only one station would be required on the tool-post and as a result would simplify the indexing.

But against this nine bearing units would be required. There would be difficulty in aligning the bearing units each time with the drive shaft and there is also the possibility of the bearings becoming contaminated with cutting fluid and damaged during their transit about the machine.

A sketch of this arrangement is shown in fig.D5/1. It is seen that both types of tools use a common housing, but whereas a bearing unit supports the rotational tool, the housing for the turning tool is just a clamp block. The lower sketch shows a turning tool in position for machining.

In view of the latter points it has been concluded that separate sockets in the tool-post would offer the best solution.

Cutting tests were conducted in the Machine Tools Laboratory to establish actual tool forces when machining steel on a centre lathe, and the results are tabulated as follows:-

Tool Material	-	H.S.S.
Tool Angles	-	10° x 10°
Cutting Speed	-	400 ft/min (122 m/min)
Feed rate	-	0.010 in/rev. (0.25 m/rev.)

Depth of Cut in.	m.m.	Tangential Force lbf.	K.N.	Axial lbf.	Force Ft K.N.	Power at tool point h.p.	Watts
0.125	3.22	270	1.2	100	0.445	3.3	2460
0.250	6.44	540	2.4	200	0.890	6.7	5000
0.375	9.66	810	3.6	300	1.335	10.0	7460
0.500	12.88	1080	4.8	400	1.780	13.4	10,000

It should be stated that although a cutting speed of 400 ft/min. (122 m/min) was used 100 ft./min (30 m/min) is normally recommended for H.S.S. tools, but as 400 ft./min. (122 m/min.) is the recommended cutting speed for tungsten carbide tools when machining steel, then these results are relevant as no doubt carbide tools will be used on this machine.

D5.5 Design Recommendations.

Tool-Post Station.

Separate sockets will be used to house the single point cutting tools and rotational tools. The sockets will be best positioned at 180° to each other and this will result in the most compact unit as well as providing the best clearance for both headstock and tailstock.

A sketch of this arrangement is shown in fig. D5/5/2.

The adapter for holding the rotational tools will have a taper shank which will fit in to a mating taper socket in the tool-post, and be secured by a latching and locking device. A sketch of this taper shank engagement is shown in fig. D5/5/3.

The drive to the rotational spindle will be a hydraulic motor and it will have a number of selected speeds up to 1,500 rev/min.

These speeds will be selected by command signals from the punched paper tape.

The indexing of the tool-post will be achieved by a pneumatic cylinder mounted underneath the tool-post housing driving through a rack and pinion. The shaft on which the tool post station rotates will be locked by a hydraulic clamp following the indexing operation.

Magazine.

The sixteen tool stations required can be obtained either by the use of a belt type holder or by a circular drum. Working on the basis of the tools stored at a 3 inch. (75 m.m) pitch the length of belt required will be about 50 inches (1.25 m.) and in the case of a drum about 16 inches (400 m.m) diameter. It is considered that a drum loaded with tools will be easier to handle than a belt, therefore, an aluminium drum will be used.

The tools with their adapters will be held in the drum by spring retainers and in this way the drum can be pre-loaded and then fitted to the machine.

D5.6. Design Solution. (Fig. D5/5/6)

Description of Tool Post Station.

The tool-post station is mounted on a cross-slide which moves at right angles to the work axis. The slideway, which moves on hydro-static pads, is constrained in all other axes

The tool-post is able to rotate on the cross-slide this movement being controlled by a pneumatic cylinder mounted underneath the cross-slide. The

rotational movement can be either 90° or 180° , the position being determined by the command signal to the cylinder and the position of the piston ram being detected by trip switches. In the intermediate position, the piston ram is securely held by pressure on both sides of the piston through the action of a centre position control valve.

The tool-post station will house both turning and rotational tools. The turning tools can be single point or form tools, the only requirement being that they have a common shank. The tool shank will fit into a tool holder and this in turn will fit into a socket in the tool-post.

The rotational tools can be drills, reamers, end mills, slotting mills and tool cutters up to 1 inch (25 mm) diameter. The rotational tools that have a straight shank will fit into a collet and taper shank tools will fit into a taper socket sleeve.

The drive to the rotational tools will be by a hydraulic gear type motor which is mounted on top of the tool-post station. This motor will have a speed range of up to 2000 r.p.m and a rating of 2-horse-power (1492 watts) at a supply pressure of 1000 lbf/in.² (6.9 MN/m²).

The cross-slide is actuated by a hydraulic cylinder which is mounted on the underside of the slide, the cylinder ram moves at a controlled feedrate over the length of its stroke. The rotational movement of the tool-post station is controlled by a pneumatic cylinder; the piston rod is coupled to a rack which in turn is in mesh with a pinion mounted on the tool-post shaft.

The piston rod imparts two movements to the tool-post, a 90° and 180° , the piston rod then retracts to its neutral position ready for its next command signal. It is necessary to control the movement of the tool-post in this way. For if a stopping motor were used damage to the pneumatic and hydraulic pipes supplying various units on the station would result.

At each indexing position the tool-post station is accurately located by a pneumatically operated vee register, this ensures that the tool-post takes up the position for turning and drilling operations and also for ejecting and receiving new tools from the magazine.

Following each indexing operation the tool-post station is locked to the cross-slide by a hydraulic clamp; this will ensure that the tool-post is rigidly held during machining operations.

Principle of operation.

Rotational tools can be fitted into either a collet chuck or a taper sleeve, the shank on both of these fittings is common and is a quick tool-change device known as the ACIERA. This is of Swiss origin and is a device which enables tools to be easily and quickly fitted into a driving socket, and when in position the tool shank is securely held by two locking balls.

These balls are retained by a mechanical latch which ensures a very positive drive. It is impossible for the balls to be depressed and so release the tool shank during a machining operation.

When a tool shank is presented to the driving socket, a signal is passed to a control valve which in turn operates a cylinder, the ram of which moves a slide which withdraws an inner drive spindle allowing the locking balls to drop into a recess. Under this action the spindle depresses a number of coil-springs. In this position the tool shank is fitted into the sleeve, the piston ram is released allowing the springs to slide the inner drive spindle forward and so allowing an outer sleeve to mechanically lock the balls in a driving position around the taper shank.

It is necessary for the drive spindle to move axially relative to the housing to perform this operation, and, therefore, the bevel gear which drives the spindle is fitted into outer bearings in the housing so the drive spindle is coupled to the bevel gear through the action of a spline shaft. This will allow the drive spindle axial freedom of movement but at the same time be locked to the bevel gear in a rotational direction.

The turning tools selected are of the throw-away tip type and all turning tools whether single point or form tools will have the same size tool shank. The tool shank is fitted into a holder and this in turn is fitted into the tool-post. All tools are carried in the magazine and when the appropriate tool is in position, the magazine moves forward, fitting the tool shank into the toolpost socket. A pneumatic cylinder actuates a slide and locks the tool shank in position with a wedge like action. The cutting tool is held rigidly in this position during the cutting operation.

Description of Magazine. (Fig. D5/5/7)

The magazine is a circular drum which can store a total of sixteen cutting tools, nine of these are rotational tools, and seven are turning tools. The rotational tools cover drills up to 1 inch (25 mm) dia. and 8 inches (200 mm) long, end mills, reamers, tee cutters and slotting mills and can also include any other rotary cutting tools, such as centre drills, which are required to perform machining operations on a component.

All the cutting tools are inserted into the magazine so the shank faces outwards. This means that the longest tool governs the size of the magazine, which in this case is a 1 inch (25 mm) dia. drill 8 inches (200 mm) long. The magazine, therefore, has a diameter of 24 inches (600 mm).

The magazine, which is of aluminium alloy, can be pre-loaded with cutting tools away from the machine. The total number of tools required to complete the machining operations on a given component can be loaded into the magazine; this is assuming that the total number of tools required does not exceed sixteen.

Each tool position is identified so that on a command signal from the tape, any tool can be indexed into position ready for loading into the tool-post.

The magazine when ready is loaded on to a keyed shaft, and this ensures that it will take up the same position on successive occasions. The rotary position of the magazine is determined by an encoder mounted on the slide underneath the magazine. The underside of the magazine is drilled in the appropriate positions and so acts as the rotary encoder. The sensing head is attached to the slide on which the magazine moves.

The slide is actuated by a pneumatic cylinder which has a stroke of 12 inches (300 m.m). This length of stroke is required in order to move the magazine forward so the tool shank is inserted into the tool-post and then withdrawn so that the magazine clears the projection of the longest drill.

The slide moves on crossed rollers on one side and on plain rollers on the other, this will impart a smooth motion to the magazine feed.

The indexing movement of the magazine is obtained by a pneumatic cylinder, the ram of which is coupled to a rack which is in mesh with a pinion. Coupled to the inside of the pinion is a pawl in contact with a ratchet, so that as the pinion is made to rotate, the pawl indexes the ratchet, and also the magazine, one division. The number of teeth in the ratchet corresponds to the number of tools in the magazine, so that for one stroke of the cylinder the pawl will index the magazine exactly one tool station. A number of stations can be indexed very quickly by successive strokes of the cylinder. This does mean that the magazine can only rotate in one direction, which could be rather time consuming if for example it is required to index from station number 2 to station number 16. But it is considered that suitable planning of the tools in the magazine, by arranging that for successive operations on the work the corresponding tools are positioned in similar order in the magazine, should overcome this problem.

D5.7. Sequence of Operation.

The sequence of operations that is carried out in tool changing depends upon the type of tool selected, such as turning tool, drill, or end mill, and also whether the tool, such as drill, is to be used radially or axially.

Some of the operations are common to both sequences, and so a typical example is shown for a single point turning tool and a drill to be used axially where 1 machining operation is followed by the other.

It is assumed that there are no tools in the tool-post station and that this is the first operation to be performed on the component.

1. Tool-post station moves to neutral position in line with magazine.
2. Tool-post station indexes so that turning tool socket is facing magazine.
3. Magazine indexes to bring correct tool into line with tool-post.
4. Magazine moves forward on slide so that tool shank is in position in tool post socket.

5. Pneumatic cylinder operates, locking tool shank in tool-post socket.
6. Magazine retracts to its neutral position.
7. Tool-post indexes through 90° .
8. The vee register latching mechanism is operated positioning the tool-post station.
9. The hydraulic clamp operates, clamping tool-post station to cross-slide.
10. Tool-post moves forward to take up turning position dictated by command signal from punched paper tape.

Assume now that the turning operation is complete and that the following sequence will discharge the turning tool and pick-up drill.

11. Tool-post withdraws to neutral position in line with magazine.
12. The hydraulic clamp on tool-post station is released.
13. The vee register latching mechanism is released.
14. Tool-post station indexes through 90° , so that turning tool socket is facing magazine.
15. Magazine moves forward to receive turning tool.
16. Pneumatic cylinder releases tool shank locking mechanism.
17. Magazine retracts to neutral position withdrawing turning tool from tool-post.
18. Magazine indexes to bring drill into line with tool-post.
19. Tool post indexes through 180° .
20. Pneumatic cylinder operates slide and so releases locking balls.
21. Magazine moves forward positioning tool shank in tool-post socket.
22. Pneumatic cylinder retracts slide so allowing balls to lock tape shank.
23. Magazine withdraws to its neutral position.
24. Tool-post indexes through 90° .
25. Tool-post moves forward until centre line of tool-post coincides with axis of work.

26. Tool-post indexes a further 90° , bringing the drill into line with the axis of the work.
27. The vee register latching mechanism is operated positioning the tool-post station.
28. The hydraulic clamp is operated, clamping the tool-post to the cross-slide.

Drilling is now ready to commence by either rotating the drill or by rotating the work.

The two sequences described above could be operated by a sequentially controlled circuit, and as a result it would be only necessary for the tape to issue a command signal to start the sequence of operations.

Four different sequences would cover the range of operations performed by all the cutting tools. These could be identified by a particular code, and a command signal from the tape could select the one required for a particular cutting tool and operation.

D5.8. Selection of Cutting Tools.

The case for using high speed or carbide tip cutting tools on this machine was considered and it was decided that to use carbide tools as these offered a number of distinct advantages over high speed tools which are particularly significant when used on a numerically controlled machine.

The use of carbide tools enables much higher cutting speeds to be used, they are very much harder, they have good resistances to wear by abrasion and therefore, will result in longer tool life. This last point is important when the cutting tools are to be used on an N.C. machine, in that once a programme has started is desirable that it completes its cycle without interruption. With carbide tooling there would be less chance of failure during a cycle than would be the case with high-speed tools. The higher cutting speeds obtainable would result in a shorter cycle time and lower machining cost for component.

The case for brazed tip or throw-away tip carbide tools was also considered. Brazed tip tools are compact in size and as a result would not take up a great deal of space in the magazine or tool-post station. But against this they would be much more difficult to pre-set in the tool setting fixture, the function of which would be to position the cutting tool so that the cutting edges were in the correct position in the 'x', 'y' and 'z' planes. With subsequent regrinding of cutting tools the tool holder would have to be provided with some form of adjustment in all three planes.

With throw-away tip tools the same problem does not arise to the same extent, because the tips can be placed in the tool-holder and take up a position which would be the same for successive tips. This would simplify the pre-setting of the tools. With throw-away tip tools the cost of regrinding

is eliminated, because the cost per cutting edge is so low, it would cost more to regrind the tip than to buy another one. The number of cutting edges per tip can be eight with a negative rake tip or four with a positive one.

D5.9. Copy-Turning Attachment.

The inclusion of a copy-turning attachment for the machine either as a unit included with the machine or as one supplied as an optional extra was considered. The result of this investigation is as follows.

D5.9.1. Survey.

During the past years, tracer controlled copy turning has been employed on an increasing scale in industry.

The main advantage is the simplicity of machining with a single point tool and producing contours which can normally only be obtained by elaborate form tools or multiple-tool setups on expensive and complicated machines.

One limitation to copy turning, however, is the angular presentation of the tool which introduces difficulties when operations such as the machining of both sides of the flange, or square shoulders are to be turned on shafts with decreasing diameters.

Copy turning is classified as:-

1. By the number of dimensions under control
i.e. one dimensional system has one movement under automatic control and one or two independent movements under control of the operator.
2. By method of control and actuation
 - a) Direct non-positive actuation - springs, weights, pneumatic or hydraulic pressure to maintain contact.
 - b) Electric control and actuation with further sub-divisions.
 - (1) Electric tracers with contacts, contactors and electric motors.
 - (ii) Tracers using contacts, relays, electro magnetic clutches, variable speed motors.
 - (iii) Contactless tracers, electronic amplification and field control of variable speed motors.
 - c) Hydraulic control and actuation may have either hydraulic or pneumatic stylus.
 - d) Electro-hydraulic control and actuation.

Hydraulic control and actuation

Since fluidic control and hydraulic actuation are employed to control headstock, tailstock etc. the hydraulic system is basically simple using a spool valve directly operated by a stylus and controlling the direction and speed of motion.

When the stylus is free, spring biases the spool valve towards the template and allows oil to flow to the piston rod side of the piston, thus causing the slide, which is attached to the piston rod, to move towards the template taking the tool with it. When the movement has progressed until the spool valve is in its central position the oil flow is interrupted and the movement stopped. Further movement inward, as on a slope would uncover the control ports leading to the top of the position resulting in a slide movement in the opposite direction.

The feed rate in the transverse direction is in proportion to the slope of the template and the accuracy is limited by boundary layer and contraction phenomena.

Conclusions

Most of the companies surveyed consider copying as a special function and manufacture special purpose machines for this particular requirement. Because of this fact and because the machine has been designed to produce linear stepped components, it was considered that the incorporation of a copy turning attachment could not be justified either as a built in feature or as an accessory.

D5.10. Sequential Control of the Tool-Post and Tool Loading.

Prepared by: G.M. Rajkumar
B.S. Sodhi

List of Symbols.

Symbols.

- 1 Sequence start signal
 - 2 Comparator of x-axis
 - 3 Comparator of y-axis
 - 4 Comparator of the Tool magazine
 - 5 Manual Input
 - 6 Signal from limit switch E
 - 7 Tool change signal
 - 8 Cutting sequence finish signal
 - 9 Signal from tape for tool post to index through 90°.
 - 10 Signal from unit switch E
- E
M
R
S
T
- } Limit Switches
- T Denotes the signal when the tool post has indexed to zero position.
- A signal to unclamp the tool post and clamp the tool magazine.
- B signal to clamp the tool post and to unclamp the tool magazine.

Introduction.

This report describes the design of the control circuit for the automatic loading and unloading of the cutting tools.

The tool magazine permits the use of sixteen different tools (seven stationary and nine rotational) and performs the necessary operations automatically with the help of a programme cycle punched on a paper tape as shown in fig D5/10/6. The punched tape carries the following information for automatic loading.

- (i) Tool change signal
- (ii) Nature of the tool i.e. rotational or stationary. This information is used for starting the motor of the tool post.
- (iii) Angle of indexing of the tool post according to the nature of the cutting operation.

Sequence of operation in a cycle.

The machining operation is started with the tool magazine loaded and the tool post empty.

- (i) With the start of the sequence operation signal the tool magazine indexes to present the correct tool in line with the tool post.
- (ii) The tool post indexes so that correct socket is ready to receive the tool.
- (iii) The magazine moves forward that the tool shank is in position in the tool post socket.
- (iv) Latching mechanism holding tool in magazine is released and locking mechanism securing tool shank in socket is operated.
- (v) Tool magazine now retracts to its neutral position S as shown in fig. D5/10/7.
- (vi) The tool post indexes through appropriate angle according to the nature of the machining operation. The information is obtained from the tape.
- (vii) Tool post moves forward to take up turning position directed by command from the control system.
- (viii) At the end of the cutting sequence if the tool post is at 180° , then the spindle moves to datum M fig. D5/10/7 and only then the tool retracts backwards, and if the tool post is at 90° position the spindle moves backward after the tool post has retracted.

- (ix) The tool post rotates to zero degrees at the datum point E.
- (x) The tool magazine moves forward to receive the tool.
- (xi) Latching mechanism clamps the tool in magazine and locking mechanism securing tool shank in socket is released.
- (xii) The tool magazine now retracts to its neutral position S.

The tool magazine is now ready for the next operation. The rotational signal from the tape is only used to start and stop the tool post motor.

Fig D5/10/7 shows the main elements of the machine to be actuated pneumatically by means of switches.

Pneumatic Equipment.

This consists of high pressure circuit effecting the specified motion of the machine. The normal logic circuit controls the sequence of motion of the high pressure elements such as working cylinder and step up relays etc. Some of them are built in as constructive units.

Magazine Encoder.

The binary coded pattern fig D5/10/8 would appear to be the simplest of the known methods with the added advantage that this system does not require any decoding circuit before being fed into the comparator fig D5/10/9. The tool magazine itself acts as an encoder drum and positioning is achieved by pawl and ratchet mechanism there by giving it a unidirectional indexing for each stroke of the piston. The tools could be laid out in the order of their operation to save time in indexing. But this should not eliminate the encoder as in some cases the same tool may be required for a similar operation.

Normal pressure logic circuits.

The sequence of motions is controlled by the pneumatic logic circuit. The signal emitted by the switches are used as input signals to the logic control circuit. The logic control circuits have been discussed individually for different sequential position of the machine.

Headstock Control logic (Fig D5/10/10)

The headstock moves towards H where the following signals are present

- (i) cutting sequence finish signal
- (ii) Tool change signal
- (iii) If the tool post is at 180° , then the spindle moves to datum E and only then the tool post retracts to datum E.

If the tool post is at 90° for turning operations the spindle moves to datum E only after the tool post had moved to datum H.

The headstock moves forward to cutting operation only when the comparator of the x-axis shows an error signal and the sequence start signal is on.

Tool post linear movement control circuit (Fig D5/10/11)

Tool post retracts towards E only when the following signals are present

- (i) Tool change signal
- (ii) Cutting sequence finish signal.

Tool post retracts first towards datum E (fig. D5/10/11) if it is at 90° position, otherwise the headstock moves towards its datum first for 180° position.

It moves up towards operation only when the comparator of the y-axis shows an error signal and the start sequence signal is present.

Tool post indexing control circuit (Fig. D5/10/12)

The tool post indexes anticlockwise only when

- (i) Tool post is at datum position E
- (ii) Tool change signal is present
- (iii) Tool magazine is at datum position E.

It indexes to 90° and 180° as per command from the tape

Tool magazine linear movement control logic (Fig. D5/10/13)

The magazine moves to load or unload only when the following conditions occur:

- (i) Tool post must be indexed to zero position
- (ii) The comparator of the tool magazine should show a null signal.

During unloading there must be tool change signal present.

During loading sequence start signal must be present.

Tool magazine indexing control logic (Fig D5/10/14)

Conditions for the tool magazine to index are:

- (i) Tool post must be at datum E

- (ii) Tool magazine must be at datum S.
- (iii) Tool magazine comparator shows no error.

Conclusion.

The sequential control circuit is so designed that it incorporates absolute safety and could be operated manually if necessary. Limit switches are used to detect positioning signals. These switches are readily available on the market.

D5.11. References.

- (1) Town H.C. Tracer Controlled Copying Lathes - John Wadsworth Ltd.
- (2) Town H.C.
The increasing development in the use of copy turning lathes.
- (3) Kelly, T. - John Wadsworth Ltd.,
"Project on Copy Turning"
Production Engineer 46 11 651-660 (November 1967)

Literature on Copy Turning Equipment was obtained from the following firms:-

- (1) Hepworth Iron Co. (Engineering) Ltd.,
- (2) Hallmaerone Ltd
- (3) B. Elliott (Machinery) Ltd.
- (4) T.S. Harrison & Sons Ltd.
- (5) Myford Ltd.
- (6) Charles H. Churchill Ltd.
- (7) Mimik (U.K.) Ltd.

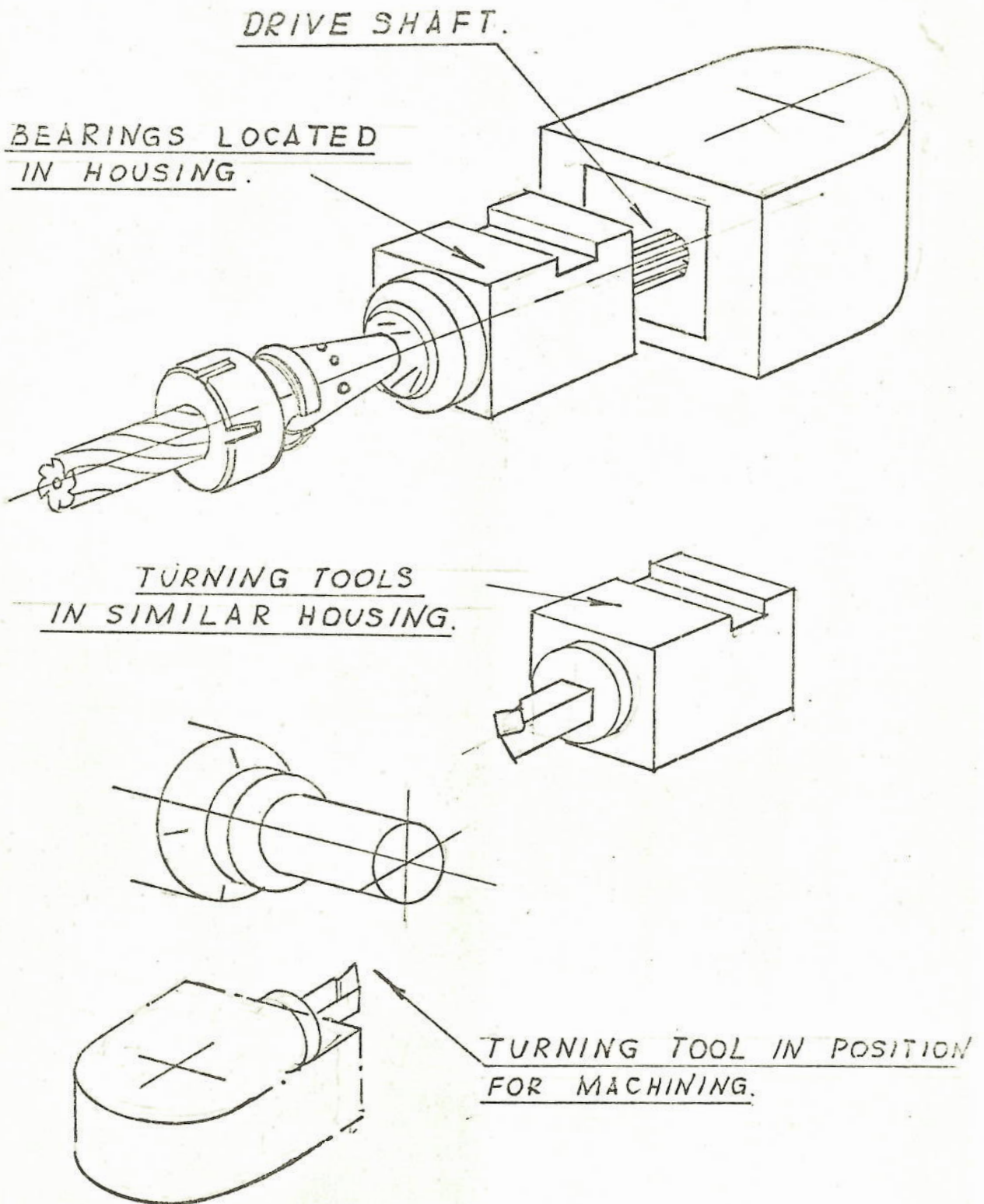


FIG. D5 / 4 / 1.

SOCKETS FOR TURNING AN
ROTATIONAL TOOLS.

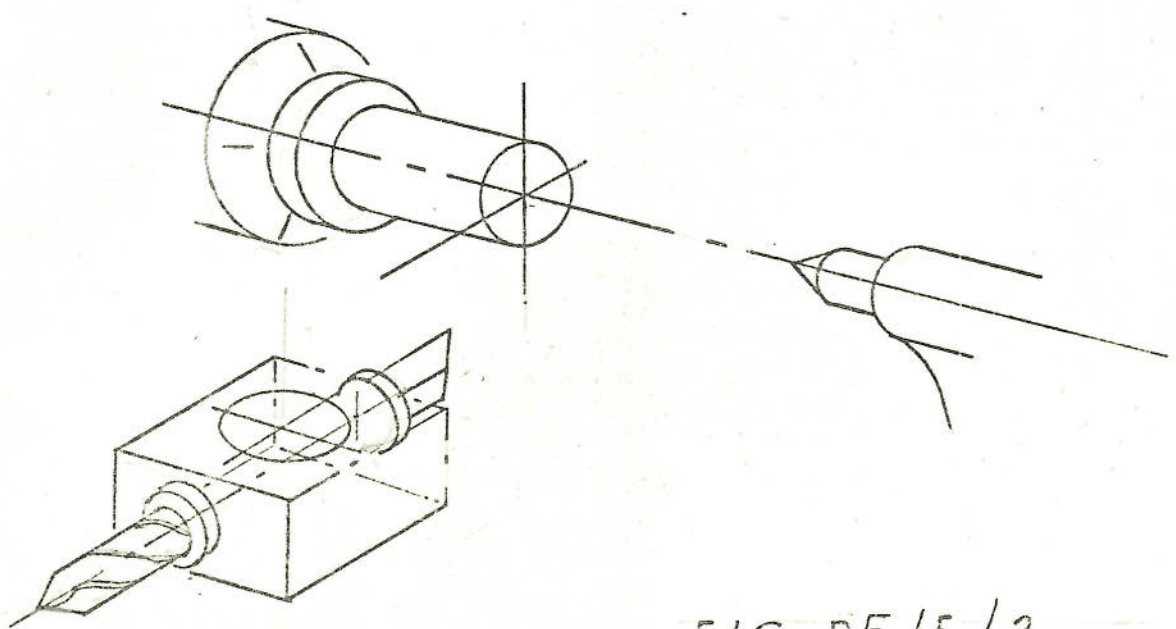


FIG. D5 / 5 / 2.
TOOL POST STATION.

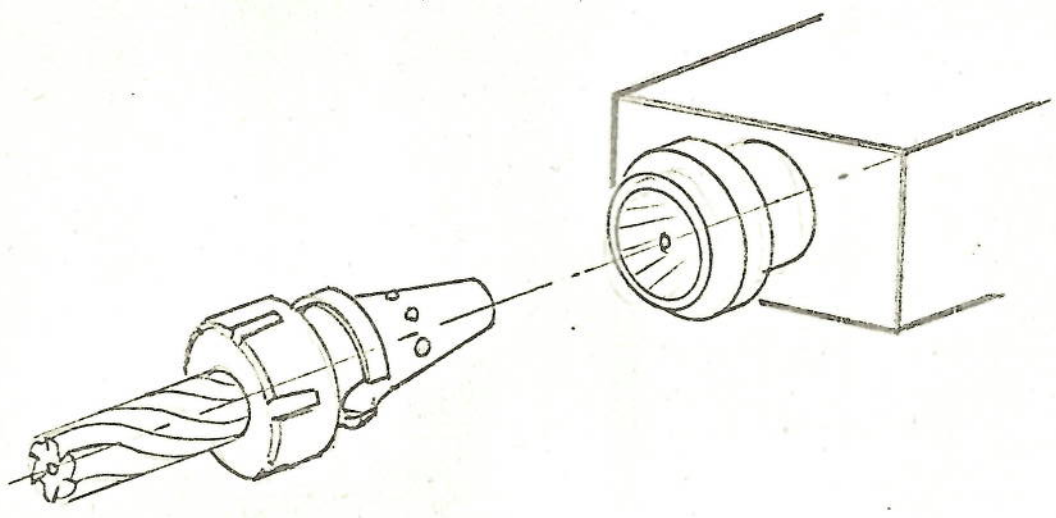
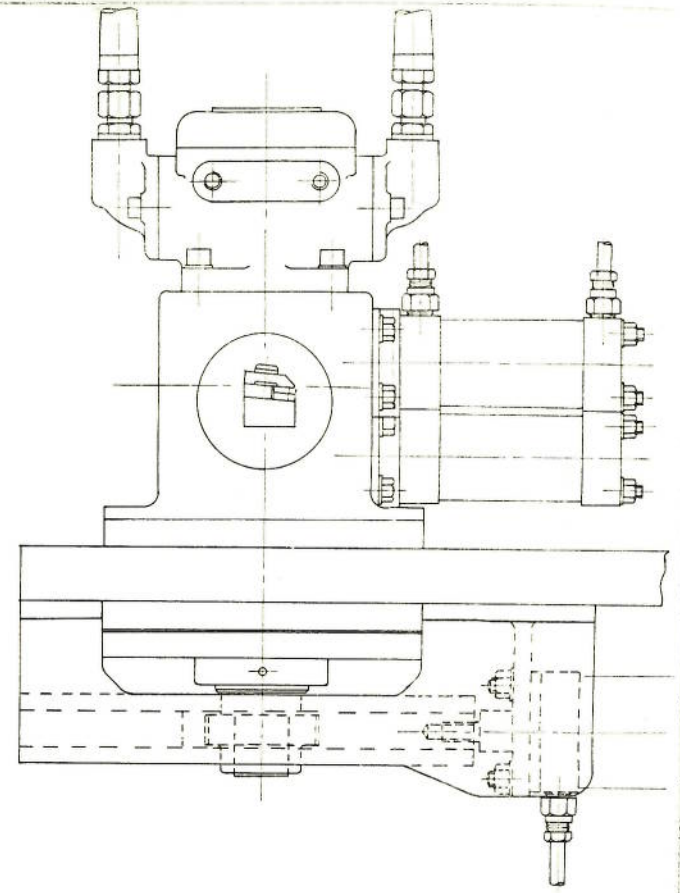
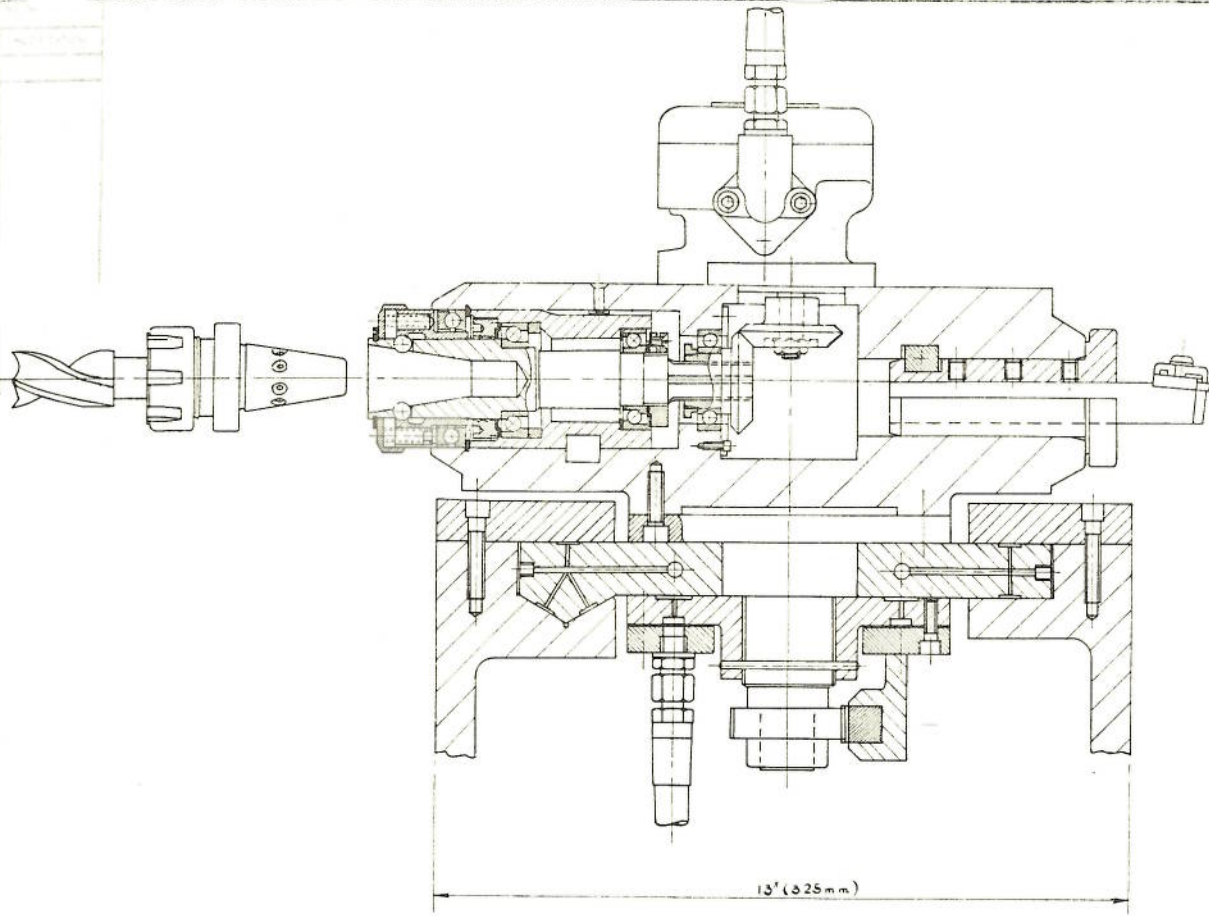


FIG. D5 / 5 / 3.
ACTERA QUICK RELEASE SOCKET.

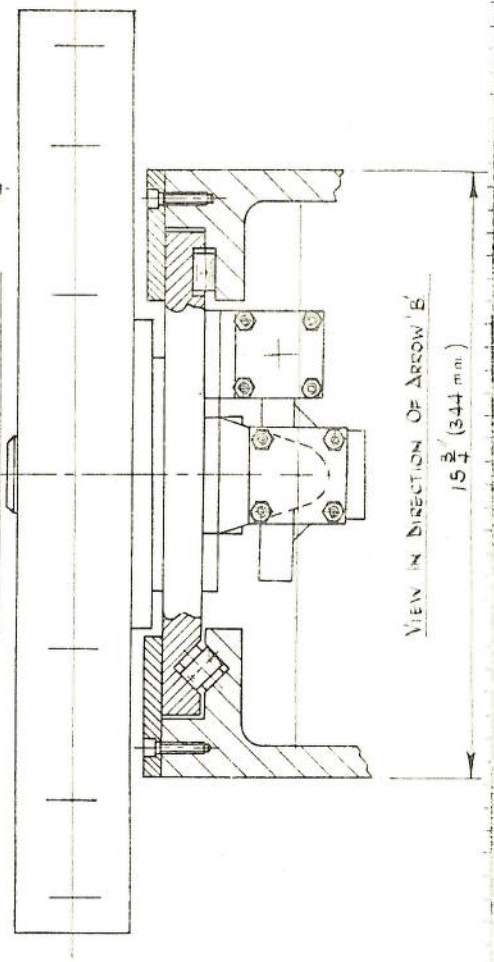
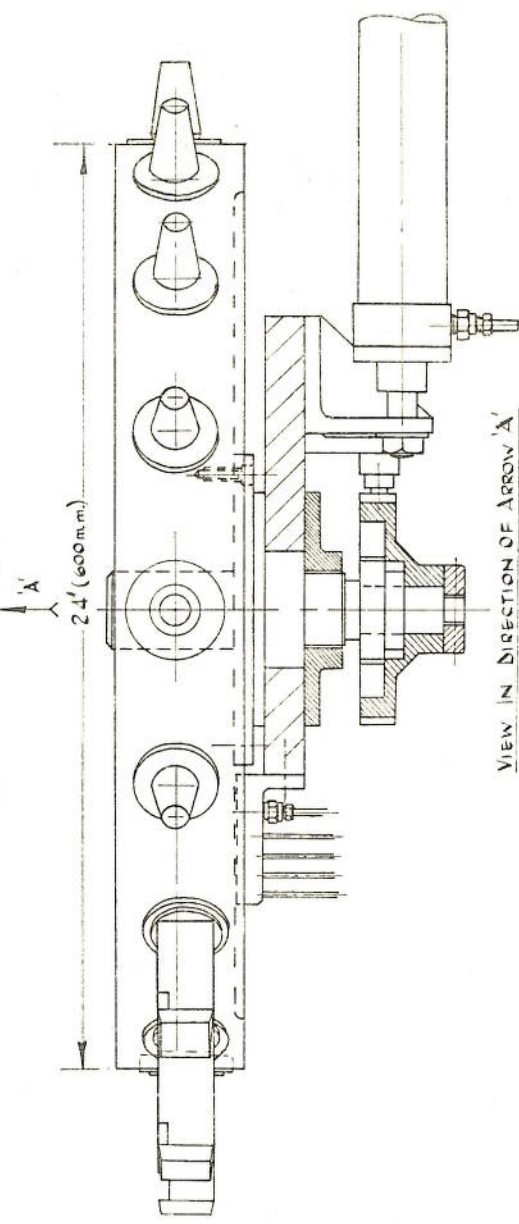
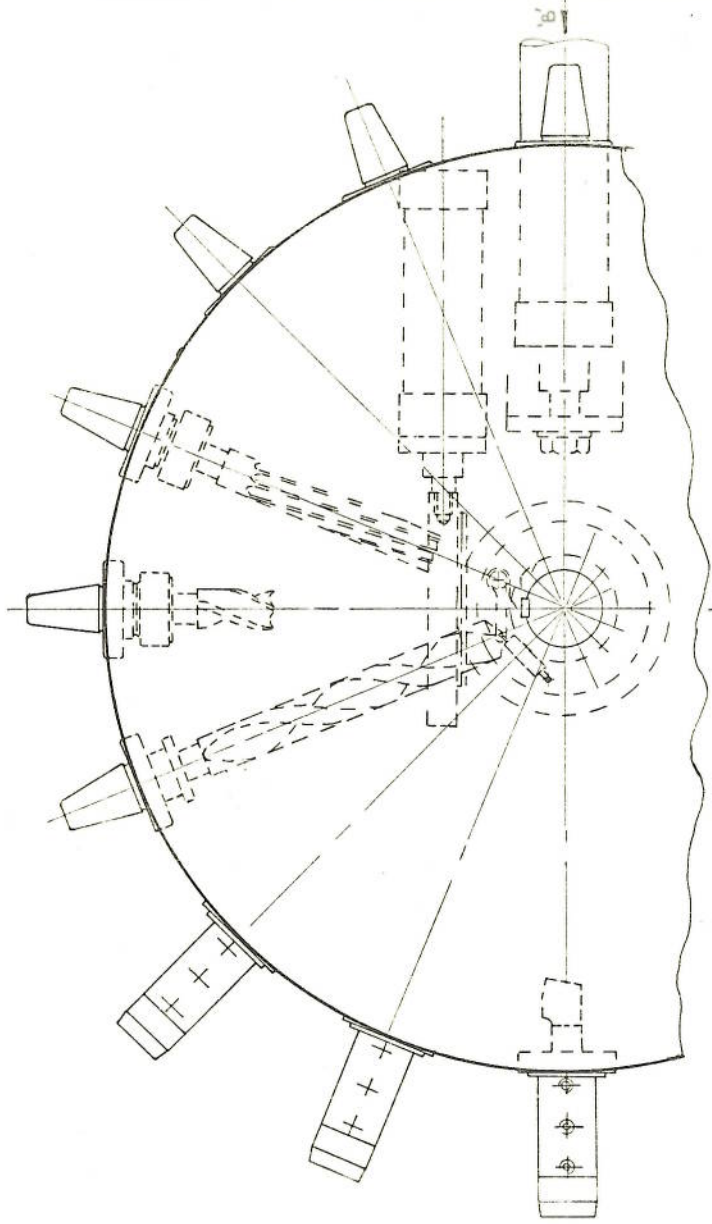


THIRD ANGLE PROJECTION

GENERAL TOLERANCES ON DIMENSIONS:	JOB. NO.	DATE	STATE	DESIGNATION	TITLE - TOOL POST STATION PRECISION TURNING MACHINE
UNLESS OTHERWISE SPECIFIED					
FINISH DIMENSIONS AS SHOWN					
ALL DIMENSIONS IN INCHES					
UNLESS OTHERWISE SPECIFIED					
				THE COLLEGE OF AERONAUTICS	22/5/e

DRAWING No.

REV. MODIFICATION.



THIRD ANGLE PROJECTION

DRAWN BY: []
 CHECKED BY: []
 DATE: []

TOOL MAGAZINE
 PRECISION TURNING MACHINE.

THE COLLEGE OF []
 []

DS/5/7

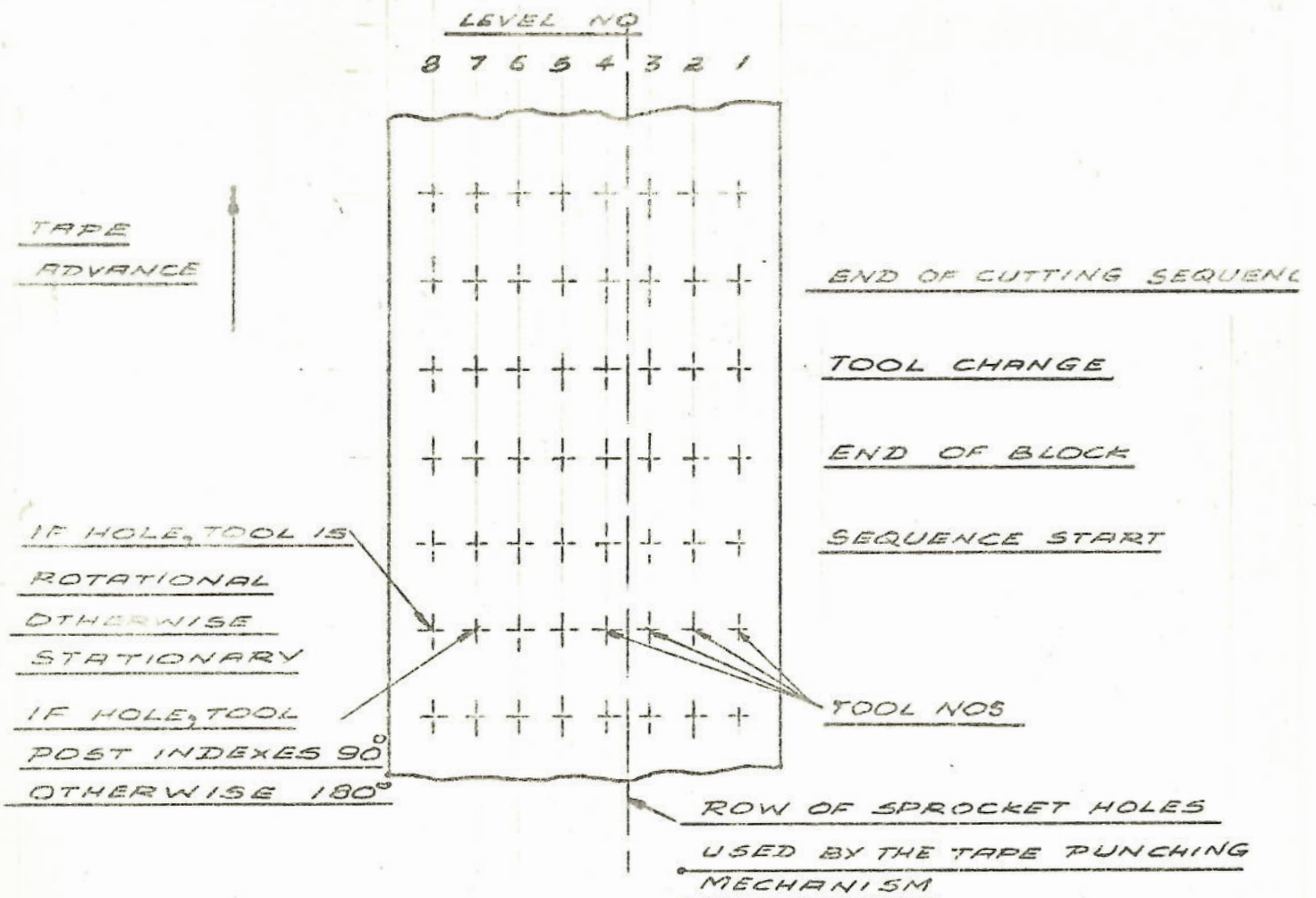


FIG D5/10/6 CHARACTER CODING FOR SEQUENTIAL
CONTROL OF TOOLING

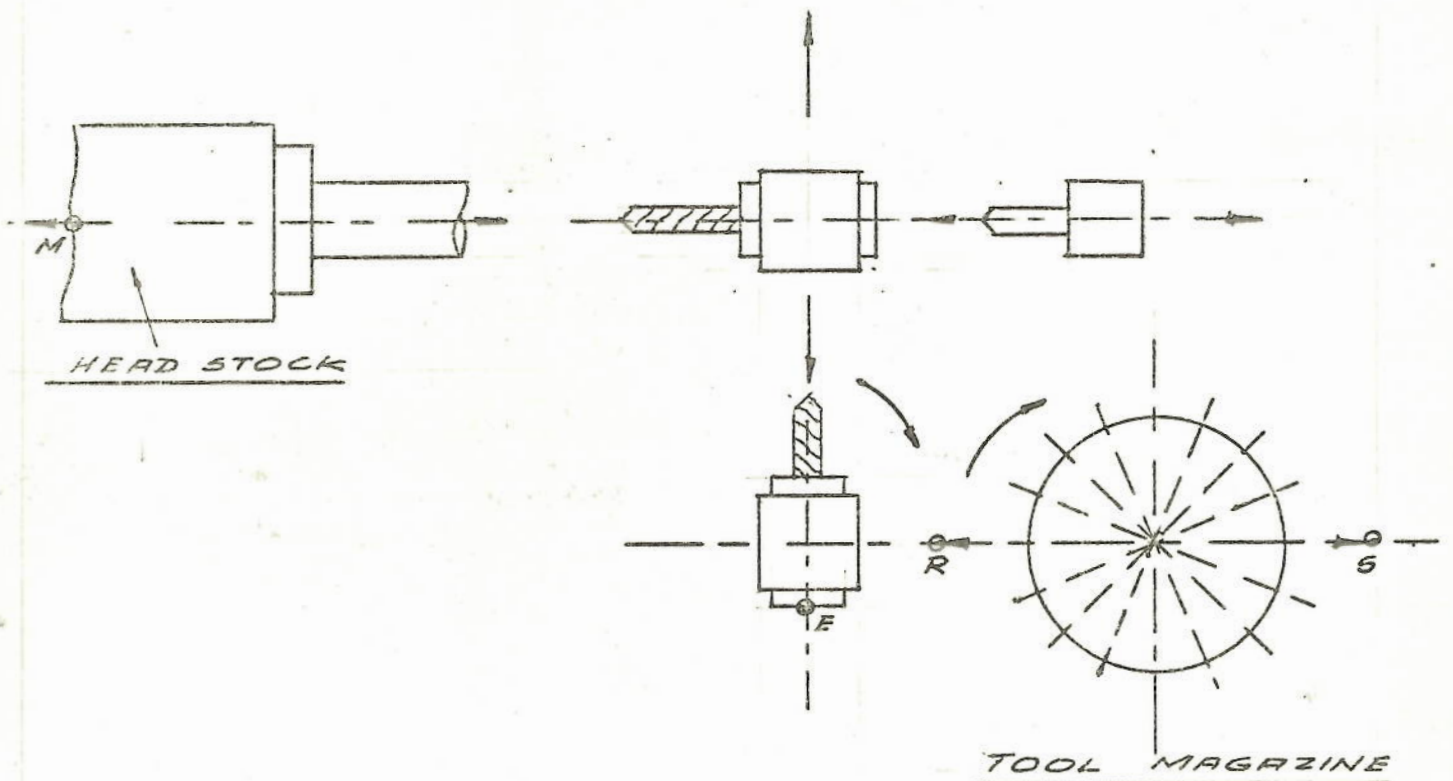


FIG D5/10/7 CONTROLLED MOVEMENTS OF THE
MACHINE

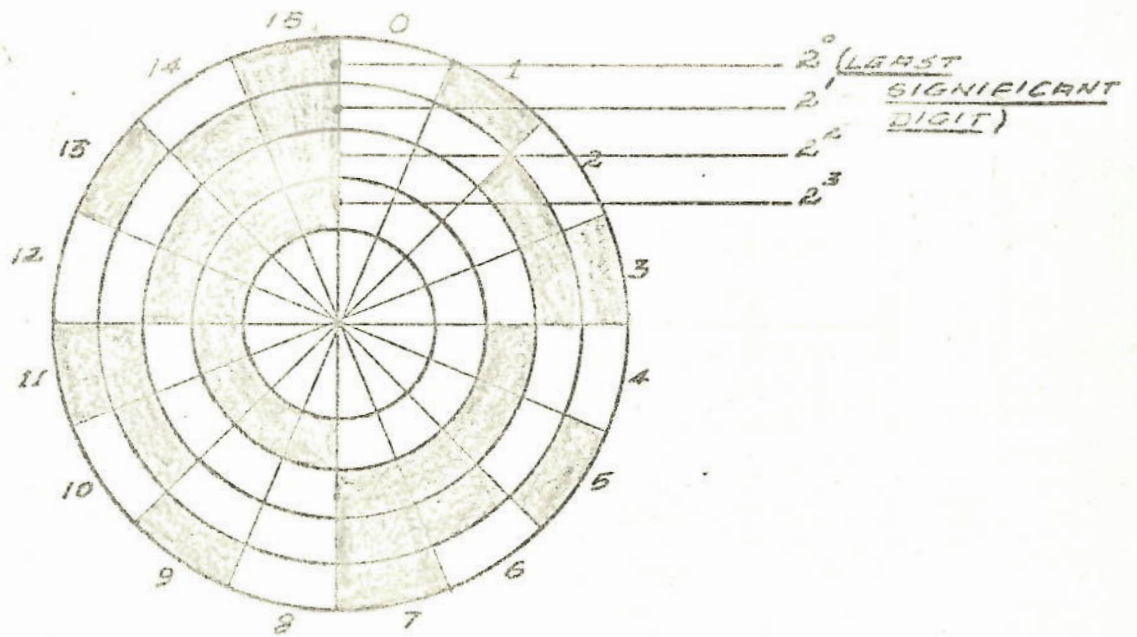


FIG D5/10/8 BINARY PATTERN

SIGNAL FROM
TAPE

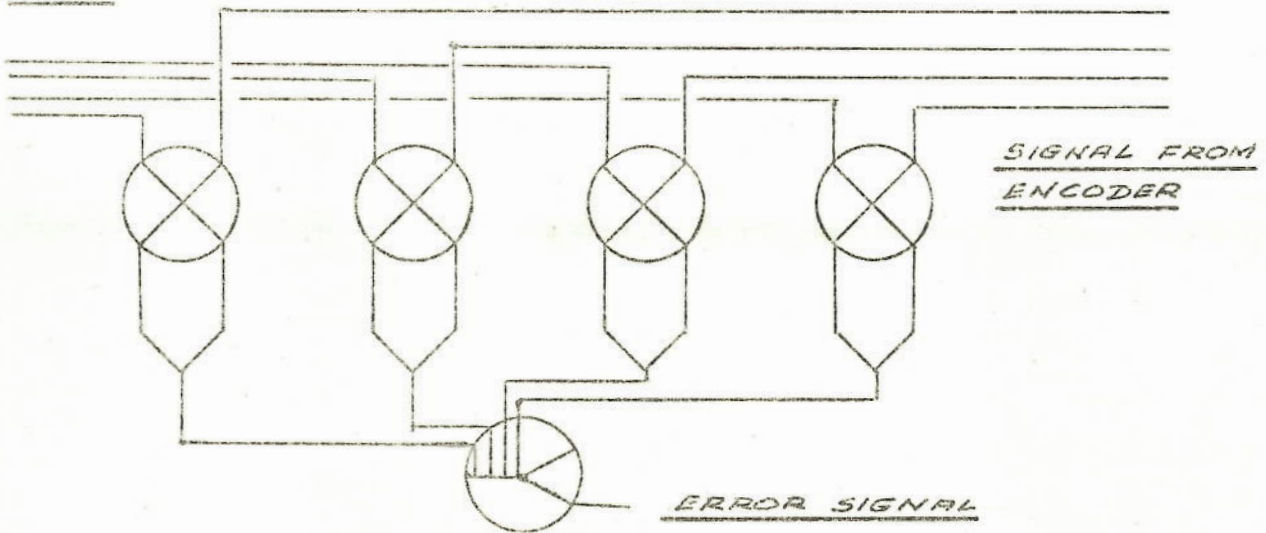


FIG D5/10/9 COMPARATOR

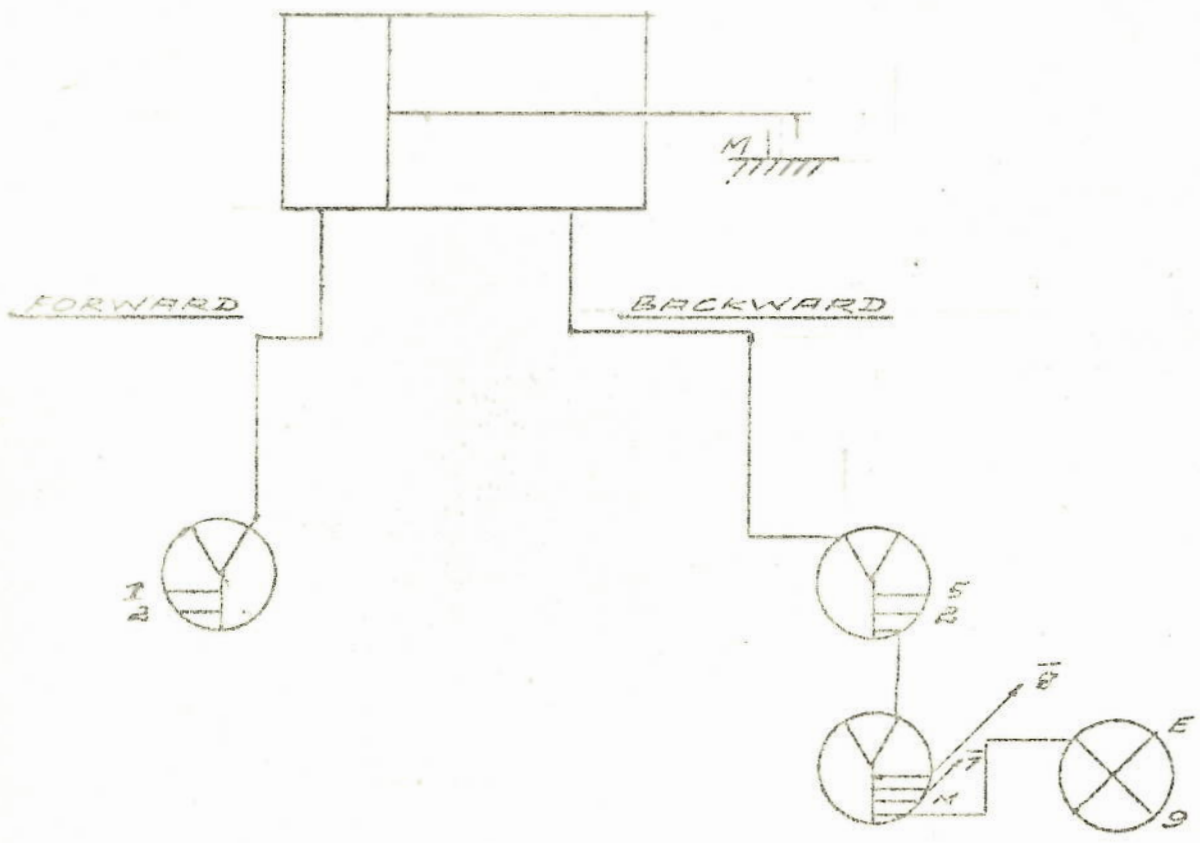


FIG. 25/10/10 HEADSTOCK MOVEMENT CONTROL LOGIC

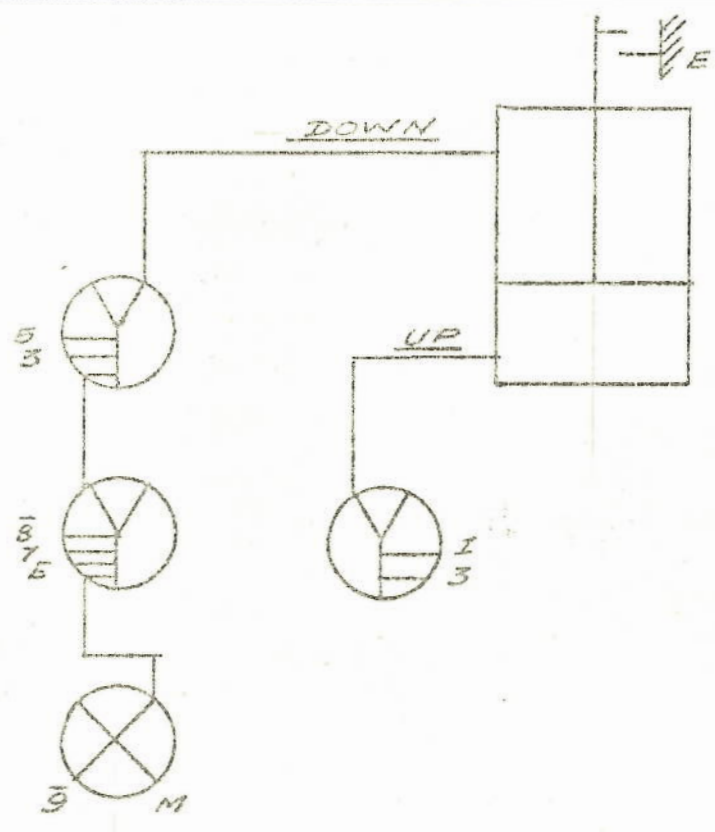


FIG. 25/10/11 TOOL POST MOVEMENT CONTROL LOGIC

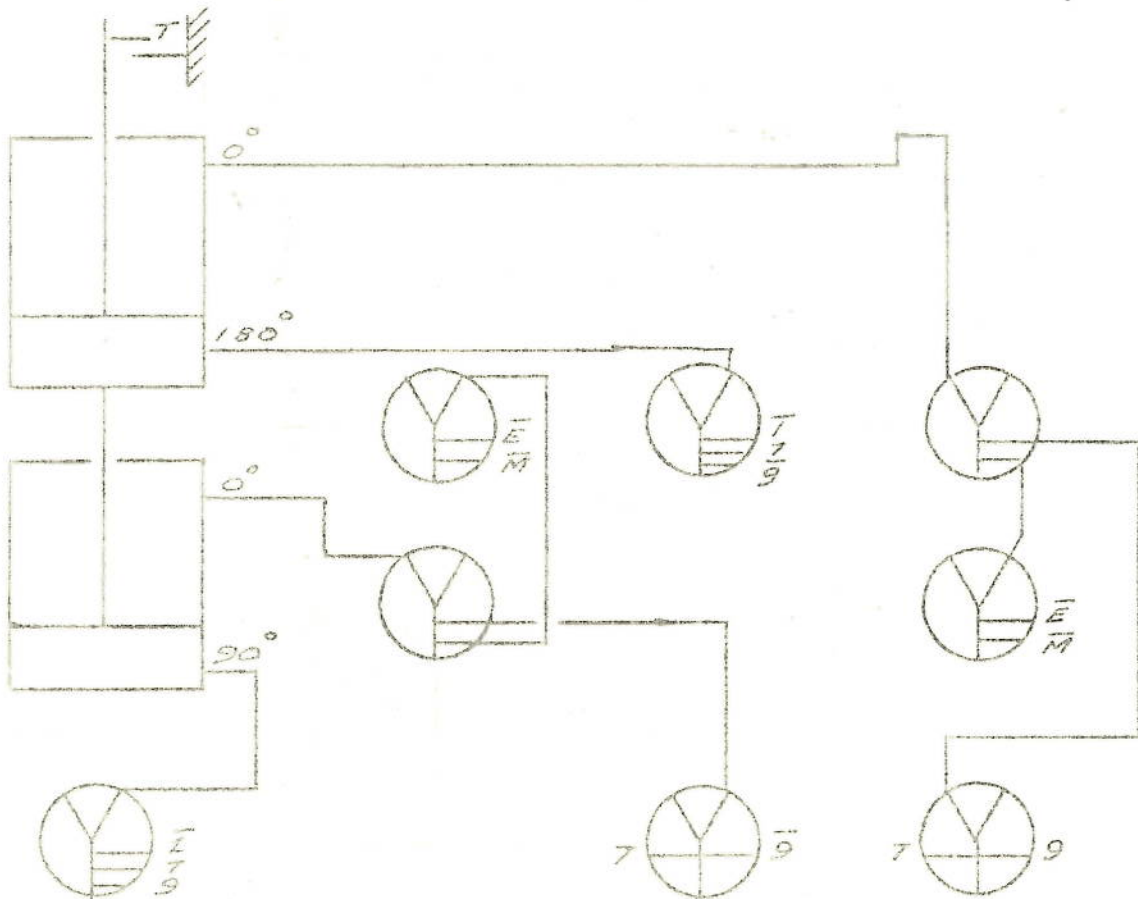


FIG D5/10/12 TOOL POST INDEXING CONTROL LOGIC

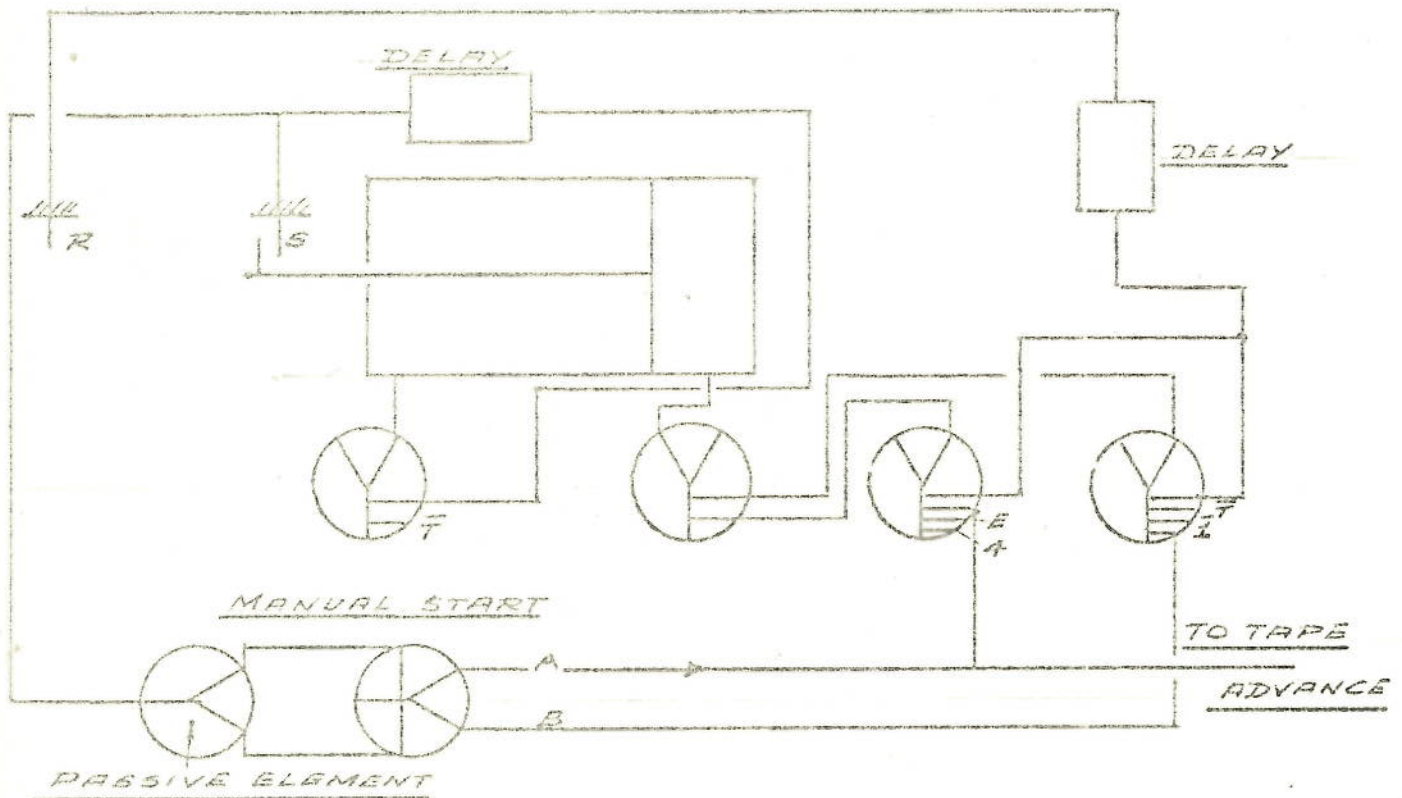


FIG D5/10/13 TOOL MAGAZINE LINEAR MOVEMENT CONTROL LOGIC

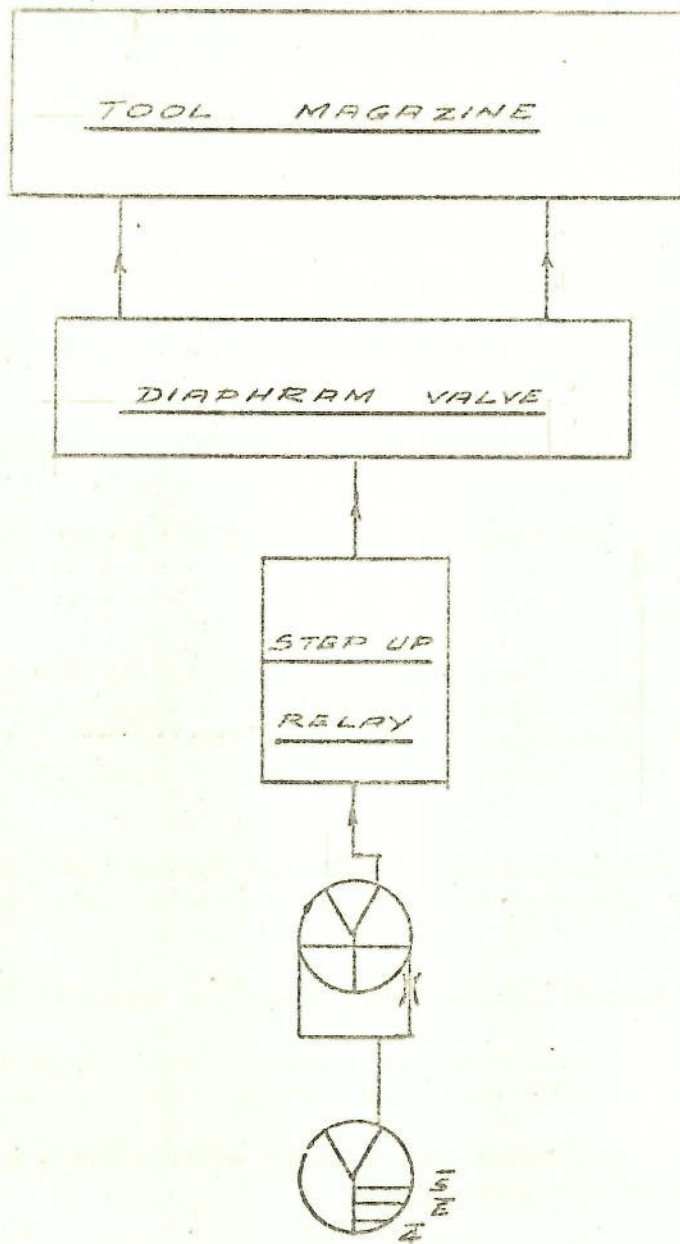


FIG. 25/10/14 TOOL MAGAZINE INDEXING CONTROL LOGIC

D6. REPORT OF THE TAILSTOCK AND MACHINE FACILITIES SUB-COMMITTEE.

Prepared by:- B.S. Sodhi
H. Bera
R.V. Jones
G.M. Rajkumar

D6.1. Introduction.

The committee's terms of reference were to:-

- (i) To submit a tailstock design that complies with the limits set by the design committee.
- (ii) To select a cutting fluid suitable for the wide range of operations to be carried out.
- (iii) To investigate swarf removal techniques and submit a finalised design.

D6.2. Tailstock Design.

The first report of the 'Configuration Committee' outlined a simple yet versatile tailstock configuration. The function of this tailstock was to provide support for the workpiece and give facility for all the operations normally carried out from a turret. In addition the tailstock was to be capable of bar feeding.

Following the conclusions of the 'Technical Survey Committee' the bar feeding capability was discarded and the turret configuration retained. Ref. Fig CP3/7.

Three methods of turret indexing were therefore investigated:-

- (i) Indexing was achieved from the straight line motion of a pneumatic or hydraulic cylinder operating through a pawl and ratchet mechanism.
- (ii) Indexing was carried out with a small motor driving a vane which acted on one of six turret beams.
- (iii) The use of a diaphragm valve for turret indexing.

Precise angular alignment of the turret for the above designs was achieved by a spring loaded plunger fitting into a slotted plate. The slots in the plate had one face radial to the centre and the second face at 30° to the first face. This has been found to be the most accurate method of location for indexing, is not susceptible to wear and provides for simple and accurate manufacture.

Subsequent investigation of the machine elements revealed that a turret tailstock gave duplication of machining stations since all the operations associated with a turret could be performed from the toolpost which possessed a sixteen tool magazine. Ref. Section D.

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The toolpost would move onto the centre axis of the machine, turn through 90° and perform turret operations. This enables a simplified tailstock to be designed with a specification as follows.

Tailstock Specification.

- (i) The tailstock should move independently and in unison with the headstock.
- (ii) It should accommodate a No. 4 morse taper.
- (iii) The force exerted by the centre should be variable between zero and 400 lbf (1.77 KN) Ref. D6.5 Appendix

Design Solution.

The tailstock body consists of a simple casting which incorporates the hydrostatic slideways, Ref. Drawing D6/2/11. The actuation ram is secured to the body by a steel bracket, which will withstand shock loads far better than a flange thrown out from the casting.

The tailstock is shown with a revolving centre which can be removed from its taper by a blow on the ejector cap. Ejection of the centre could have been achieved by continuing the centre hold right through the body and striking the end with a rod. However this would have complicated the hydrostatic oilways considerably. A conventional slot and drift was undesirable since the drift could be driven into the actuation cylinder. Special drifts preventing this were considered unsuitable, because they would soon be lost in a machine shop and a conventional drift used.

The solution was an integral ejector which would be a little more expensive than the alternatives but have none of their disadvantages. A dowel was used to guarantee the correct relationship between the centre and ejection cone.

The variation of the force to be applied by the tailstock between zero and 400 lbf (1.77 K.N) was achieved by placing a relief valve in the actuation cylinder supply line. This valve will be set by the machine setter to a value obtained from the maximum cutting conditions for a given component by using the formula in Appendix 1.

In practise a nomogram or disc calculator etched on a metal plate could be fixed onto the machine to facilitate the calculation. However the setter would probably use his experience to set the relief valve and this could well be sufficiently accurate.

D6.3. Cutting Lubricant.

For the purpose of selecting a cutting lubricant the following conditions have been set which correspond to the specification and function of the project machine.

- a) Tools to be used (i) High speed steel
(ii) Tungsten carbide
(iii) Ceramic
- b) Speed range 100 to 3000 rev/min
- c) Feed 0.001 in. to 0.010 inch/rev (0.0254 mm to 0.2540 mm/rev)
- d) Depth of cut 0.250 in. (max) (6.23 m.m. max)
- e) Surface finish 16.0 C.L.A. obtainable under optimum conditions.
- f) Work materials to be used - Aluminium, Magnesium, Brass, mild steel, Cast Iron and alloy steels.

Function of cutting fluids.

The widely varying conditions found in turning and milling make it extremely difficult to establish rules to cover the selection of cutting fluids for each operation. Heat treatment of work material, tool material, cutting speed, depth of cut and feed affect the choice of a suitable cutting fluid.

In considering cutting fluids for individual operations, the following increasing order of severity of machining operation is generally accepted. Ref. 2.

- (i) Drilling
- (ii) Automatic lathe machining
- (iii) Milling

The severity of cutting operations is usually considered to be an indication of lubricant requirements and this should be considered in conjunction with the toughness and hardness of the material to be machined with the depth of cut being taken.

Cutting fluids have better lubricating properties and extreme pressure qualities in the following order:-

- (i) Water solutions
- (ii) Soluble oil emulsions

- (iii) Mineral oils
- (iv) Mineral lard oil
- (v) Fatty oil
- (vi) Oil containing sulphur
- (vii) Oil containing sulphur and chlorine

In all cases of fluid containing oils including emulsions an increase in the fatty oil content improves the lubricating properties and in the case of soluble oils, additives to the base oils such as sulphur, materially have the same effect.

Apart from the special properties mentioned above, the cutting fluid must have general properties. Should not cause distortion, staining, smoke or fog in use, objectionable odour, decompose chemically, contain ingredients harmful to the operator and retain its general properties.

Due to the widely varying conditions found in the machine tool operation, it is very difficult to select a single cutting fluid which would provide for all the requirements of the project machine. However the selection of cutting fluid is governed by the system of application. For the "Hi-jet" system (Ref.4) used, it is appropriate to use low viscosity, water emulsion oil and the 'Gulf Hi-jet' oil is best suited for this requirement which is easily available in this country.

System for the cutting fluid application.

The ideal method of applying a cutting fluid is by jets carefully directed and at sufficient pressure to ensure that the fluid has the best possible chance of reaching the cutting edge. In addition a copious flow of fluid at low pressure should flood the tool and workpiece to cool them. The importance of properly directed nozzles and adequate flow cannot be over emphasized.

The 'Hi-jet' system Ref. 4, Fig D6.3.1, of cutting oil application has been claimed as a revolutionary advance in metal machining practise (Ref. 3) Longer tool life is said to be obtained during turning operations and it has also been applied to other operations such as drilling and milling.

The system involves the projection of small jet or jets of a special neat oil at a high pressure of 400 lbf/in² (Fig D6.3.4) into the clearance between the tool and the work. The jet is aimed directly at the cutting edge from below (fig D6.3.2) In addition an oil curtain (fig D6.3.3) and overhead flood of oil are provided for the purpose of reducing fumes. The system of jets is in direct contrast with the conventional method of cutting fluid application in which a much larger quantity of fluid at low pressure is used to flood the tool from above. The increased tool life (Fig D6.3.4) is attributed to improvements in both lubrication and the cooling attained near the cutting edge of the tool. At normal cutting speed the increase in tool life is claimed to be seven to eight times. While at higher speeds or on

tough materials an improvement in tool life of twenty to thirty times is achieved.

It is possible then when changing from turning to milling the "Hi-jets" stream will not exactly coincide with the cutting zone. However in this case the oil curtain will provide lubrication and the Hi-jet striking work or cutter will provide a spray mist inside the oil curtain.

A spray mist cutting lubricant system could easily be applied to the machine if rapid and frequent changes of cutting fluid are necessary to suit different materials. The reason for spray mist lubrication consists of a small bottle which can be changed in less than a minute.

Recommendations.

It is recommended that the "hi-jet" system of cutting oil application is applied to the project machine and the approximate cost of the whole system as a unit with pipe fittings reservoir and other accessories would be £50.0.0.

D6.4 Swarf Removal.

It is essential that swarf is removed from the machine if it is to function correctly. Retention of chips in milling cutters will cause breakage and with turning, milling, drilling and reaming operations swarf at the cutter will mark the component. However the magnitude of this problem is difficult to estimate and its solution can only be based on experience.

The form of swarf, i.e. dust, chips or continuous can influence the problem greatly. Therefore tooling is an important variable and chip breakers for all types of operations will help to alleviate the problem.

High rate automatic machines produce copious volumes of swarf and have no operator present to remove it. It is therefore usual to delegate someone to remove swarf from a number of machines with the consequent increase in running costs.

Survey.

Swarf removal can be considered in two parts.

- a) Removal of swarf from the cutting zone.
- b) Transport of swarf from the machine tool.

a) is often achieved by the cutting action itself. A tangential cutting operation such as milling or turning will assist the removal of swarf by ejecting the chip at a relatively high velocity. This particular feature is used in pneumatic removal systems to impart momentum to the chips in the duct

which normally surrounds the cutter. For other operations the designer has to provide a means of deflecting the swarf in the desired direction.

Drilling and tapping are less helpful as the chips are not ejected axially at high velocity. However swarf can be washed blown or sucked from the work-piece. This often involves the use of an additional station, where the component and swarf are separated.

The problem of swarf build up on the machine surfaces can be minimised by ensuring that the surfaces are inclined wherever possible with an adequate flow of fluid. Fulmer Research Institute Ref. 7, carried out an experimental programme to study the effects of machine bed surface finish and slope. It was shown that minimum fluid velocity required to keep machine surfaces swarf free varied between 1 and 2 ft/sec (914 mm/sec).

- b) Transport of swarf from the machine could be achieved in three ways:
 - i) Hydraulic:- This method uses either a velocity trench or an ejector system.

Velocity trenches are suitable for removing all types of swarf except large entwined bundles. A stream of cutting fluid is forced into the cutting area and flushes the swarf into the trench. Ideally the fluid must transfer all the swarf away from the cutting area and keep it in suspension until the lubricant has been drained off.

A variety of trench designs are available, the most commonly used being a U-section shape with a semi-circular base. The U-section prevents the swarf being built-up at the bottom. A combination of a trench slope of 0.25 to 0.375 in/ft. (.2 to .3 cms/metres) and a series of coolant flushes keep the swarf in motion. The optimum length of the trench depends on maximum depth permissible and the form of the swarf.

Ejector systems are mainly applied to very fine chips. The swarf with coolant is collected in an ejector by inducing the mixture into a stream of clean coolant. The mixture is then conveyed to a collection tank through piping which must be sloped towards the tank. To avoid sludge deposits occurring when the system is shut down. The ratio of the clean conveying coolant to the contaminated coolant is three to one.

- (ii) Pneumatic:- Fine swarf dust and small chips can be removed by means of a high velocity air jet which sweeps the swarf from the cutting area into a chute. The swarf and dust can be exhausted pneumatically from the cutting zone by fitting an individual head over the cutter. The essential features of pneumatic methods are:-

- a) Machine hoods
- b) Ducting from the machine to the separation unit.
- c) The separation unit.

a) The hood should be designed so that the swarf falls into them. Hoods can be made to envelop the cutter almost completely.

b) Ducting from the machine to the separation unit should be located as soon as possible to the cutting zone so as to reduce the length of the duct work.

c) The function of this unit is to remove all the swarf and the dust from the air. Air carrying the dust is passed through a water spray to wash out the fine particles. A drag conveyor then removes the swarf from the bottom of the separator and the air is discharged through a series of moisture eliminators. This type of separator is known as a wet separator.

In a dry separator, the first stage is in the form of a cyclone, which removes the swarf and fine particles. The fine dust is removed in the second stage which can be a self cleaning oil pad or a bag collector.

(iii) Mechanical:- This involves brushes, scrapers, conveyors and similar equipment. These have been discussed in this section.

Tube conveyors :- (fig D6/4/5) are used for small broken chips. A particularly useful application for this type of conveyor is the removal of swarf from the inside of a machine if the machine design will not permit the gravity removal.

Pusher Conveyors:- These are of two types:-

- a) Paddle hinged type
- b) Tins type

a) This type has paddles hinged to the pusher bar which push the swarf forward on the power stroke and retracts parallel and adjacent to the pusher bar on the return stroke.

b) The reciprocating pusher bar propels the swarf in the trough by tins. The tines (fig D6/4/6) fixed in the trough hold the swarf during the return stroke of the pusher bar.

Belt conveyors:- (fig D6/4/7) The conveyor is made of porous material to separate the cutting fluid from the swarf.

Rotating Bar type conveyor:- (fig D6/4/8). The swarf is collected in any of four boxes which discharge into the swarf bin in their course of rotation. This type of conveyor is suitable for all types of swarf.

Rotating Bar Type Agitator:- The agitator is a rectangular bar rotating about an axis normal to the plane along which the swarf moves (fig. D6/4/9)

The bar rotates at a constant speed in order to give the swarf a tangential velocity and thereby providing an initial momentum to the chip and sweeping the plane clear of the chips. The continuous motion prevents them from building up on the plane. This is suitable for both broken chips and large bushy swarf.

Tilting Plane Type Agitator:- The forces causing the movement of the swarf include the gravity and the momentum of the particles. In addition to these a mechanical agitation of the swarf is obtained by a sudden increase of the slope of the plane along which the swarf travels.

Pneumatic Agitator:- Ifig D6/4/10. In this a jet of air provides an initial motion to the swarf. This type of agitator is suitable only for fine chips.

Design Solution.

As the machine design provides gravity swarf removal facilities, it was considered that adequate swarf removal could be achieved by a simple yet cheap pneumatic agitator. (fig D6/4/10). Such an agitator fits easily into the frame design occupying a minimum of space and supplements the swarf removing action by the cutting lubricant washing down the slope of the bed.

Other types of swarf removing techniques are complicated, expensive and occupy more space. They need a separate drive or a gear reduction box.

Pneumatic agitator logically conclude to our choice. Sequencing of the agitator could be achieved with work loading.

D6.5. Appendix - Calculation.

Maximum Force to be exerted by the Tailstock.

The maximum force to be exerted by the tailstock was calculated as follows:-
(Ref. 1)

$$\text{Force lbf} = \text{Depth of cut (inches)} \times \text{Feed (inches/rev)} \\ \times 150,000 \times x$$

For workpiece weights up to 440 lb (1.95 K.N.)

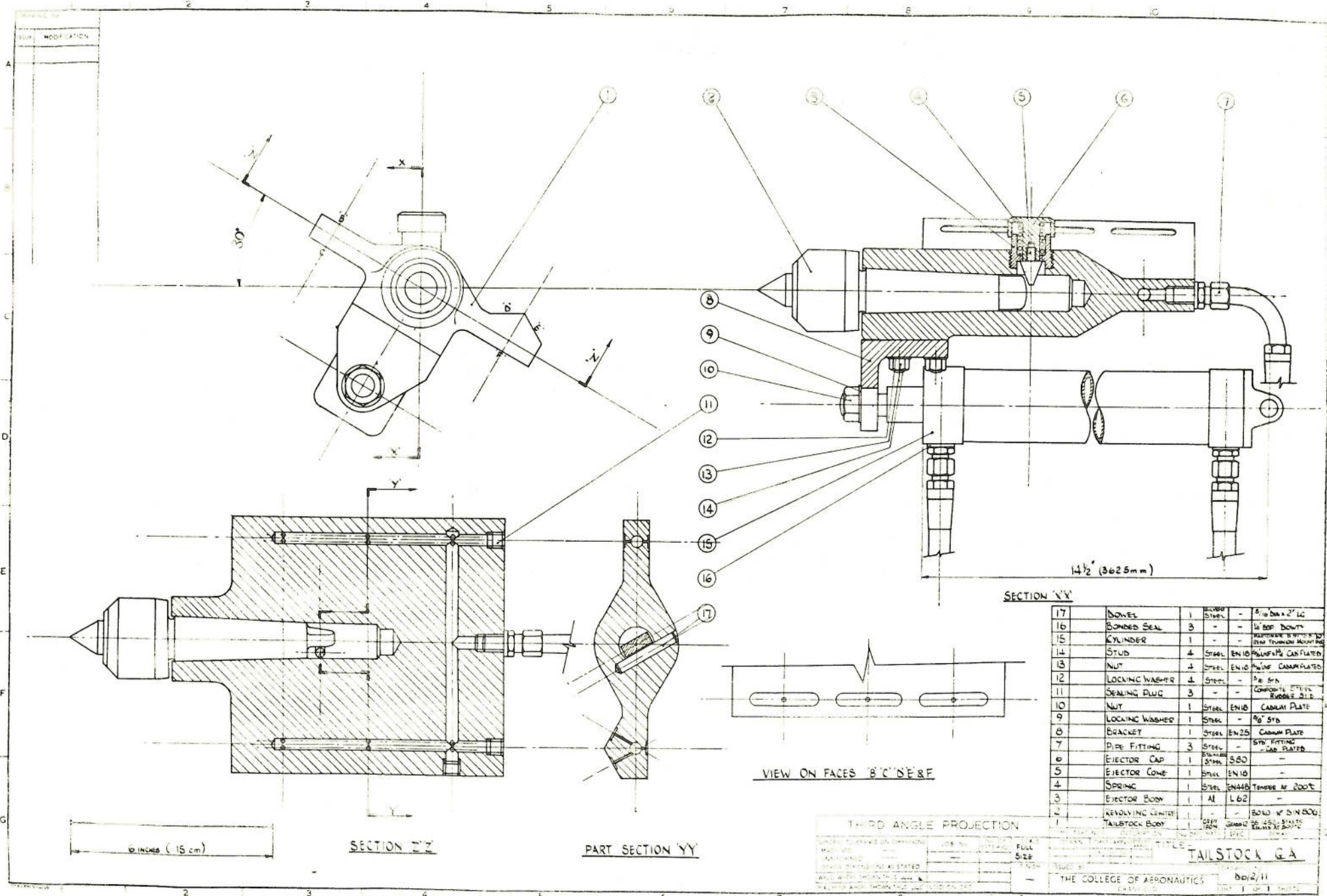
$$x = 0 \quad \text{and for weights over this value} \\ x = \frac{\text{weight}}{2}$$

Therefore for the machine $x = 0$

$$\begin{aligned} \text{Maximum Force required} &= 0.25 \times 0.010 \times 150,000 \\ &= 375 \text{ lbf say} \\ &= \underline{400 \text{ lbf (1.77 K.N)}} \end{aligned}$$

D6.6. References.

1. - "Precision Tailstocks"
Stanton Machine Tools Co. Ltd
2. - "Cutting Fluids"
Esso Petroleum Co. Ltd.
3. Pigett, R.J.S.
Colwell, A.T. - Hi-jet system for increasing tool life
4. Penn Report - Hi-jet system of cutting fluid applications
5. Henderson, G.S. - Swarf Removal - The Machine Tool Designers
Problem
(James Archdale & Co. Ltd.
M.T.I.R.A.
6. Abbot, A.C.
Thorp, H.
Griffiths, R. - Swarf Handling at Ford of Britain
Ford Motor Co. Ltd. M.T.I.R.A.
7. Schomburgke, I.C. - Research on Swarf Removal System.
(Fulmer Research Institute Ltd)
M.T.I.R.A.



SECTION XX

17	BOWEL	1	BRASS	-	5/16 DIA x 2" LG
16	BONDED SEAL	3	STEEL	-	1/4 BOP DOWN
15	CYLINDER	1	STEEL	-	TRANSVERSE BUSHING FOR OVER TENSION MOUNTING
14	STUD	4	STEEL	EN10	PHENOLIC CAP PLATED
13	NUT	4	STEEL	EN10	PHENOLIC CAP PLATED
12	LOCKING WASHER	4	STEEL	-	PHENOLIC CAP PLATED
11	SEALING PLUG	3	STEEL	-	PHENOLIC CAP PLATED
10	NUT	1	STEEL	EN10	CADMIUM PLATE
9	LOCKING WASHER	1	STEEL	-	90° STD
8	BRACKET	1	STEEL	EN25	CADMIUM PLATE
7	PIPE FITTING	3	STEEL	-	STD FITTING
6	EJECTOR CAP	1	STEEL	S60	-
5	EJECTOR CONE	1	STEEL	EN10	-
4	SPRING	1	STEEL	EN48B	TENSILE AT 200°
3	EJECTOR BODY	1	AL	L62	-
2	REVOLVING CENTER TAILSTOCK BODY	1	STEEL	EN25	BUILD UP IN 1500; 1/4" DIA. BUSHING 1/4" DIA. BUSHING
1	TAILSTOCK BODY	1	STEEL	EN25	BUILD UP IN 1500; 1/4" DIA. BUSHING 1/4" DIA. BUSHING

VIEW ON FACES B'C'D'E&F

THIRD ANGLE PROJECTION

NO.	DESCRIPTION	DATE	BY	CHKD.	APP'D.	SCALE	PROJ.	TEMP.
1	DESIGNED					1:1	3RD ANGLE	
2	DRAWN							
3	CHECKED							
4	APPROVED							

TITLE
TAILSTOCK GA
THE COLLEGE OF AERONAUTICS
Box 2/11
1 of 1

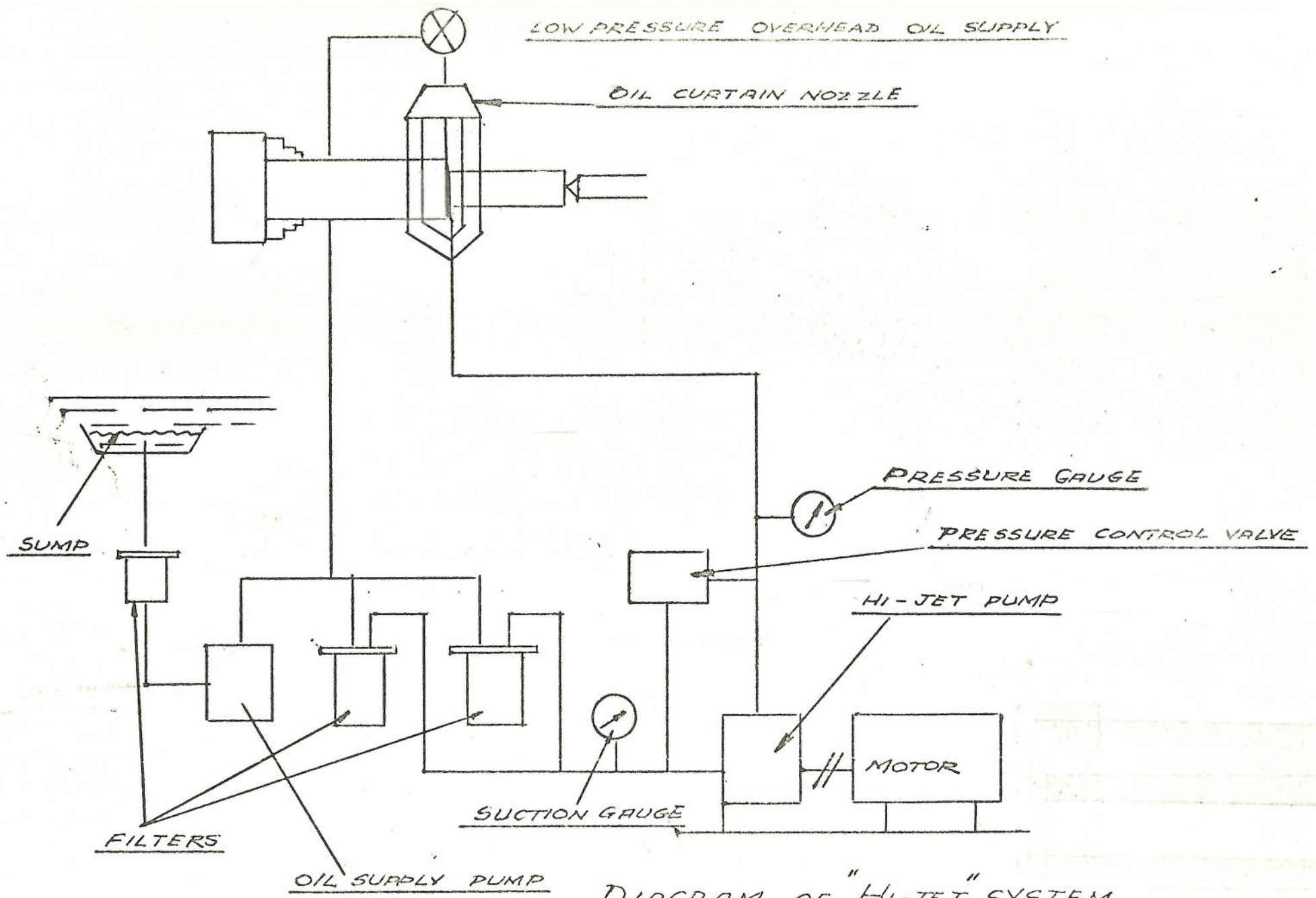
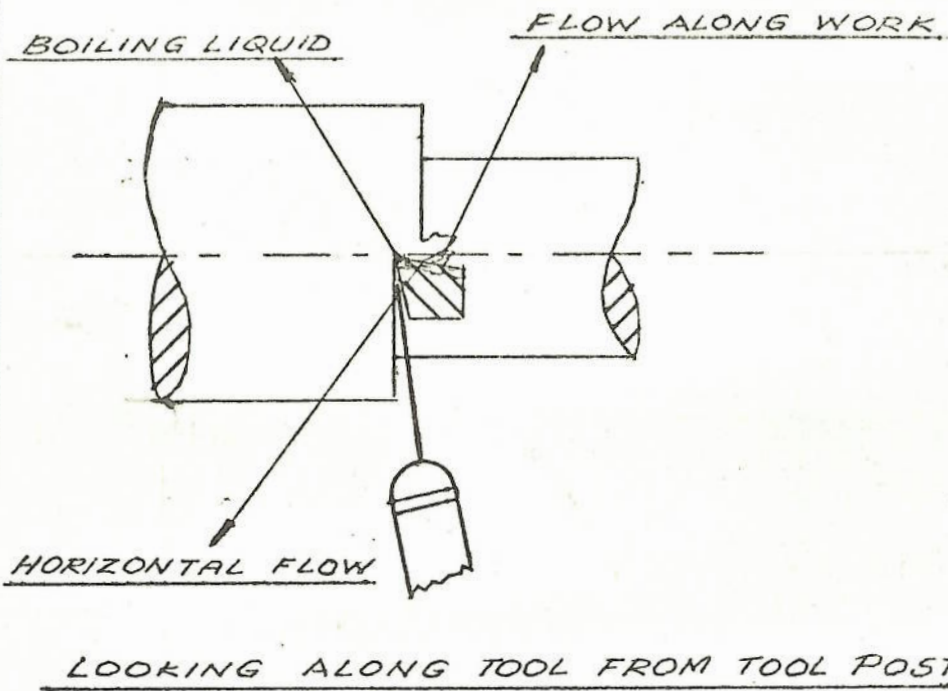


DIAGRAM OF "HI-JET" SYSTEM

FIG D6/3/1



CHIP

FIG D6/3/2 JET LOCATION AND
TOOL POST

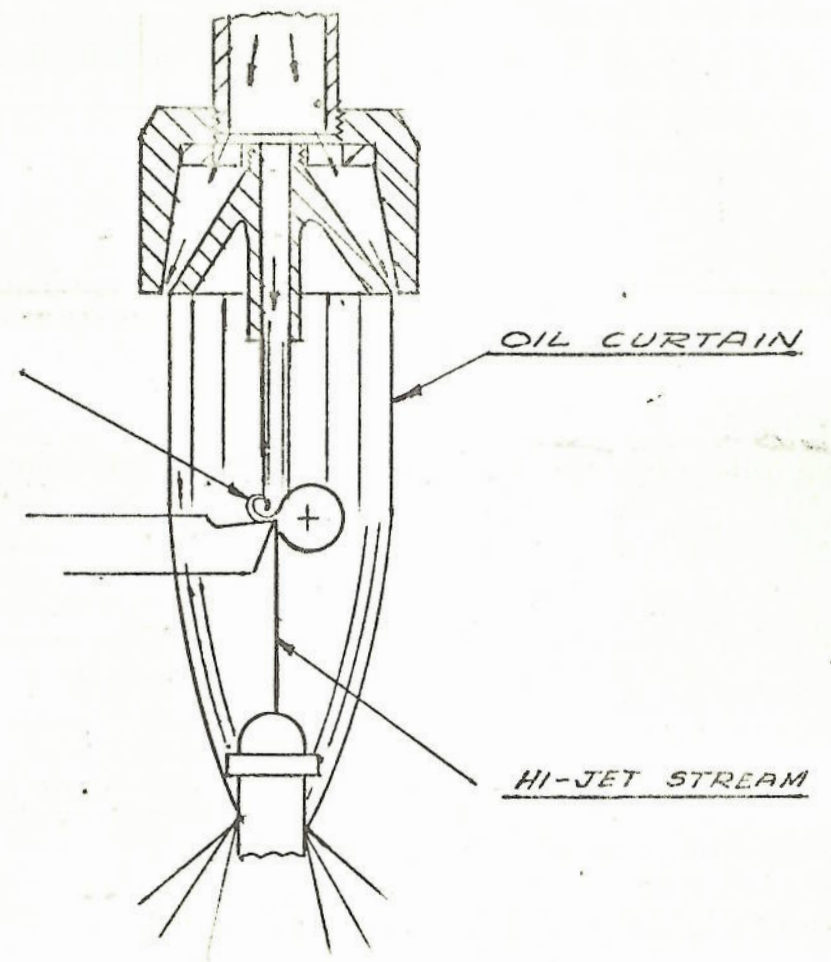


FIG D6/3/3 METHOD TO ABSORB
SMOKE AND VAPOUR

SINGLE TYP 0.014" DIA
SAE 3150 1 MIN DRY LIFE

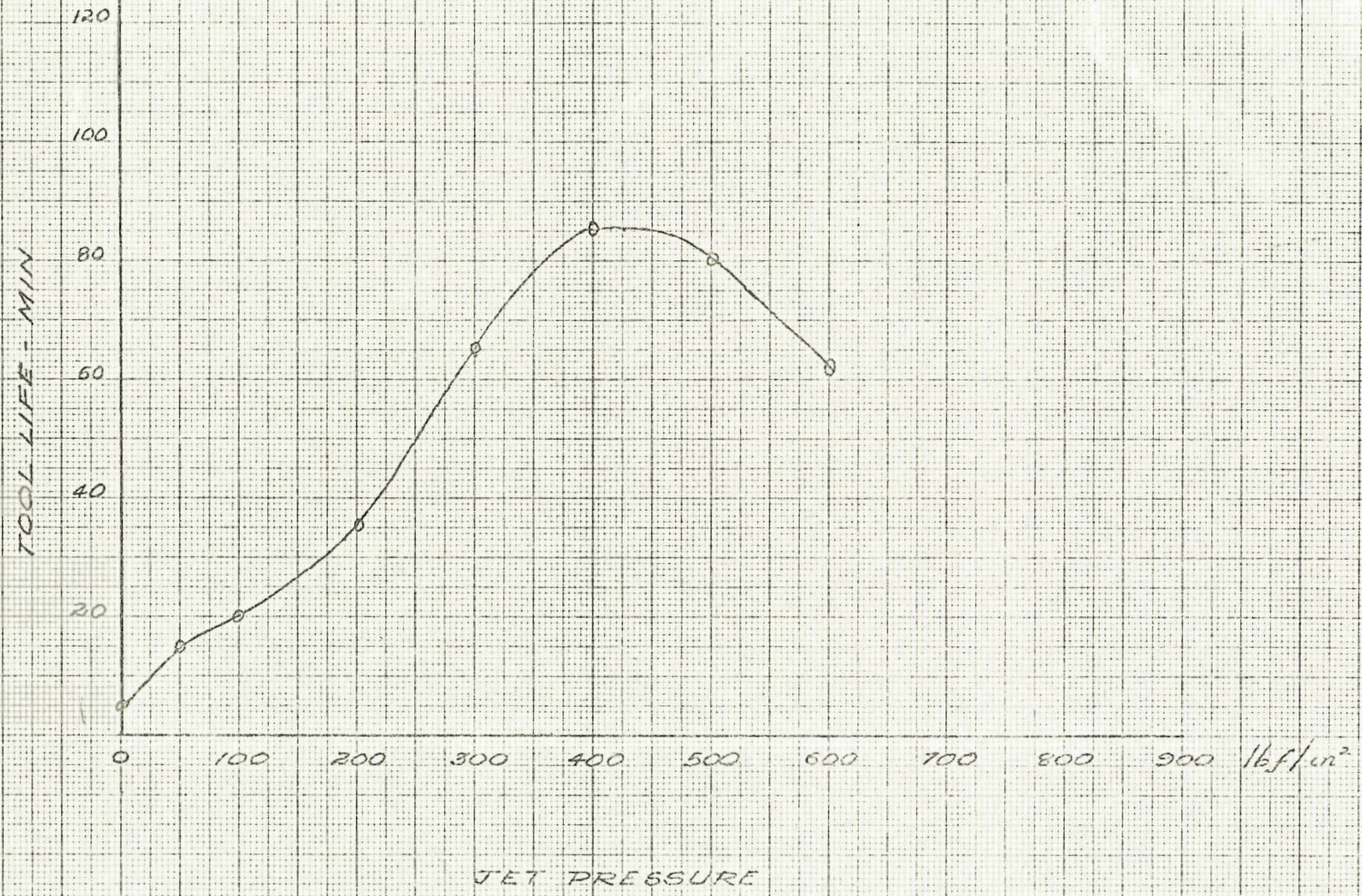
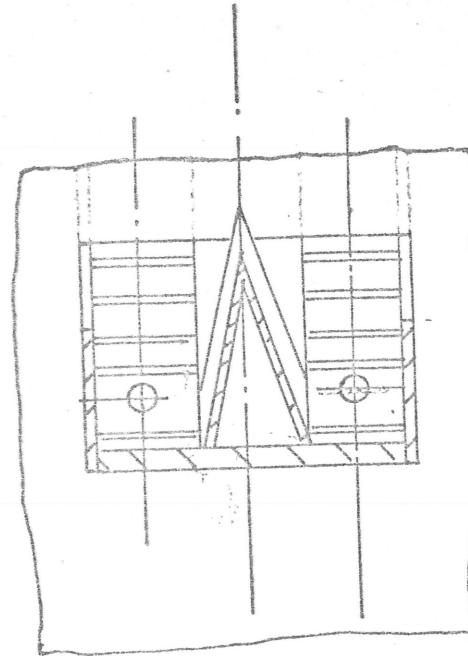
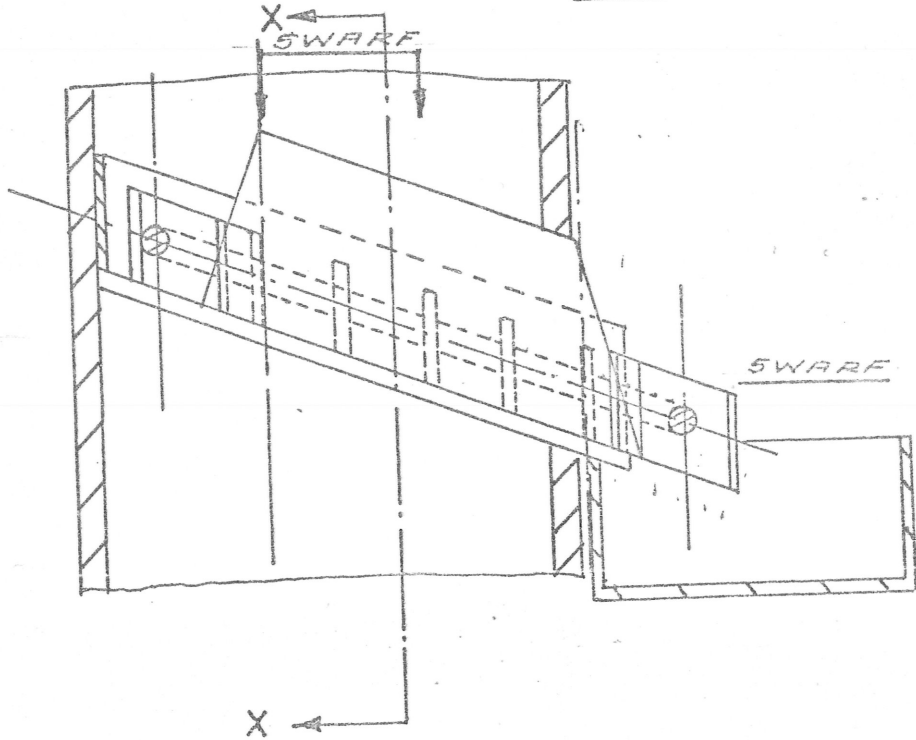
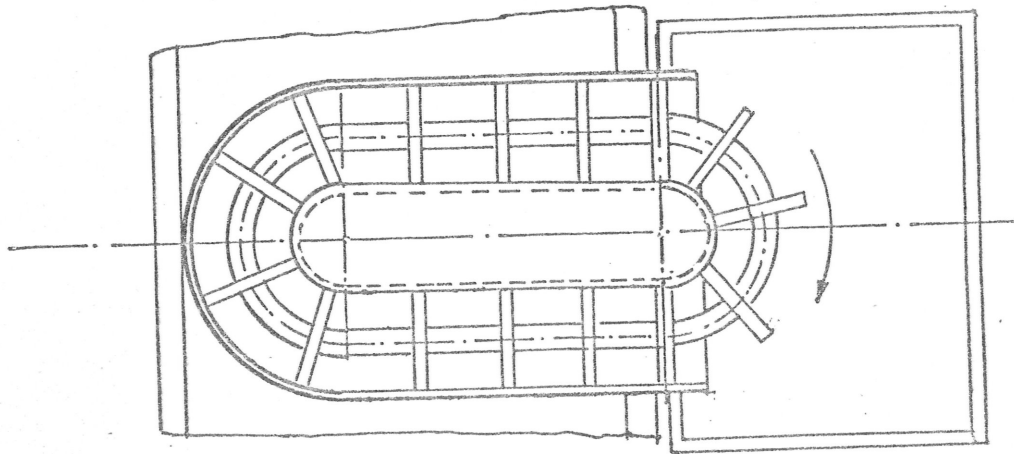


FIG D6/3/4 TOOL LIFE VS JET PRESSURE



SECTION-XX

FIG DS/4/5 TUBE CONVEYOR

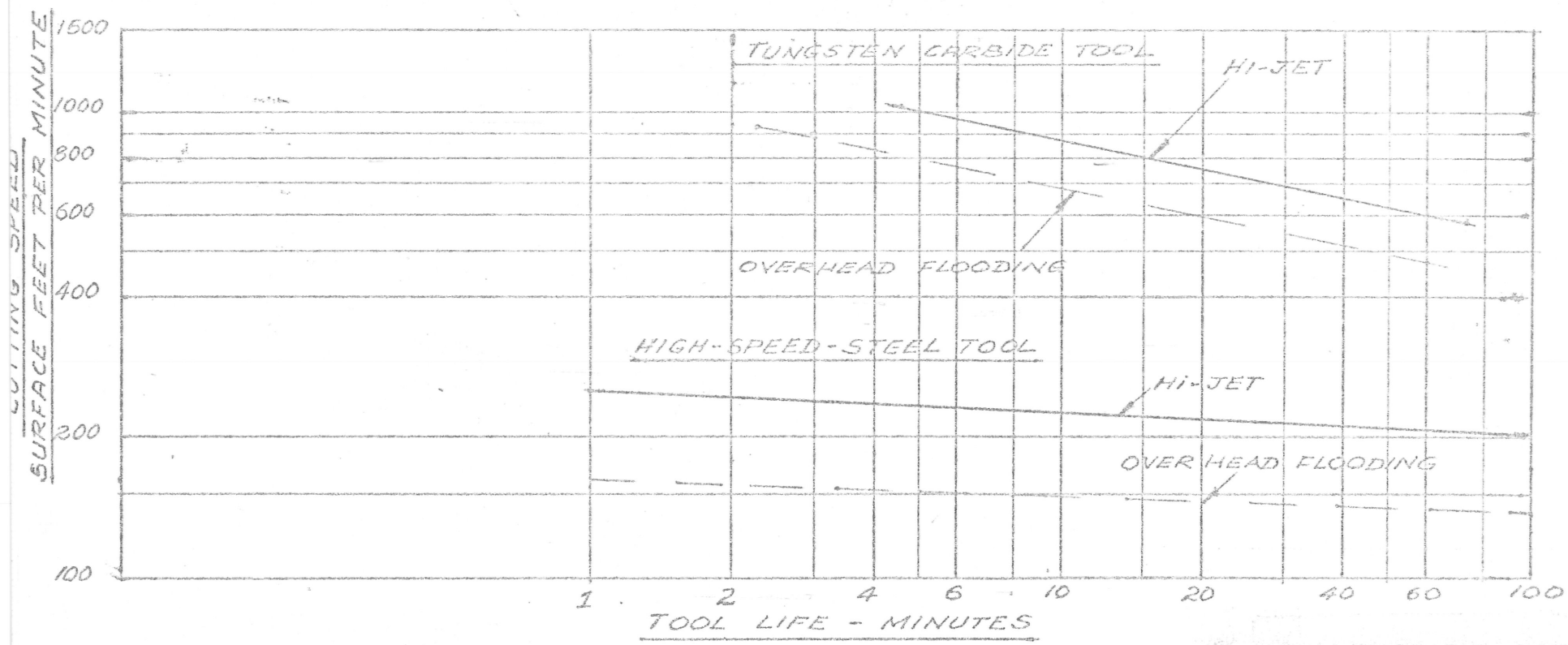


FIG 26/3/5 RELATION OF TOOL LIFE TO CUTTING SPEED

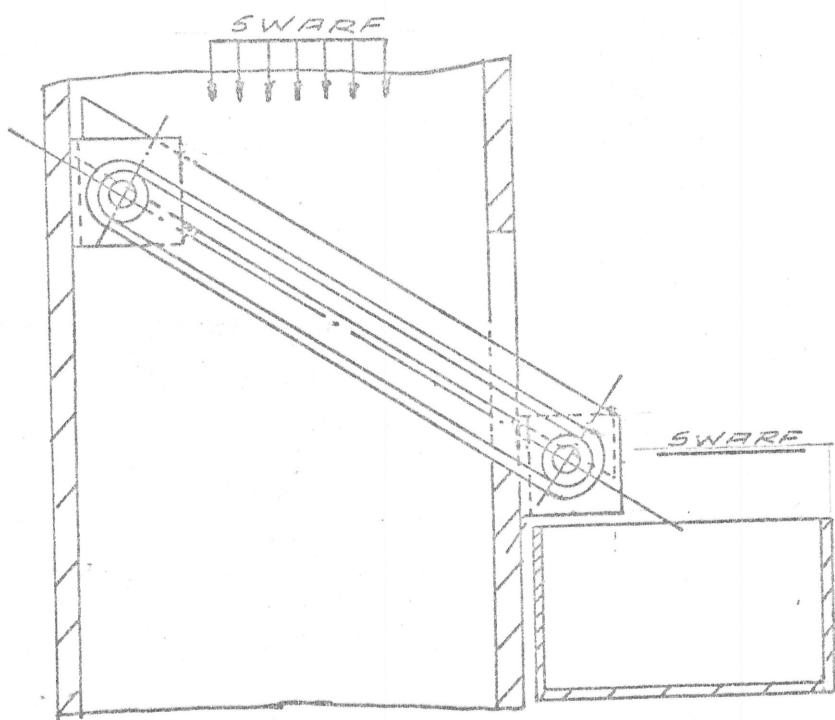
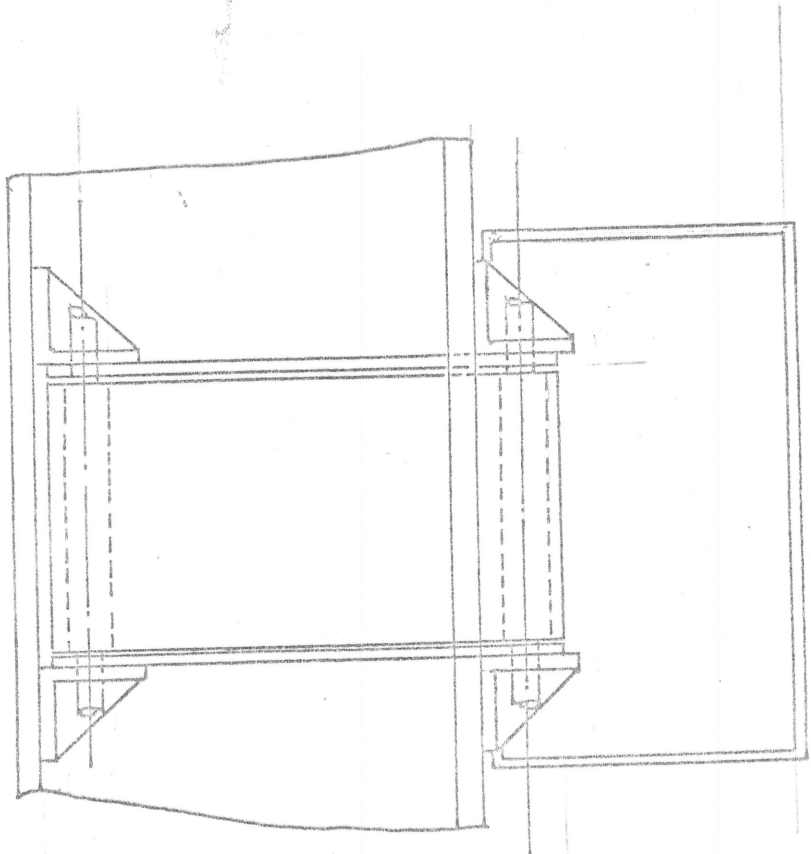


FIG DC/4/7 BELT CONVEYOR

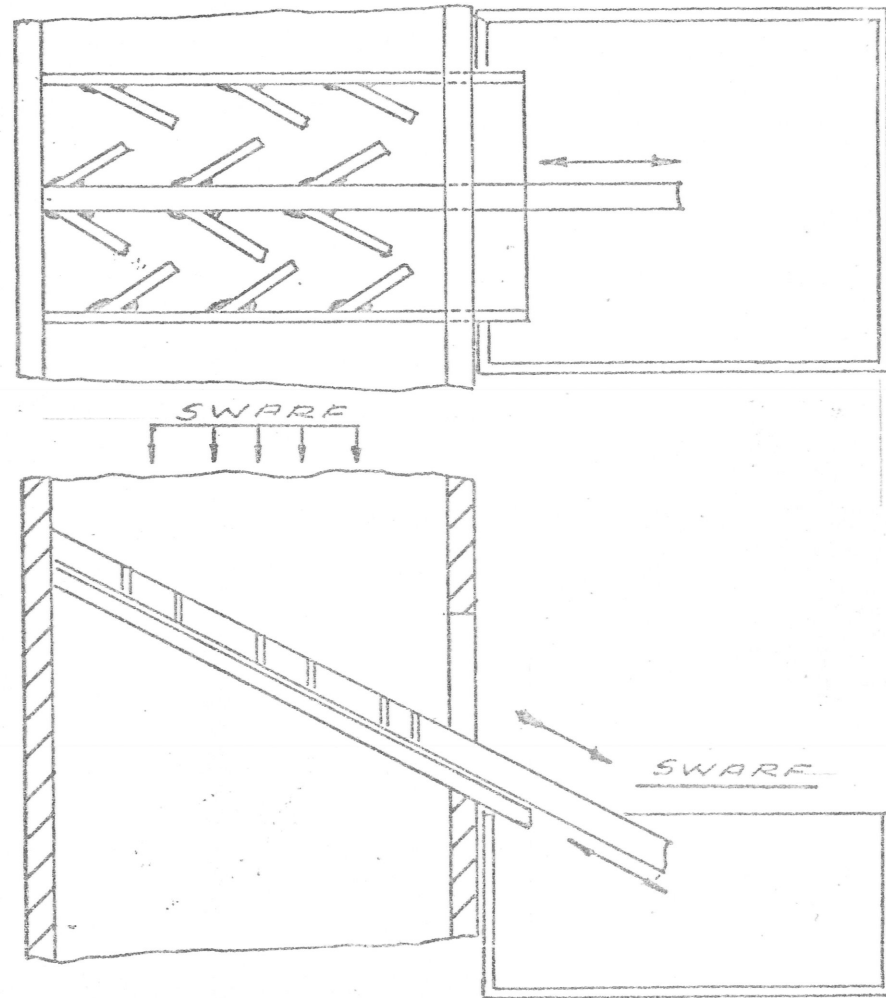


FIG D6/A/6 PUSHER CONVEYOR (TINES TYPE)

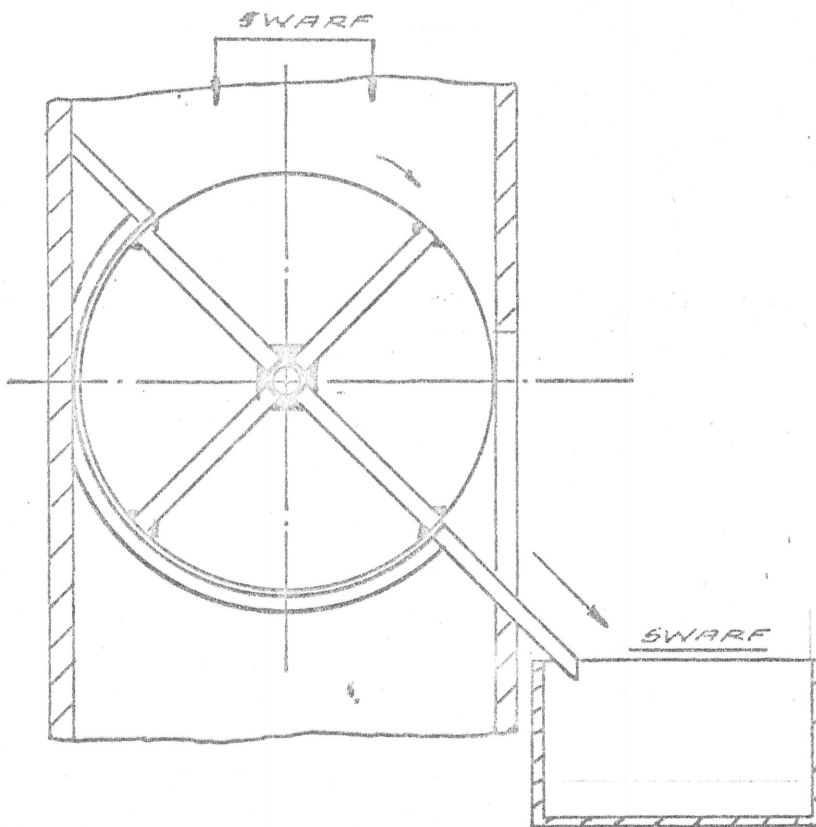
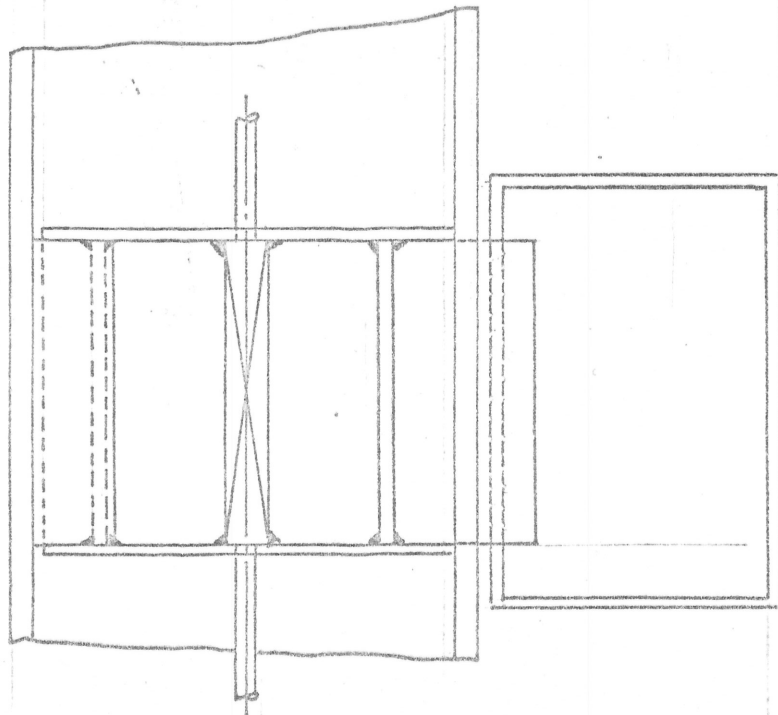


FIG D81412 ROTATING BOX TYPE CONVEYOR

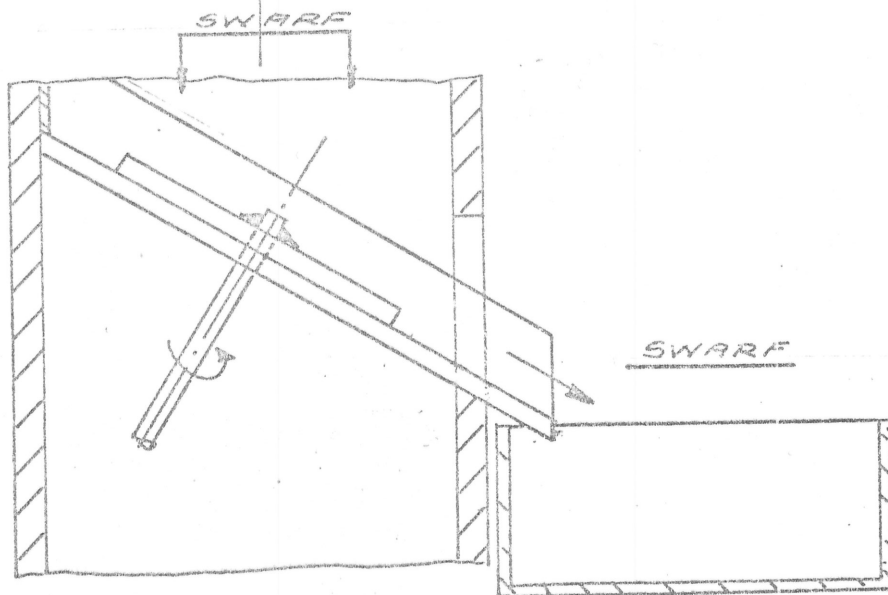
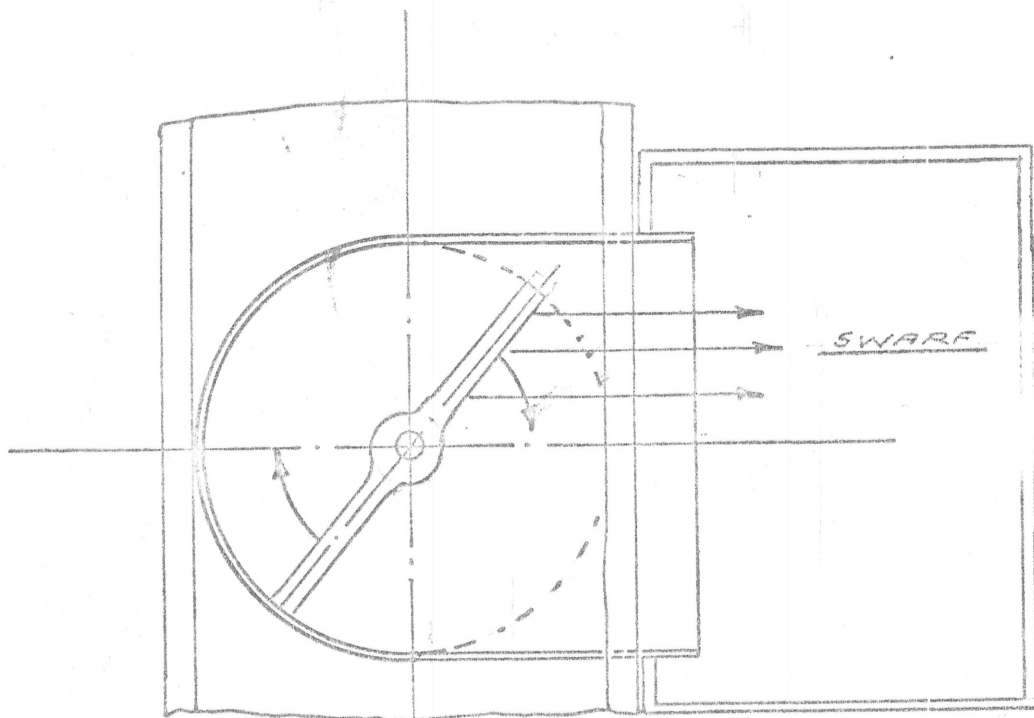


FIG DS/4/9 ROTATING BAR TYPE AGITATOR

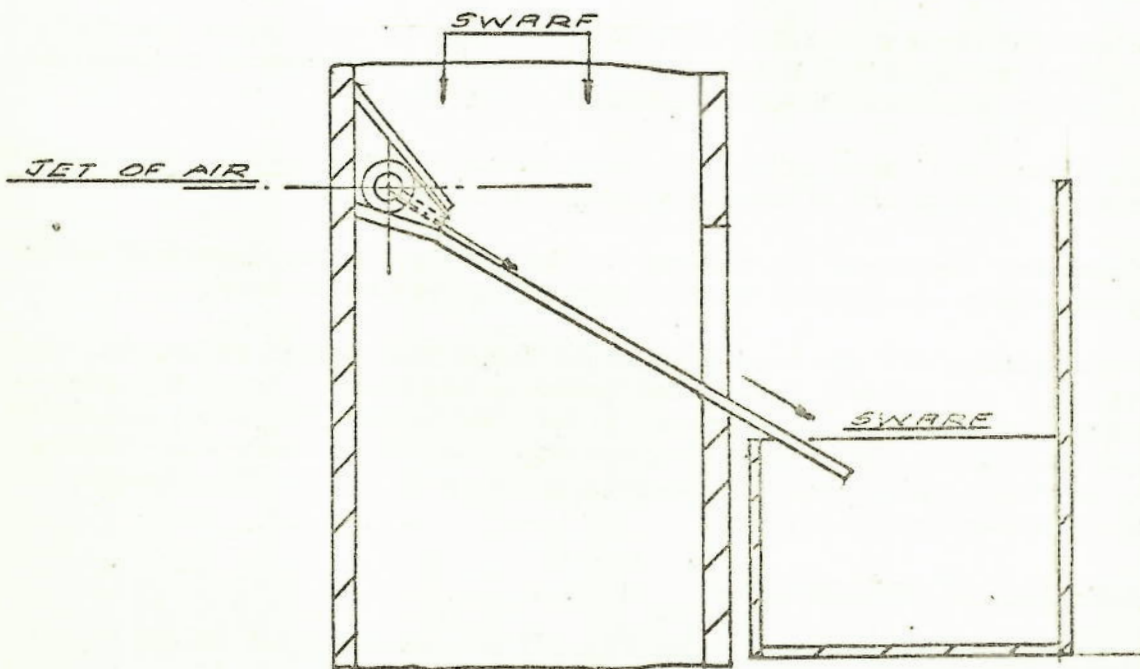
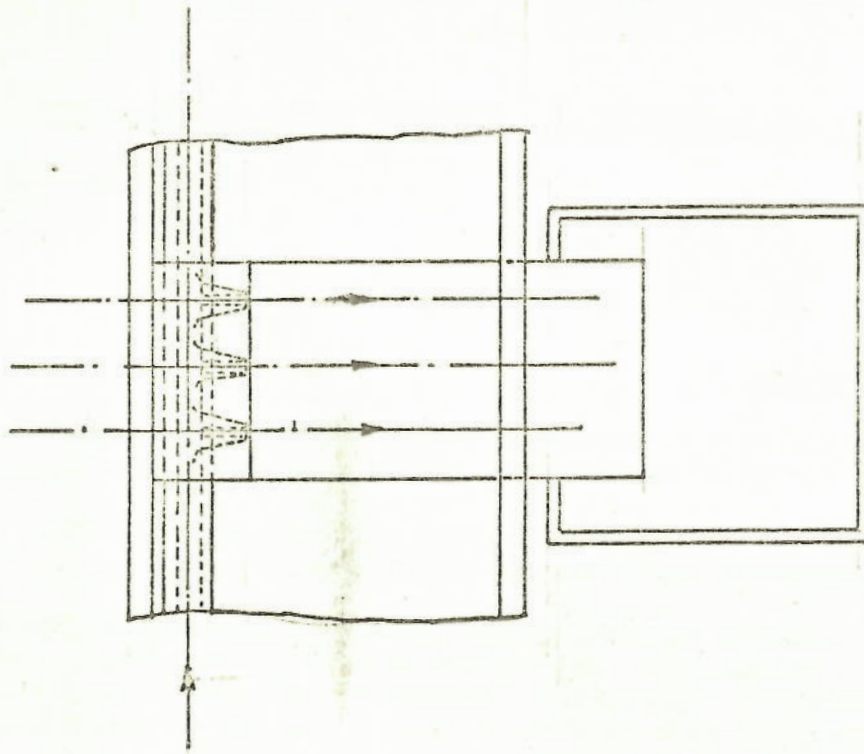


FIG D6/4/10 PNEUMATIC AGITATOR

D7. REPORT OF THE AESTHETIC AND ERGONOMIC DESIGN SUB-COMMITTEE.

Prepared by- M.F. Scarffe
A.V. Torry
R. Bera.

D7.1. Introduction.

The function of this committee was to give consideration to the aesthetic and ergonomic design of the machine, and to construct a quarter full size model.

Because of the influence of fashion and ergonomics, machine tools can no longer be designed on purely functional grounds.

Fashion dictates that the machine should be pleasing to the eye, therefore such things as colour, size and outline should be considered. However aesthetics cannot be considered in isolation, as it must complement basic factory requirements such as minimum floor space and accessibility.

Ergonomics, Ref. D7.4., gives the designer information, so that the man-machine interface functions with the maximum efficiency.

The general ergonomic considerations that must be applied to a machine tool are, that the operator should be able to set up and control the machine rapidly, safely and with the minimum amount of fatigue.

This means, the adoption of standards concerning the logical arrangements and direction of movement of operating handles, knobs etc.

Unnecessary movements and fatigue can be avoided if the layout of levers and control panels are adapted to the geometry of the human body.

As the machine has not been brought to the detail design stage, the numbers and function of its control levers and knobs is not known. However a part of the machine that requires an operator is the tool loading magazine, and it is suggested that the magazine be fitted with handles. This would make loading simpler, and the handles could be positioned so as to locate the magazine in a near zero position on the machine.

D7.2. Appraisal of Configuration.

The configuration of the machine Fig. D7.2/1 as advocated by the configuration committee was examined.

It was considered that this was an aesthetically unacceptable machine that occupied a large floor area, with extremely poor accessibility. In addition to raise it to operator height would have incurred an excessive dead volume under each axis.

D7. REPORT OF THE AESTHETIC AND ERGONOMIC DESIGN SUB-COMMITTEE.

Prepared by- M.F. Scarffe
A.V. Terry
R. Bera.

D7.1. Introduction.

The function of this committee was to give consideration to the aesthetic and ergonomic design of the machine, and to construct a quarter full size model.

Because of the influence of fashion and ergonomics, machine tools can no longer be designed on purely functional grounds.

Fashion dictates that the machine should be pleasing to the eye, therefore such things as colour, size and outline should be considered. However aesthetics cannot be considered in isolation, as it must complement basic factory requirements such as minimum floor space and accessibility.

Ergonomics, Ref. D7.4., gives the designer information, so that the man-machine interface functions with the maximum efficiency.

The general ergonomic considerations that must be applied to a machine tool are, that the operator should be able to set up and control the machine rapidly, safely and with the minimum amount of fatigue.

This means, the adoption of standards concerning the logical arrangements and direction of movement of operating handles, knobs etc.

Unnecessary movements and fatigue can be avoided if the layout of levers and control panels are adapted to the geometry of the human body.

As the machine has not been brought to the detail design stage, the numbers and function of its control levers and knobs is not known. However a part of the machine that requires an operator is the tool loading magazine, and it is suggested that the magazine be fitted with handles. This would make loading simpler, and the handles could be positioned so as to locate the magazine in a near zero position on the machine.

D7.2. Appraisal of Configuration.

The configuration of the machine Fig. D7.2/1 as advocated by the configuration committee was examined.

It was considered that this was an aesthetically unacceptable machine that occupied a large floor area, with extremely poor accessibility. In addition to raise it to operator height would have incurred an excessive dead volume under each axis.

In an attempt to reduce the dead volume and floor space, three arrangements were considered.

Fig D7/2/2 shows an arrangement in which the headstock-tailstock axis is vertical. This gave a tall unstable machine, that would have required an extra support, also swarf would have fallen directly onto either the headstock or tailstock.

For these reasons this arrangement was rejected.

Fig D7/2/3 shows an arrangement with the headstock - tailstock axis horizontal, but with the toolpost axis vertically above it. This was done to reduce the floorspace and dead volume, but increase accessibility.

However, if the workpiece is brought to operator height, the tool loading position becomes excessively high.

If the tool loading height is reduced, then the workpiece height becomes too low. As a compromise was not possible this arrangement was also rejected.

Fig D7/2/4 shows an arrangement with the headstock - tailstock axis remaining horizontal but with the toolpost axis vertically downwards. Whilst it was appreciated that this arrangement would give the same disadvantages as Fig D7/2/3 it was investigated in case it introduced any additional advantages or disadvantages.

It produced a useful storage volume, but swarf would have fallen directly onto the moving toolpost.

D7.3. Final configuration.

The arrangements shown in Figs D7/2/2 and D 7/2/4 were carefully considered and it was concluded that the stability and swarf disadvantages could be overcome if the machine were tilted at an angle to the horizontal.

An angle of 45° appeared to be ideal since it brought the tool loading and workpiece positions as near level as possible, also it is the optimum angle for swarf to fall clear of any moving parts.

However the overall height of the machine would be ()ft. and it was considered that this was too large.

The situation was compromised by changing the angle to 30° thus arriving at the final configuration shown in Fig D7/2/5.

Fixing the slideways at 30° to the horizontal could have affected the control of the moving parts, consultation with the Control Systems Committee confirmed that this would not be a problem.

The final configuration has the advantage that it occupies the minimum floor area, and has good accessibility. Also aesthetically a 30° angle

is pleasing to the eye, and gives a modern appearance.

D7.4. References.

- 1) Koenigsberger, F. Design Principles of Metal Cutting
Machine Tools.

Pergamon Press
- 2) Ergonomics for Industry Nos. 1-12.
Ministry of Technology Jan. 1966.

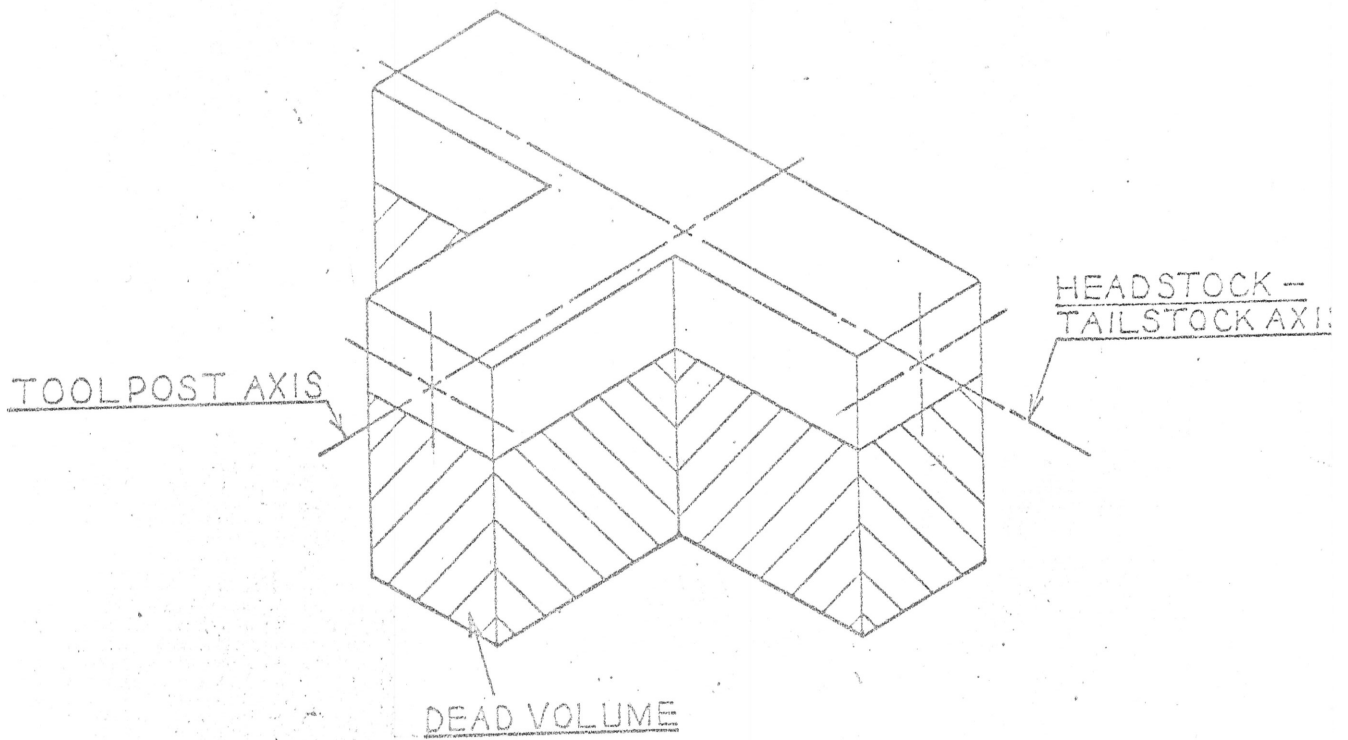


FIG. D7/2/1. INITIAL CONFIGURATION.

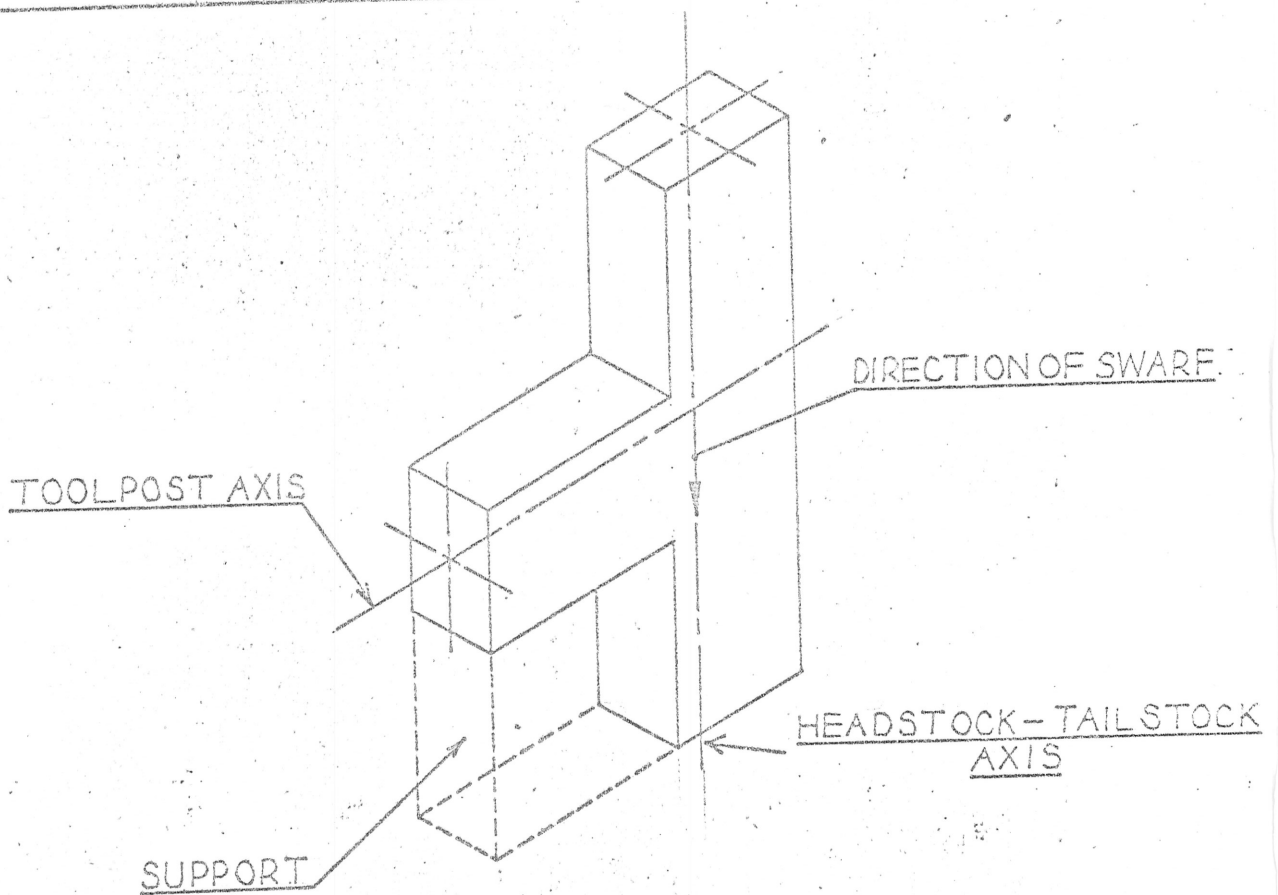


FIG. D7/2/2. VERTICAL ARRANGEMENT

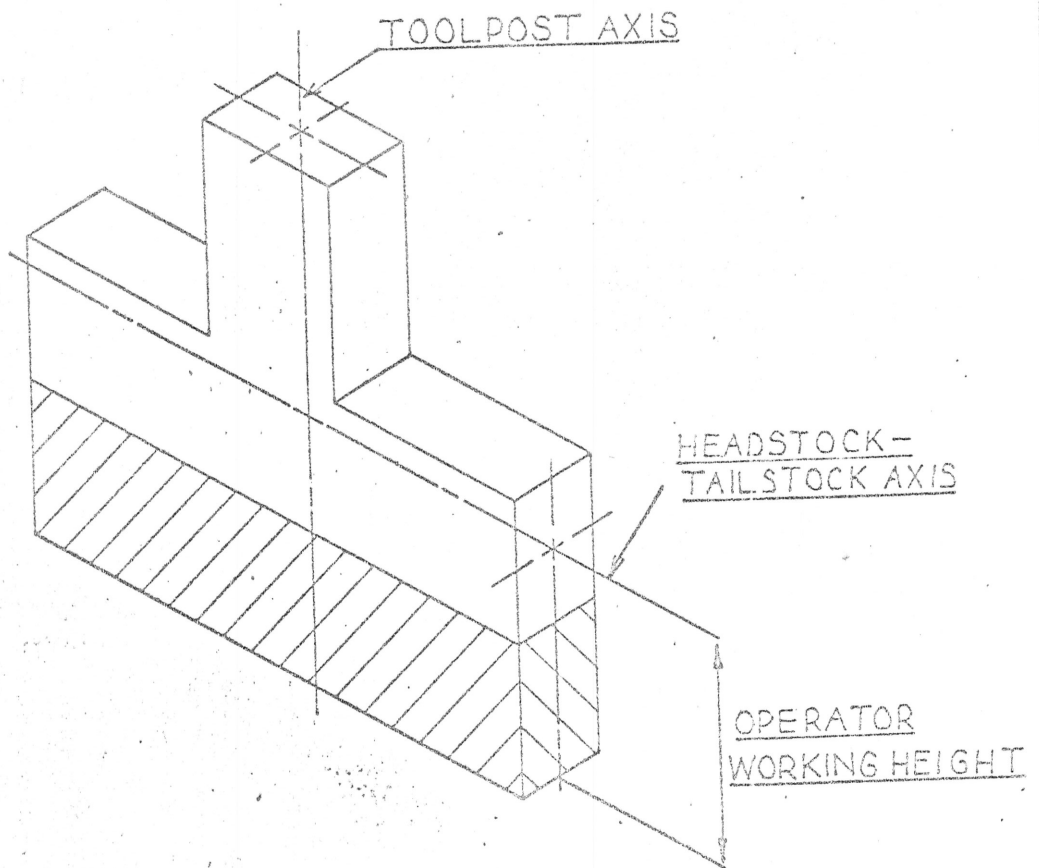


FIG. D7/2/3. HORIZONTAL ARRANGEMENT 1

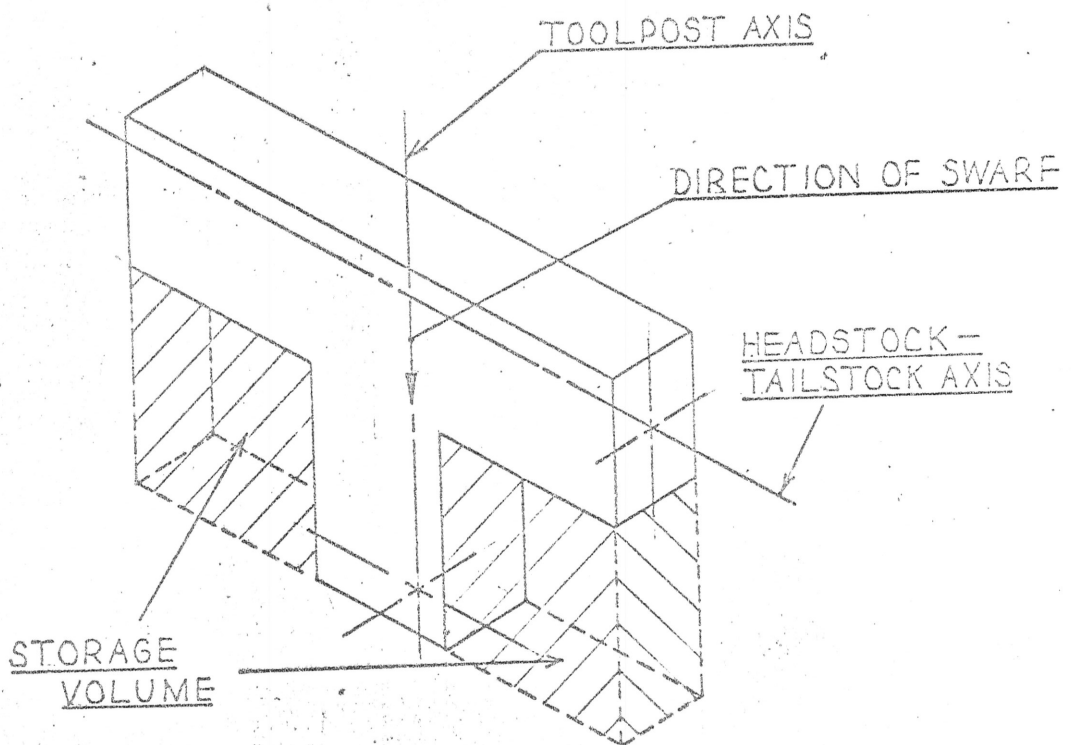


FIG. D7/2/4. HORIZONTAL ARRANGEMENT 2.

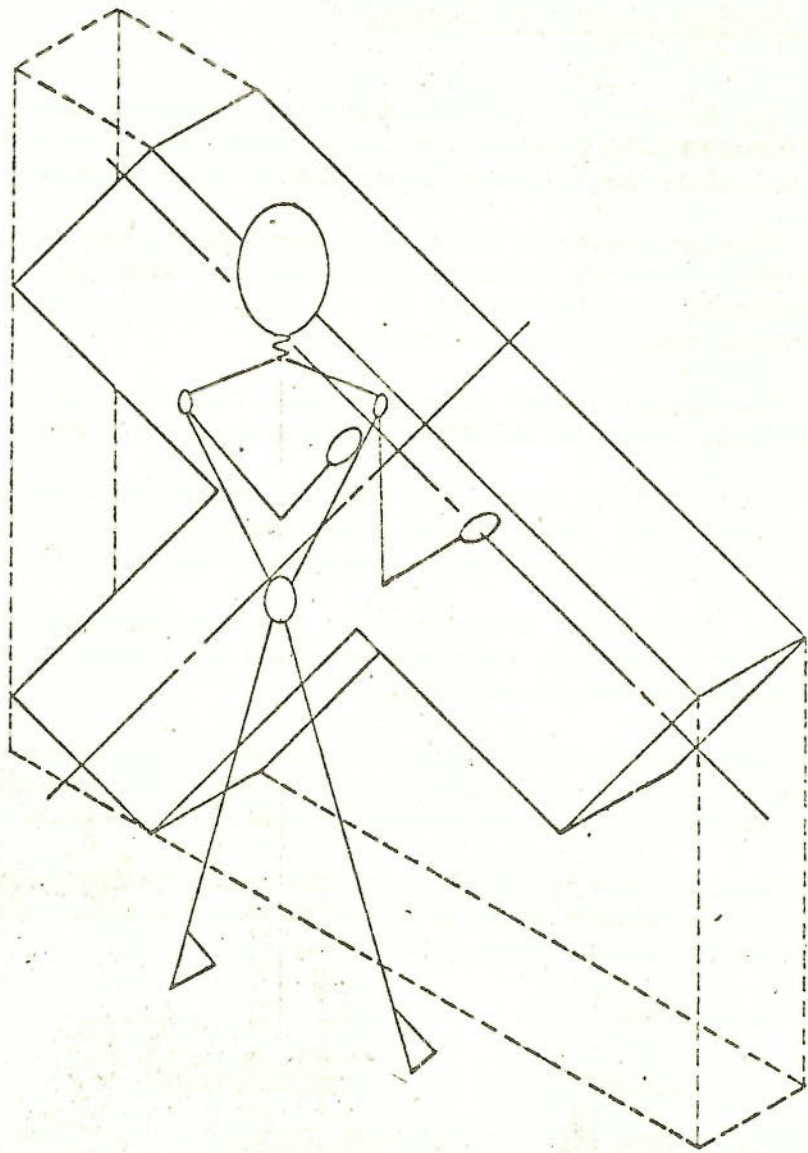


FIG. D7/2/5. FINAL DESIGN CONFIGURATION

D8. Costing of Machine Design.

The machine design cost has been estimated so that an 'order of cost' for the complete machine can be established. The treatment is not rigorous and therefore the figures given should only be taken as a guide.

Costing has been based on a comparison with similar equipment and a quantity of twenty machines per annum. Casting pattern costs and capital expenditure are written off in the first year so that pattern costs are amortised over the first twenty units.

The grade 17 cast iron have been evaluated at 1/4d. per lb and machining costs were based on an hourly machining rate of £1.10s.0d.

To obtain an exact cost for the manufacture of the machine would be very difficult. Many of the major parts have been designed specifically for the machine and as a result cannot be bought-out as standard items.

In the case of units such as the copy turning attachment and chuck a price for a complete unit was supplied by a manufacturer. These items are designated as bought-out B0.1.

Items which are bought-out and have been allocated an estimated cost are indicated by B0.2 This includes the frame which would be bought-out as a heat treated casting and machined within the machine tool makers factory.

Equipment which appears on the assembly drawings and is not included in the cost breakdown such as the actuation cylinders and hydraulic motor has been costed in the Control Section.

Cost Breakdown:

<u>Section</u>	<u>Description</u>	<u>£</u>
D3	Chuck with Drawbar B0.1.	125
D3	Hydraulic Cylinder and Distributor Assembly B0.1.	80
D3	Disc Brake Assembly with casing	30
		<u>235</u>
D4	Frame B0.2	<u>600</u>
D5.6	Toolpost Assembly	250
D5.6	Magazine	150
D5.10	Sequencing Equipment B0.1.	50
		<u>450</u>
D6.2	Tailstock Unit B0.2.	15
D6.2	Tailstock Revolving Centre B0.1.	23
D6.3	Cutting Lubricant Equipment B0.2.	50
D6.4	Swarf Removal	10
		<u>98</u>
		<u><u>TOTAL £1,383</u></u>

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D3	Chuck with Drawbar	B0.1.	125
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	and Distributor Assembly		
D3	Disc Brake Assembly		30
	with casing		<u>235</u>
D4	Frame	B0.2	<u>600</u>
D5.6	Toolpost Assembly		250
D5.6	Magazine		150
D5.10	Sequencing Equipment	B0.1.	50
			<u>450</u>
D6.2	Tailstock Unit	B0.2.	15
D6.2	Tailstock Revolving Centre	B0.1.	23
D6.3	Cutting Lubricant		
	Equipment	B0.2.	50
D6.4	Swarf Removal		10
			<u>98</u>
			<u><u>TOTAL £1,383</u></u>

Attachments sold as extras which are not included in the total.

Section	Description		£
D5.9	Copy Turning	B0.1	300
D3.	Work Loading		250

The total cost does not cover machine assembly and tests which are included in the Master Programme Section.

D.9. Recommendations for the Future.

During the project it became apparent that some aspects of the machine design could be extended to advantage. The indexing toolpost on the cross-slide with its tool magazine was one such aspect and in conjunction with this the sequencing circuit for the machine elements.

The committee considers these aspects of the design could form the basis of a future student project. This project could evaluate the design, manufacture and development of the mechanisms and associated circuits providing the basis for a universal transfer machining centre.

Ball hydraulic motors with integral hydrostatic bearings are currently being developed. When these become available they will provide cheap high performance spindle drives with a much smaller physical size than present units. The introduction of such motors into the machine design would reduce the overall size and cost.

Further cost reductions could well be achieved by the application of self lubricating, low friction composite coatings to the machines slideways. These coatings remove the need for a whole frame structure to consist of a wear resistant material, thus permitting the introduction of composite materials such as boron aluminium, steel aluminium or glass reinforced resins. Composite materials are far stronger and stiffer than steels when correctly designed into a structure. They therefore give size and weight reductions and will become less expensive as they are more widely used.