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DRILLING SYSTEM DESIGN PROJECT 1967

FINAL REPORT OF DRILL HEAD COMMITTEE

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Summary

This report outlines the design of the Drill Head for a numerically controlled drilling machine.

The Committee chose to employ a nydraulic main spindle drive. As a result of the problems experienced by Wadkin Limited, Leicester, with hydraulic feed control, it was decided to replace the original hydraulic feed actuator by a conventional 18 speed gearbox. Drill depth control is achieved by conventional gate stops employing pneumatic sensing elements to keep in line with the fluidic elements of the control system.

The latter section of the report makes recommendations for future work on the design of feed mechanisms to avoid the use of gearboxes.

CONTENTS

1.0. INTRODUCTION

The terms of reference were to design the main spindle, the drive assembly and the feed mechanism, for Z axis movement including depth of feed control, whilst providing a Z axis movement of 12 in (300 mm) and ensuring that the initial specification was met.

The head has been designed initially for single tool operation where hole depth is controlled by setting stops whilst Z axis feed rate is selected by the operator independently of the tape. Spindle speed is also set manually.

Unit construction was decided upon to facilitate ready adaptation of the basic design to accommodate future refinements including automatic tool change.

2.0. MAIN DRIVE

The factors influencing the choice of main spindle drive are outlined in detail in ref. 1. This report recommended that a hydraulic drive be employed in which a variable displacement hydraulic pump supplies a variable displacement hydraulic motor. Since then, as a result of consultation with a representative of Dowty Hydraulics Limited, the committee has replaced the variable displacement motor with a fixed displacement version. An arrangement such as this gives improved performance characteristics as shown in an idealised form in fig. 1, since there is less wasted power at low speeds.

The committee had to keep in mind the possible hydraulic power needs for the rest of the machine; it was decided therefore to employ a closed circuit between pump and motor, and use a separate small pump to make up any losses to the main pump (see drawing no. DH-004). In this way any other hydraulic services could be supplied by the small pump - the size of which could be chosen independently of the spindle circuit.

3.0. TOOL FORCES AND POWER REQUIREMENTS

The factors governing drilling forces are :-

i) Properties of material to be drilled.
ii) Shape of drill (diameter, rake angle, etc).
iii) Chip section being cut (drill diameter, feed rate).
iv) Cutting conditions (depth of hole, coolant),

Because the rake angle changes across the cutting edge, little can be done to optimise drills for given material.

By "pointing" the drill cross edge it is possible to reduce the axial thrust by 30% although this does not appreciably reduce the torque required for cutting. For this project, however, it must be assumed that the drill is not "pointed".

Axial thrust increases proportionally to drill diameter whilst torque increases by the square of the drill diameter. The increase in cutting force is less than linearly proportional to feed rate. This force decreases, however, with increased helix angle.

Axial thrust decreases with decreasing nose angle, but at the same time torque increases because at equal diameter there is a longer cutting edge presented by the drill. Smallest forces occur when the cross edge is at 55°-60°. (This is normal practice - 118° drill points).

The friction forces at the outside diameter of the bore as well as chip removal forces add to the cutting resistance so that the depth of hole affects the cutting forces. Efficiency of lubrication and the effectiveness of the coolant are two further factors which affect drill performance.

The forces of tapping and reaming are smaller than those encountered during drilling indicating that the spindle should be designed around the drilling operation whilst retaining slow spindle speeds and various feed rates for the above operations.

If designed for the worst case the forces involved would be as below :-

Using the graph of cutting forces (fig. 2).

1 in (25 mm) dia drill cutting 38 tonf/in^2 (587 MN/m^2) carbon steel at feed rate of 0.040 in/rev (1 mm/rev) at a cutting speed of 90 ft/min (2,800 mm/min).

Axial thrust	4,500 lb
Torque	2,000 in.lb

Assume no lubrication i.e. 30% increase in thrust and 10% increase in torque.

Then forces become; Axial Thrust 5,600 lb
Torque 2,200 in.lb

$$HP = \frac{2\pi NT}{33000} \qquad \text{where } N = \frac{12\,s}{\pi D}$$

$$\therefore\ HP = \frac{2.\pi.12.S.T}{\pi.D.33000}$$

$$= \frac{2.\pi.12.90.2,200}{\pi.1.33,000.12}$$

$$= 12\ \ (8.9\ KW).$$

This figure for h.p. requirements is very high. Consultation of the Vero Machine Tool Co. Catalogue for their Vero 15/24 1000 series drilling machine showed that this 1 in diameter capacity machine was fitted with a 2 h.p. spindle drive motor. However, the machine is limited to drilling a 1 in dia. hole in mild steel at 0.010 in/rev which considerably reduces the cutting forces.

Therefore, if the machine spindle is limited to 0.010 in/rev feed rate for 1 in dia holes in 38 ton/in^2 carbon steel, the power requirements are as shown below.

From fig. 2 Axial thrust 2,000 lb
Torque 800 in.lb

At 90 ft/min cutting speed.

$$HP = \frac{2\pi NT}{33000} = \frac{2\pi.12.S.T.}{\pi.D.33,000}$$

$$= \frac{2\pi.12.90.800}{\pi.1.33,000.12}$$

$$= 4 \text{ (3 KW)}.$$

It was decided to employ a 5 h.p. motor. A Lucas LM 500 fixed displacement hydraulic motor was chosen, the characteristics of which are shown in fig. 3.

Summary of power requirements and forces on spindle :-

Motor to supply 5 h.p. (37 KW) over speed range 80-2000 rev/min.

Spindle and bearings to withstand 2000 lb (9,100 kg) axial thrust and transmit 5 h.p. (3.7 KW) at 80 rev/min.

4.0. SPINDLE BEARINGS

The choice of bearings for the spindle was between

i) hydrostatic ii) aerostatic iii) rolling element

Hydrostatic and aerostatic bearings may be classed together for their very low friction properties. Both require close manufacturing tolerances, and would give a very small 'runout'. Oil and air are both available on the machine but a hydrostatic bearing would be a source of heat in the head whilst an air bearing would run cool. Since no large side loads were envisaged on the spindle, it was felt that the above class of bearing would not be employed to full advantage and therefore the cost would not be warranted. It was decided therefore to employ taper roller bearings, since these are readily available and relatively cheap.

The 'Gamet' bearing arrangement was employed where a pair of taper roller bearings are set back to back at the front of the spindle locking it axially at the front. A single taper roller bearing acts at the rear support (see drawing no. DH-003). In this way adjustment of the rear bearing can control the runout at the nose of the spindle.

The bearings selected were :-

at the nose,	two SKF 30211 bearings
at the rear,	one SKF 30210 bearing

Taper roller bearings can support side forces as well as large axial loads. This is an advantage with a horizontal spindle machine such as the one being designed since the only side load is the weight of the spindle itself. Wear is readily compensated for with this bearing type.

5.0. SPINDLE DESIGN

The main design considerations for a spindc are :-

i) Axial and transverse stiffness.

ii) Choice of bearings and their position (see Section 4.0).

iii) Shape of spindle with regard to availability of bearings.

iv) Tool location.

v) Lubrication of bearings.

The axial stiffness of the spindle is governed by the bearing stiffness, whilst transverse stiffness and bearing positioning problems normally arise with spindles which have to withstand large side loads.

It was decided that the spindle bearings should be mounted in a quill which in turn should be a sliding fit in the main casting. The clearance should be small so as to retain the accuracies laid down by the specification. The quill would be driven forward to provide the Z axis feed by some means which would afford several discrete rates of feed. This mechanism is described in a later section.

Since the machine was designed as the first of a range of machines, the basic unit being a single tool version, it was decided to employ standard tooling so as to avoid the cost of pre-set tooling. Any variation in tocl length can be accommodated by adjustment of the stops. (see Section 6.2). A No. 3 Morse taper is provided in the nose of the spindle - this being the best method of centring a tool in a spindle to achieve repeatable positioning.

The spindle bearings are to be lubricated by a drip feed of constant 10 drops of oil per minute to avoid temperature rise due to lack or excess of oil.

The spindle has to transmit 5 h.p. (3.7 KW) at 80 rev/min.

$$\text{Torque} = \frac{HP.33{,}000}{2\pi N} = \frac{5 . 33{,}000}{2\pi . 80}$$

Torque = 350 ft.lb (47.2 m.kg)

a) To find shear stress on surface of spindle when transmitting 350 ft.lb (47.2 m.kg) torque

$$\frac{T}{J} = \frac{q}{r}$$

T = torque
J = polar moment of inertia
q = shear stress
r = radius of surface

$$J = \frac{\pi(D^4 - d^4)}{32}$$ for hollow shafts

Consider the spindle to be 3 in o.d. 1 in i.d.

$$J = \frac{\pi(81-1)}{32} = 7.5$$

$$q = \frac{T.r}{J} = \frac{350.12.1.5}{7.5} = 840 \text{ lbf/in}^2 \ (5.8 \text{ MN/m}^2)$$

This is an acceptable figure if the spindle is of carbon steel.

b) To find shear stress on surface of splined drive when transmitting 350 ft.lb torque.

Consider spline to be solid shaft 1 in. o.d.

$$J = \frac{\pi D^4}{32} = \frac{\pi .1}{32} = 0.1$$

$$q = \frac{Tr}{J} = \frac{350.12. .5}{0.1} = 20{,}000 \text{ lbf/in}^2 \ (138 \text{ MN/m}^2).$$

This figure is a little high, although if a good quality steel were employed for this member, it would be acceptable.

6.0. FEED MECHANISM

With hydraulic power available on the machine it would be an obvious choice to employ an hydraulic actuator to control the quill feed by a flow control valve. It was with this method in mind that the committee sketched a proposal (drawing no. DH-002), to illustrate the mode of operation. A double acting cylinder was shown detached from the main housing to reduce heat transfer to the quill. An alternative arrangement is to use the quill as a piston and the quill housing as a cylinder casing (see fig. 4). This system is employed on the Vero 15/24 and 20/40 drilling machines. Expansion due to heating is accommodated in the splined couplings between the front and rear bearings.

The most important point raised during consultations with design staff at Wadkin Limited was the difficulty in maintaining a constant feed rate with hydraulic actuation; heating of the oil reached a maximum after approximately 3-4 hours running of the machine.

The change in oil viscosity affected the flow to the cylinders, and a greater feed resulted. It was explained that no direct readout of feed was provided on machines employing hydraulic actuation; a dummy run was necessary when tapping to adjust the feed control valve.

To overcome this basic feed rate problem, it was decided to use a conventional gearbox and clutch arrangement, (figs. 5 and 6). An added feature of this design (fig. 6) is that it will provide a dwell on the feed cycle, necessary when counterboring holes.

The choice of the number of feeds was obtained from study of figs. 7 and 8. It was decided that the limit would have to be set at a 10 t.p.i. thread (0.1 in/rev), which would facilitate tapping of 1 in BSF, and $\frac{3}{4}$ in BSW and UNC threads.

Assuming a tapping attachment is used with an end float of 0.5 in, it was found that the tapping feeds shown in fig. 8 could be grouped so as to use this facility (provided holes tapped are less than 2 in deep). Due to depression of the end float at the commencement of tapping, the slowest feed of the grouped feeds should be selected to make compensation for this during the machining cycle.

Fig. 9 shows the geometric progression of the eighteen feeds provided.

6.1. Design of gearbox

Study of figs. 7 and 8 showed that a feed range of 0.002 to 0.100 in/rev was necessary. From the previous section it was found that eighteen feeds would be sufficient to cover the range.

To obtain the step's for a geometric progression -

$$\frac{\log 100 - \log 2}{17} = \frac{2.0 - 0.3}{17} = 0.1$$

Common Ratio = 0.1 (see fig. 10).

Fig. 11 shows the selected feed rates and their groupings on the 4 shafts while fig. 5 shows how this layout can be interpreted in real terms.

The fast traverse and reverse gear is shown in fig. 6. To achieve fast traverse the clutch plate "A" is engaged on gear "e" which gives a 10:1 ratio speed increase to gear "f". Forward feed is obtained by driving straight through to gear "f" and engaging clutch B with gear "h". Reverse feed is obtained by engaging clutch B with gear "g" which then drives gear "h" through an idler.

6.2. TRIP STOPS

Using a set of stops on a gate as shown in fig. 12, each slot way would be required to switch three pneumatic sensing elements, a logic circuit being employed to select the active slot.

A signal from the tape would engage clutch A (fig. 6) in fast traverse. The first trip would switch clutch A into normal feed for drilling, whilst the second trip would switch clutch B into reverse. The third trip would be required to stop the reverse feed at the back of the quill travel.

7.0. RECOMMENDATIONS

Reference has been made to the troubles encountered in the design of the Z axis feed control. Although a gearbox has been designed for this purpose, it is far from ideal. A gearbox provides a constant feed/rev, but constitutes a large proportion of the overall size and weight of the head. It would be preferable to use some form of actuator which is not upset by temperature effects such as changing oil viscosity.

It must be appreciated that these problems would all be overcome if the Z axis feed depth and feed rate were selected direct from the tape on a closed loop control system. This requires a velocity and position feedback transducer on the quill. Since the positioning on the X and Y axes is achieved by fluidics, it would be desirable for the Z axis transducer also to be fluidic.

Since a fluidic velocity control system is not available and the use of electrical systems is not desired the Z axis must remain open loop.

In order to improve the Z axis feed control, it is recommended that future investigations be directed along the following lines :-

7.1. PNEUMATIC FEED

It has been stated that the chief problem associated with hydraulic feed, was the change in oil viscosity due to temperature rise, which affected the flow rate through the control valves.

To overcome this, oil could be replaced by air, in which case the flow rate through the valves would be more independent of temperature change. It has been suggested that the valves be replaced by a series of orifices whose flow rates ascend according to a binary code. (see fig. 13). In this way 32 discrete flow rates could be achieved from 6 holes; the rate being selected by opening the relevant holes direct from a tape signal or manually. A possible problem here could be the lower stiffness of a pneumatic cylinder.

7.2. SPECIAL TAPPING ATTACHMENT

It is true to state that the feed rates are most critical during the tapping operation. With this in mind it is suggested that a special tapping attachment be designed in which an axial thrust transducer be incorporated in the tap holder, which could sense whether the tap is being fed faster or slower than necessary. The output of this transducer could then be employed to adjust the feed control valve to a hydraulic actuator. In this way the difficulties of accurately setting the correct feed rates and also allowing for variations in feed rate due to changes in oil viscosity could be compensated within the closed loop. Setting stops are still necessary, however, for depth control.

8.0. REFERENCES

1.	Boshier G.C. & Pike G.W.H.	Drill Head Committee Report No. 1 Main Spindle Drive Production Technology Project 1967.
2.	Koenigsberger, F.	Design Principles of Metal Cutting Machine Tools. Pergamon Press.
3.	Ryder, G.H.	Strength of Materials. Cleaver Hume.
4.	-	S.K.F. Ball & Roller Bearings General Catalogue No. GB 500. The Skefco Ball Bearing Co. Ltd.

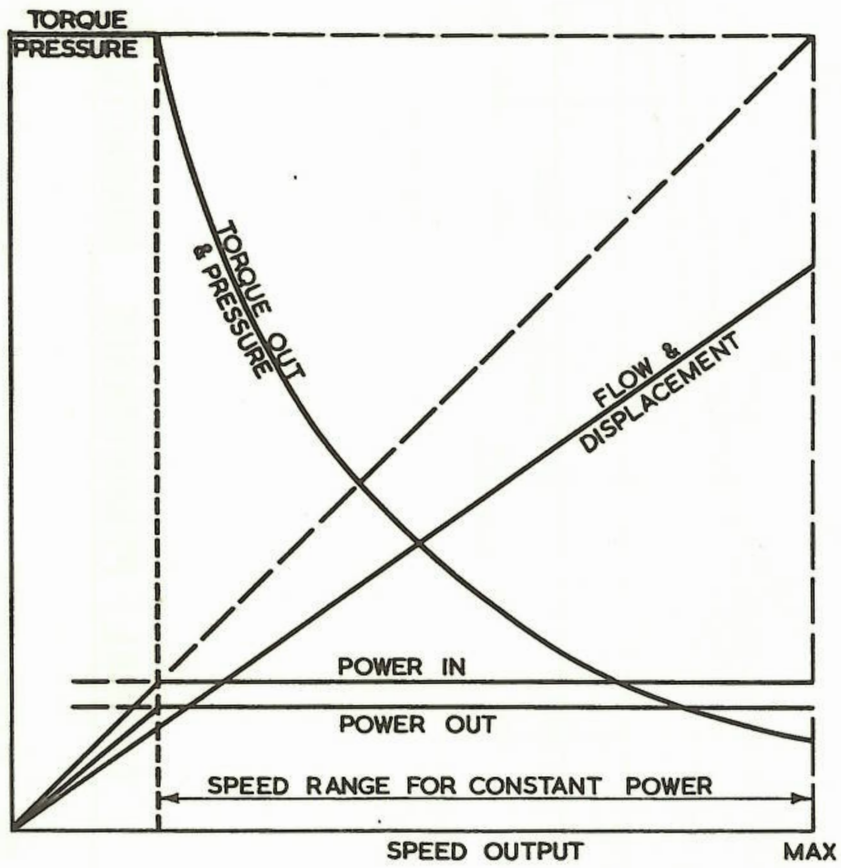


FIG.1. CHARACTERISTICS OF VARIABLE DISPLACEMENT PUMP. FIXED DISPLACEMENT MOTOR UNIT.

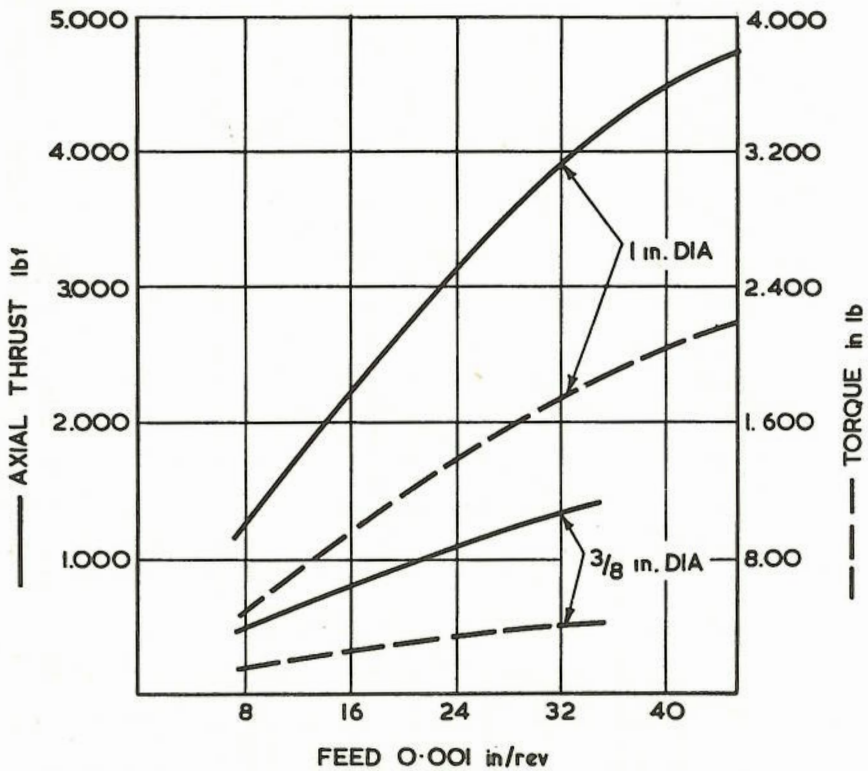


FIG.2. TORQUE & AXIAL THRUST WHEN DRILLING 38 $tonf/in^2$ CARBON STEEL.

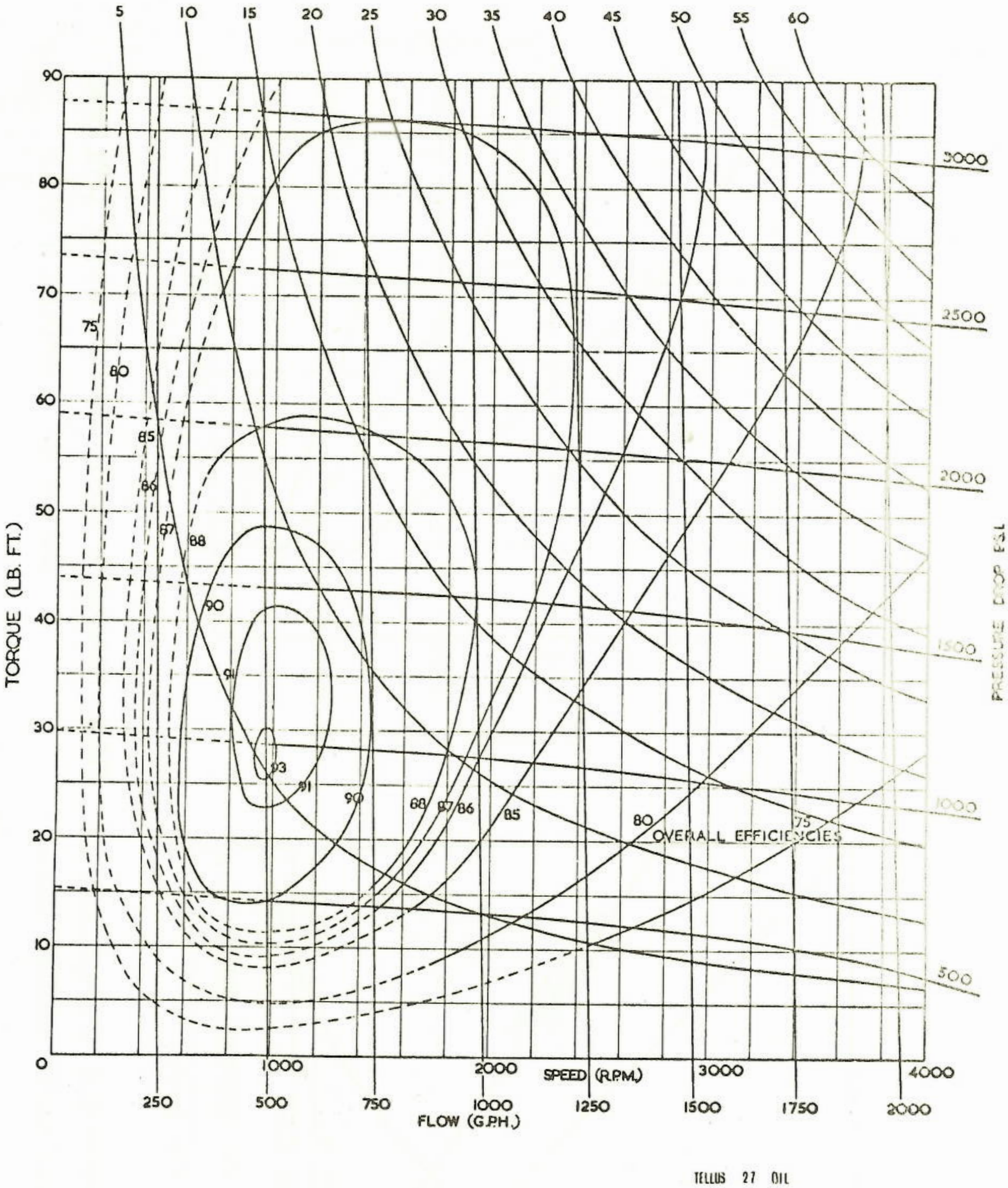


FIG. 3 TYPICAL PERFORMANCE CHARACTERISTICS OF HYDRAULIC MOTOR SELECTED

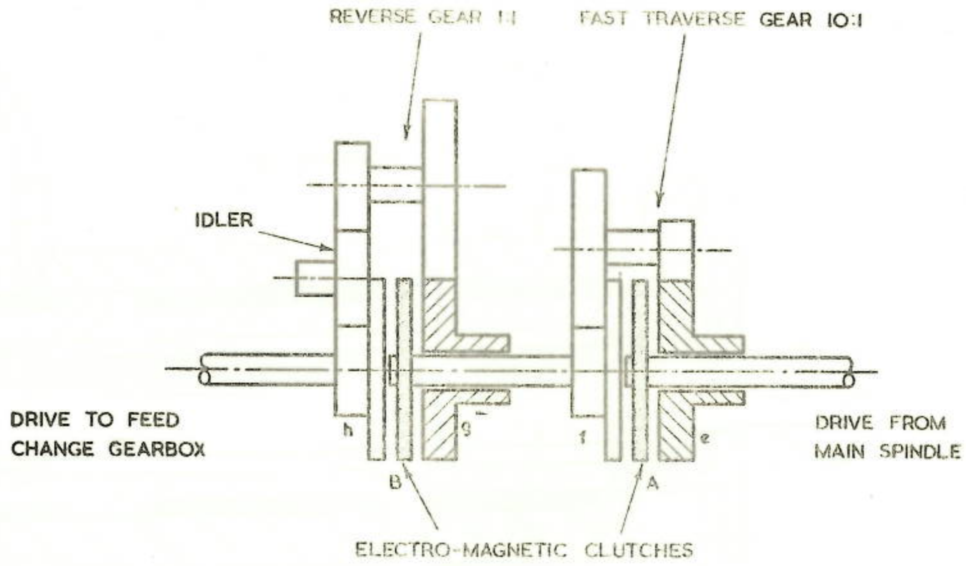


FIG.6. DIAGRAM SHOWING FAST TRAVERSE & REVERSE FEED FACILITY.

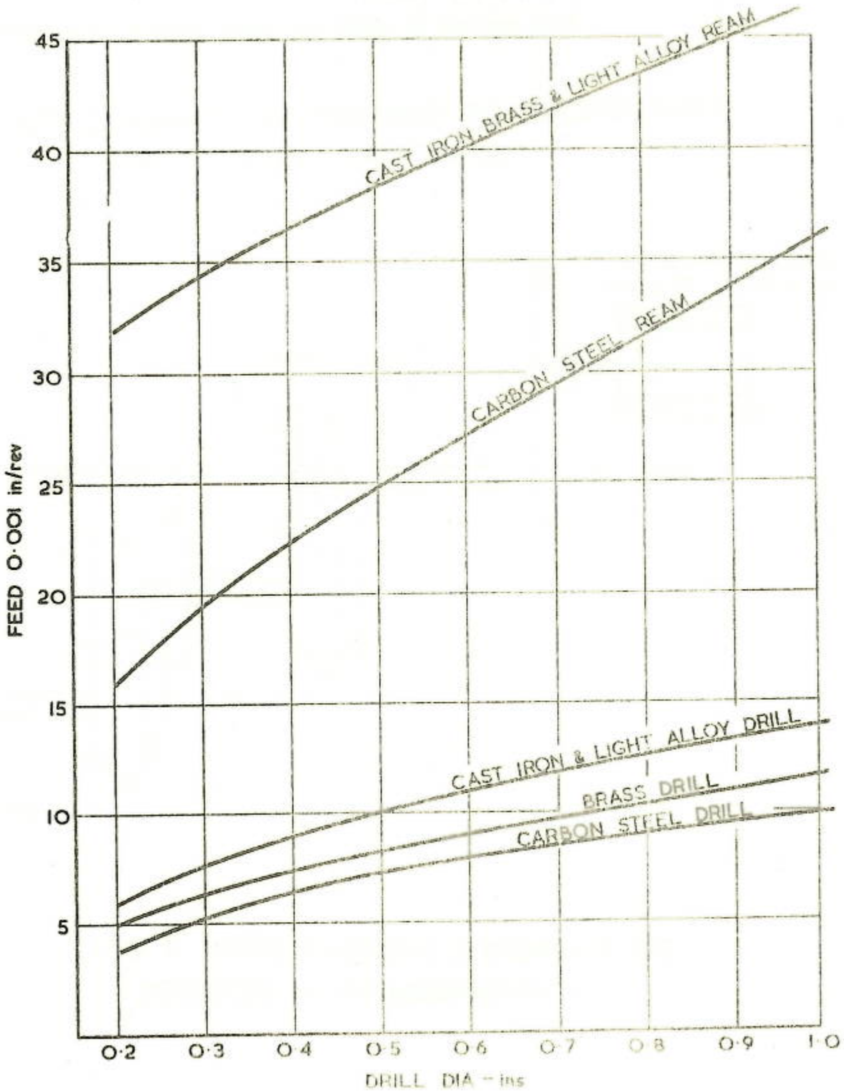
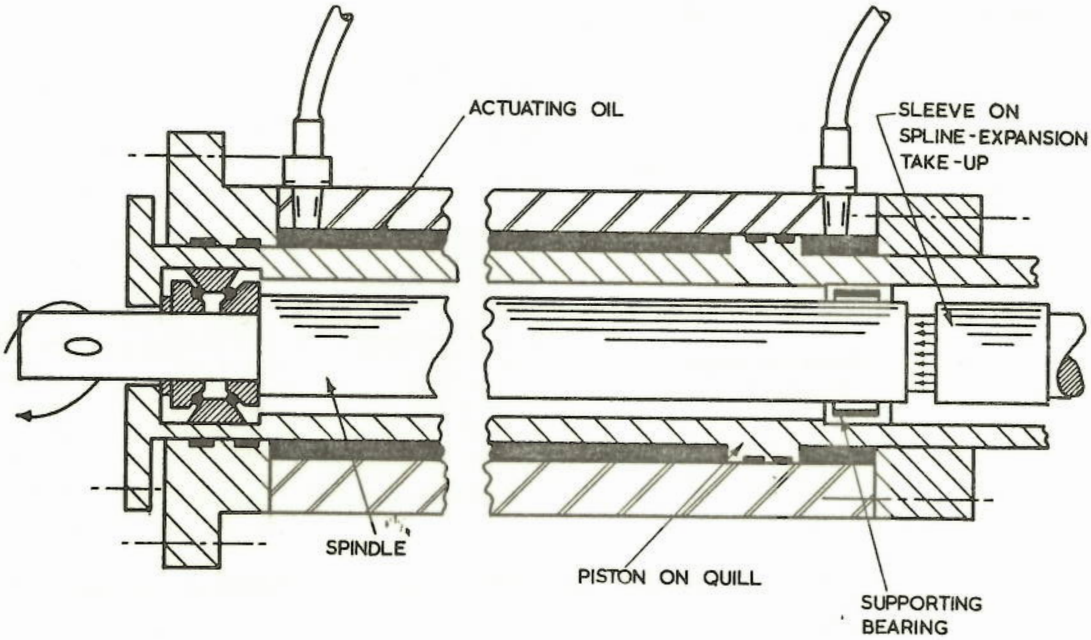


FIG.7. DRILL DIAMETER/FEED RELATIONSHIP.



THIS SYSTEM IS USED ON 'VERO' 15'24 & 20'40 M/C's

FIG.4. SKETCH OF CONSIDERED HYDRAULIC FEED USING QUILL AS A PISTON.

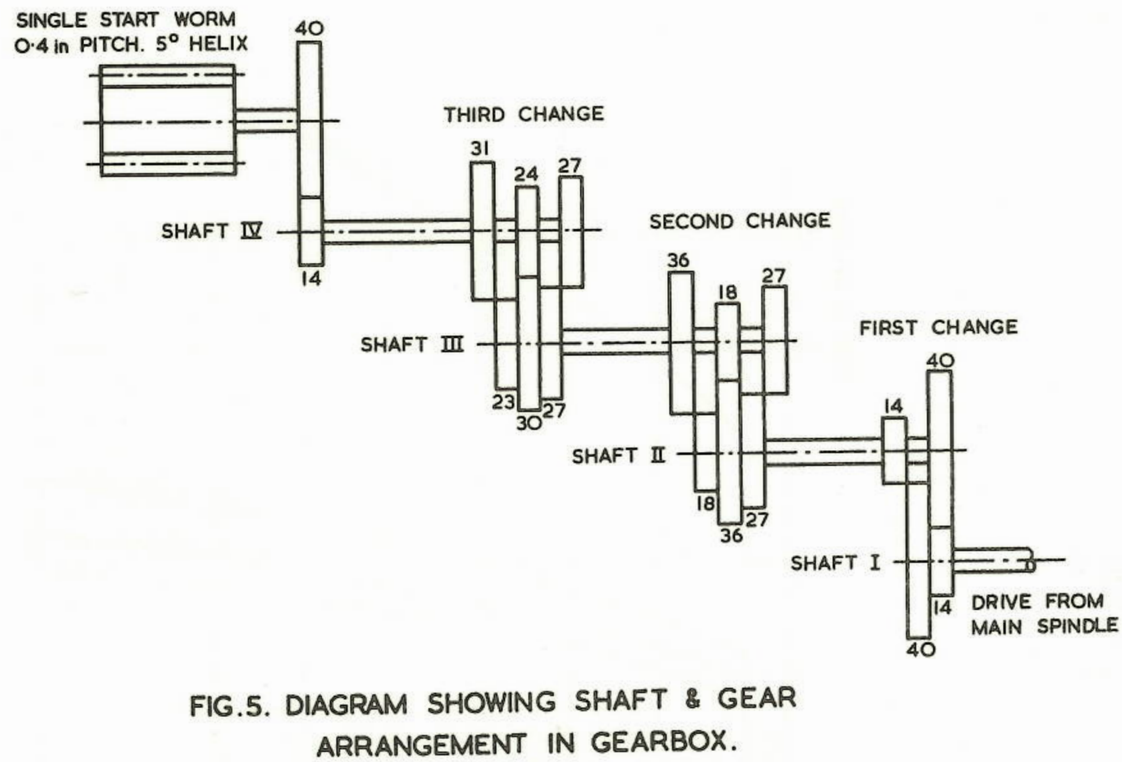


FIG.5. DIAGRAM SHOWING SHAFT & GEAR ARRANGEMENT IN GEARBOX.

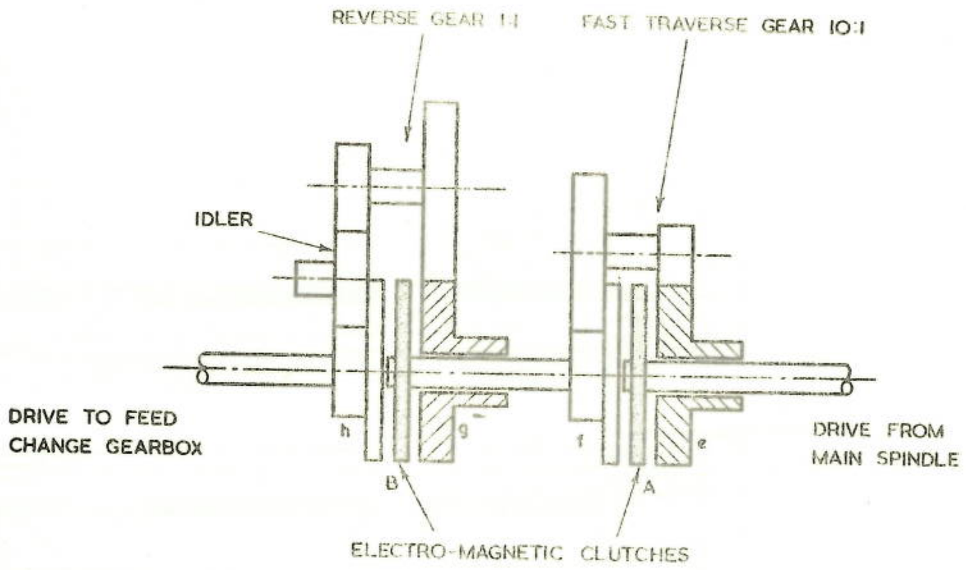


FIG.6. DIAGRAM SHOWING FAST TRAVERSE & REVERSE FEED FACILITY.

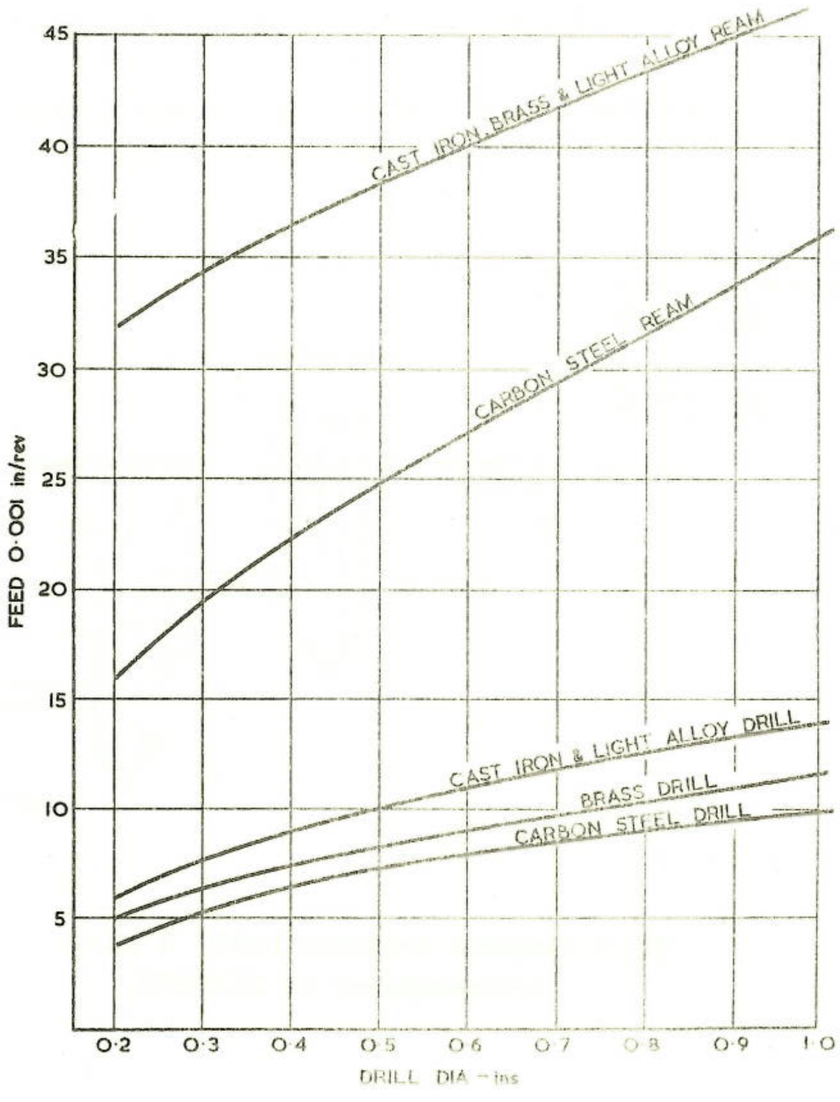


FIG.7. DRILL DIAMETER/FEED RELATIONSHIP.

DIA	THREADS PER INCH				
	BSF	BSW	UNF	UNC	BSP
1/8		40			28
3/16	32	24			
1/4	26	20	28	20	19
5/16	22	18	24	18	
3/8	20	16	24	16	19
7/16	18	14	20	14	
1/2	16	12	20	13	14
9/16	16	12	18	12	
5/8	14	11	18	11	14
3/4	12	10	16	10	
7/8	11	9	14	9	14
1	10	8	12	8	11

FIG.8. CHART SHOWING TPI FOR VARIOUS THREAD DIAMETERS.

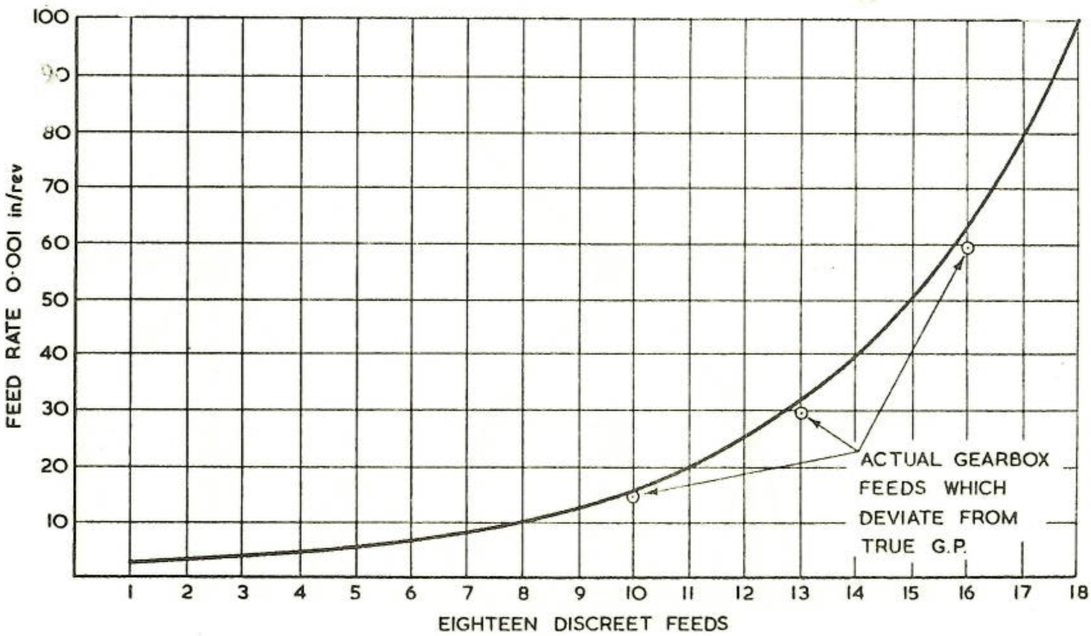


FIG.9. GRAPH SHOWING GEOMETRIC PROGRESSION OF GEARBOX FEEDS.

LOG$_{10}$ FEED	FEED 0·001 in/rev		T.P.I.
	THEORETICAL	ACTUAL	
2·0	100·0	100·0	10
1·9	79·4	80·0	11 & 12
1·8	63·1	59·3	13.14. & 16
1·7	50·1	50·0	18.19. & 20
1·6	39·8	40·0	22 & 24
1·5	31·6	29·7	26 28 & 32
1·4	25·1	25·0	40
1·3	19·9	20·0	
1·2	15·8	14·9	
1·1	12·5	12·3	
1·0	10·0	9·8	
0·9	7·9	7·3	
0·8	6·3	6·1	
0·7	5·0	4·9	
0·6	3·9	3·6	
0·5	3·1	3·1	
0·4	2·5	2·5	
0·3	2·0	1·8	

FIG.10. TABLE SHOWING RELATIONSHIP BETWEEN THEORETICAL GEOMETRIC PROGRESSION & FEEDS PROVIDED.

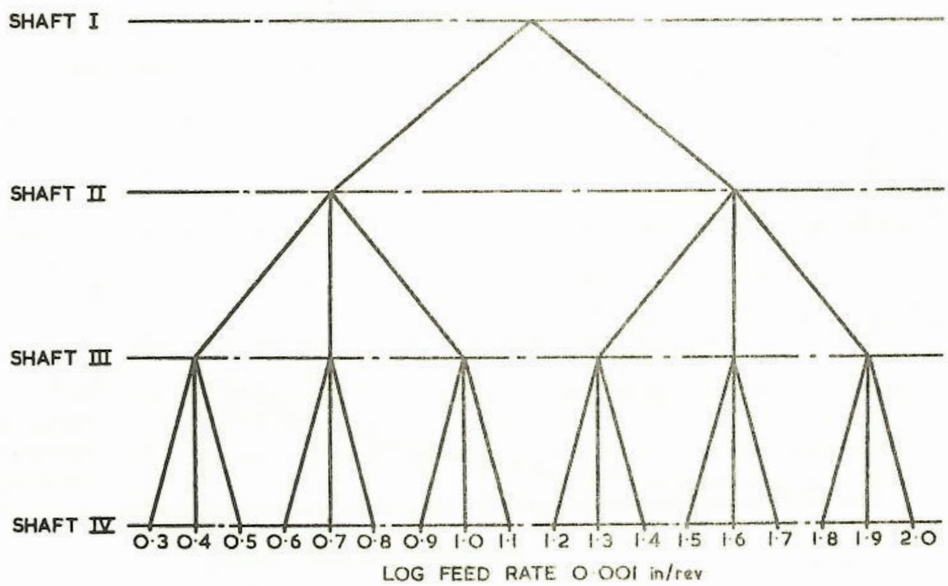


FIG.11. LAYOUT FOR FEED GEAR DRIVE.

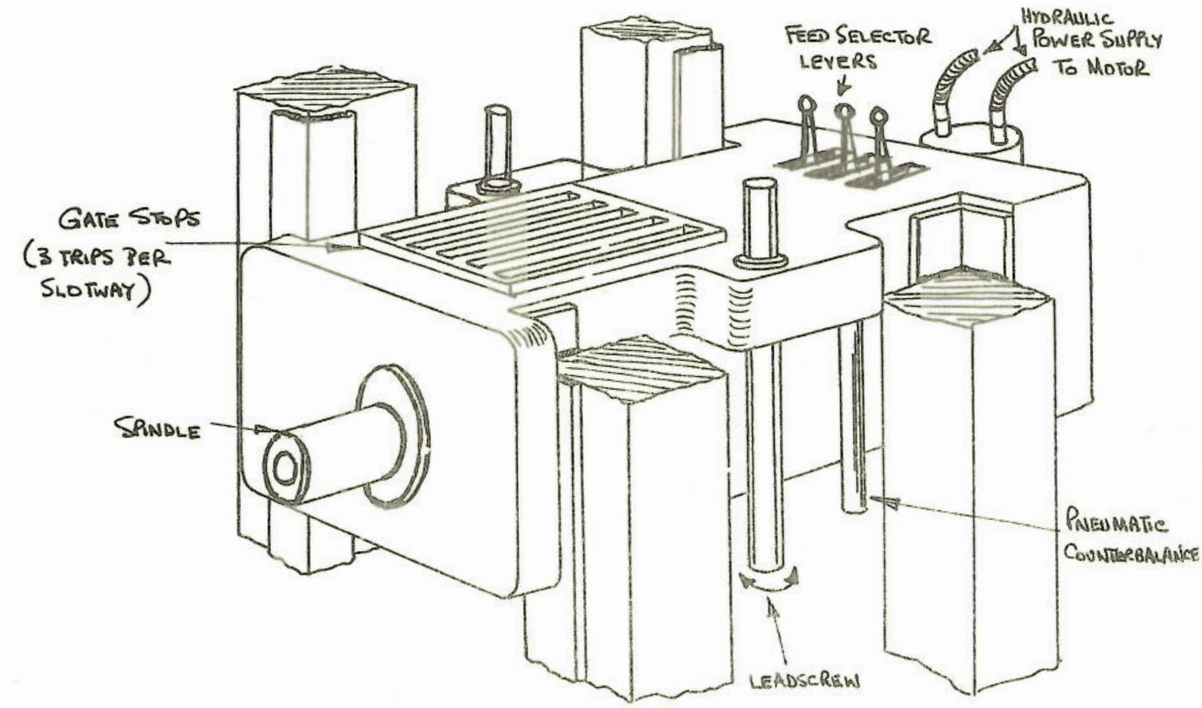


FIG 12. SKETCH OF DRILL HEAD ASSEMBLY.

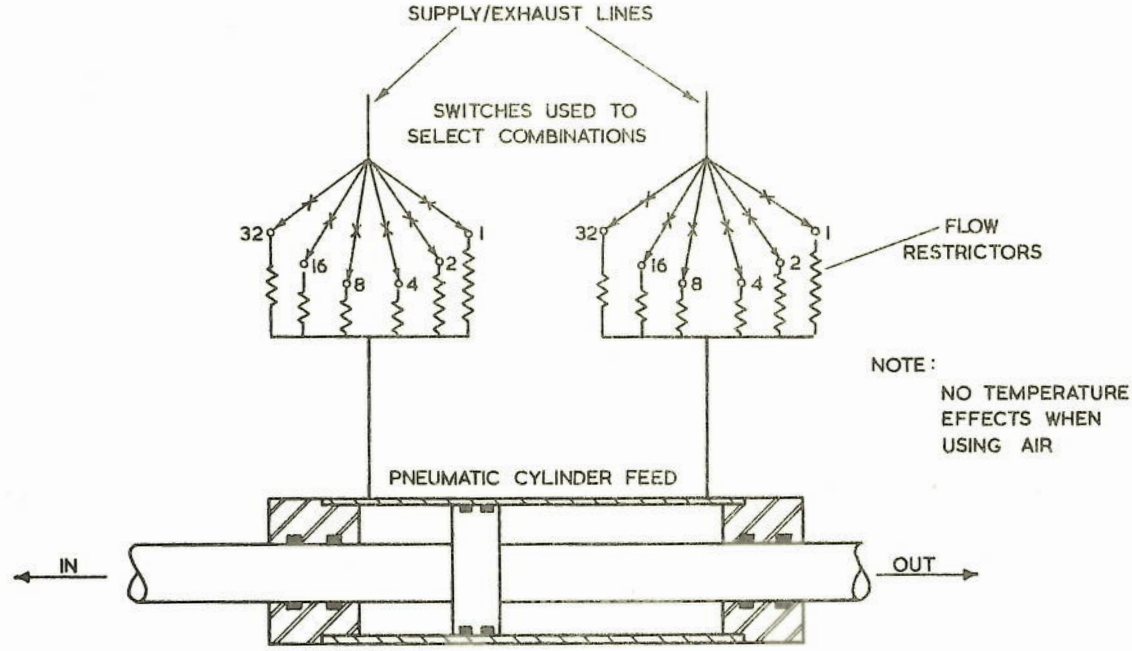


FIG.13. POSSIBLE ALTERNATIVE TO GEARBOX FEED.