THE COLLEGE OF AERONAUTICS

CRANFIELD

DEPARTMENT OF PRODUCTION ENGINEERING

DRILLING SYSTEM DESIGN PROJECT 1967

FINAL REPORT OF FRAME DESIGN COMMITTEE

Compiled by

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Summary

This report outlines the recommendations of the Frame Design Committee for the final design of the machine, each major part of the structure being considered individually in the following sections :

1. Worktables
2. Guide and Slideways
3. Drill Head Support Structure
4. Swarf Disposal and Coolant Supply
5. General Construction

CONTENTS

1.0. INTRODUCTION

The members of the Committee were responsible for the general design of the machine, see fig. 1, with the exception of the drill head and the machine control system. Throughout the project there was close consultation with both Drill Head and Control System Committees in order that the design requirements for certain parts of the frame were met.

This report outlines the recommendations of the Frame Design Committee for the final design of the machine; each major part of the structure being considered individually in the following sections.

2.0. WORKTABLES

The machine is to have two individual worktables, the configuration of which is described in the Frame Design Committee's Report No. 2. Worktable Design (appendix 1). Each table can be raised and lowered independently of each other and when a table is in the drilling position, two hydrostatic shot bolts will be positioned on either side of the table, to give a positive location. The main pivot bearing required careful selection for reasons of minimum wear, ease of replacement, and also to ensure that sine and cosine errors were eliminated as far as possible. It is recommended that a self-aligning type ball bearing is used. The table positioning boomerang arms are connected one to the other by means of a plate which also acts as a swarf disposal chute (functioning when the table is in the loading position). These positioning arms, together with their hydraulic actuators, are situated so as to allow the machine operator adequate leg room when loading and unloading, whilst the table itself gives the operator protection from the moving parts of the machine.

It is the opinion of the Committee that a potential customer should be given a choice of workpiece clamping methods i.e. tee slots, a network of tapped holes, pallet system etc. This has the advantage of allowing the customer to select his system in relation to existing machinery and drawings.

3.0. GUIDE AND SLIDEWAYS

A major requirement of the control system committee was that the stiction between the slideways and their bearings be a minimum and that there should be good friction characteristics i.e,

a) There should be no initial negative slope

b) The absolute level should be as low as possible

c) The separation of the bearing surfaces should not increase with speed.

d)   A small increase in drag with speed to help the stability of the
       servomechanism.

With these requirements in mind the committee considered five
alternatives, these being :-

1)   Conventional cast iron slides lubricated with oils having special
       chemical properties.

2)   Dry low friction materials.

3)   Rolling element bearings.

4)   Aerostatic bearings (using air).

5)   Hydrostatic bearings (using oil).

These types of bearing, are now discussed for their suitability for
the duties they have to perform. Type 1 was eliminated because, even
though the major oil companies have developed special types of oil for
slideway applications satisfying the requirements (a to d) above, it is
known that these oils tend to lose some of their properties when subjected
to high shear stresses. Also it was thought that prospective customers
might object to having to stock quantities of special and relatively
expensive oil for a limited number of machines.

The type of dry low friction material considered was PTFE im-
pregnated bronze.

This material has two major disadvantages in that little is known
on its performance under workshop conditions, and if it is used unlubricated
its friction factor increases and as much as 0.002 in wear can occur after
150,000 ft of sliding.

Although when lubricated the coefficient of friction is very low,
considerable wear is still considered likely to take place, especially
during "running in".

On the credit side, with regard to initial fitting and subsequent
servicing, this material is supplied in thin strips on a steel backing,
and relatively inexpensive.

Because of disadvantages outlined above, it was not considered
practicable to use this material for the horizontal slideway which in
particular would be susceptible to damage by swarf. However, as the
vertical slideways are not high load carrying, it was thought that this
material would be suitable and the committee recommends that it should
be used for the vertical slideways.

Rolling element slides have been successfully used in many machine

tool applications. The rollers give very long service before they eventually wear, and then they are very easily replaceable.

For the purposes of this machine, freedom from stiction is very important, and this necessitates careful shielding of the bearing surfaces from dust and swarf.

This problem may be overcome by fitting thin flat coil spring covers at each end of the bed. These unwind to cover the bearings as the structure moves along the bed. Alternatively, the standard flexible 'bellows' type of covers may be employed.

The Committee decided to reserve judgement on this type of bearing, because it considered that some practical measurements of stiction would be required for the combined bearing and cover arrangement.

Much has been written outlining the advantages of hydrostatic bearings (Ref. 1). The Committee therefore decided to compare the relative merits of bearings operated with air and oil.

In the present situation, air gives the lowest drag but requires more pumping power than oil. Oil however produces scavenge problems that do not exist with air. Also there must be adequate filtration of the returning oil to remove swarf. There would also be the problem of separating the coolant from the bearing oil.

For safety reasons, air at pressures in excess of 200 lbf/in^2 is not recommended for general factory use; however, for hydrostatic bearings, pressures of 400 lbf/in^2 and greater are quite acceptable. It is expected that the maintenance of an air system would be less than that of an oil system. However, a final decision is made based on bearing area calculations, see Appendix III.

DRILL HEAD SUPPORT STRUCTURE

The estimated weight of the drill head is 400 lbf and the maximum estimated thrust exerted by the drill 4000 lbf (Ref. 2).

The function of the structure is to equilibrate these forces whilst allowing the drill head to move freely in the vertical plane with the minimum of positional error.

To achieve this the committee decided to use four integrated support columns having a slieway on each and positioning the drill head between them (fig. 1). The P.T.F.E. impregnated bronze bearing surfaces as recommended for this part of the machine are situated as shown on the figure. The design of the structure is considered in Appendix II.

The counterbalance to the drill head is to be provided by a pneumatic system, the two cylinders of which will be situated one on either side of the drill head, between the front and rear columns of the structure. The opinion of the committee was that the pneumatic system would be less expensive than the traditional counterbalance weight where the cost of the lead alone would be approximately £30.

The Control Committee decided that the drill head should be positioned by means of a single leadscrew, actuated by a fractional horse-power electric motor with a gearbox at one end and having the fluidic position encoder at the opposite end. To safeguard the fluidic unit from loose particles of swarf and coolant mist etc., the encoder was placed on the top of the machine, thus leaving the motor and gearbox in a position between the front and rear columns and underneath the leadscrew connection to the drill head.

SWARF DISPOSAL AND COOLANT SUPPLY

As stated in section 2, the 'boomerang' arms-connecting plate acts as a chute for the swarf when the table rises to the loading position, but when in the drilling position tends to collect the swarf as it falls. Because of the slight inclination the coolant flows away via the conveyor belt and through a coarse filter, back to the reservoir. A splash guard is fitted on the front of the main drill column to protect the slideways. The conveyor belt runs the length of the machine and deposits the swarf into a collection bin.

The coolant supply, to be most effective, is required to move relative to the movement of the cutting tool. The supply nozzles to the workpiece are therefore attached to the drill head and because of the spare capacity of the hydraulic motor driving the spindle, it is proposed to attach the coolant pump to the drill head and to drive it off the bevel hearing. This method would have the added advantage of stopping the coolant supply when machining stops.

GENERAL CONSTRUCTION

It has been increasing policy of some machine tool manufacturers to use welded steel construction as opposed to iron castings for their structures.
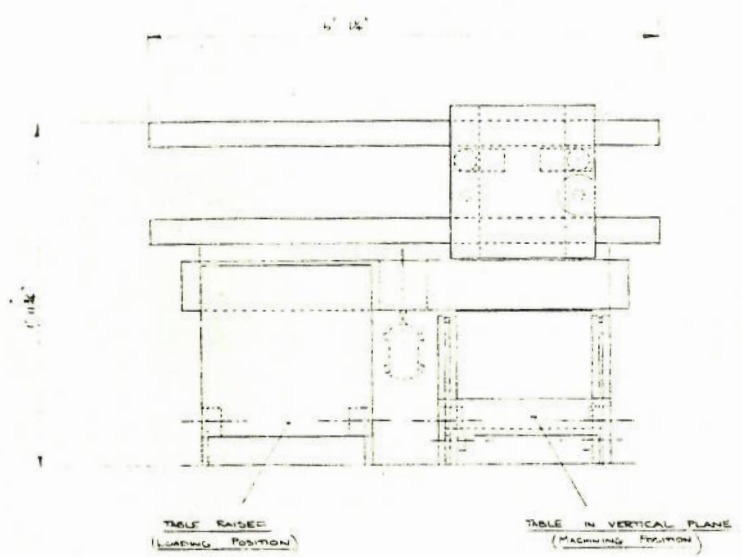
Welded steel structures have many advantages over the equivalent cast iron structures, especially when a limited amount of components and slight variations in design are required.

This machine has been designed with unit construction in mind, therefore, a family of machines could be manufactured using standard parts e.g. drill head, column base, table actuators, etc. Having these features in mind the Committee recommended that the machine should be basically of welded steel construction.
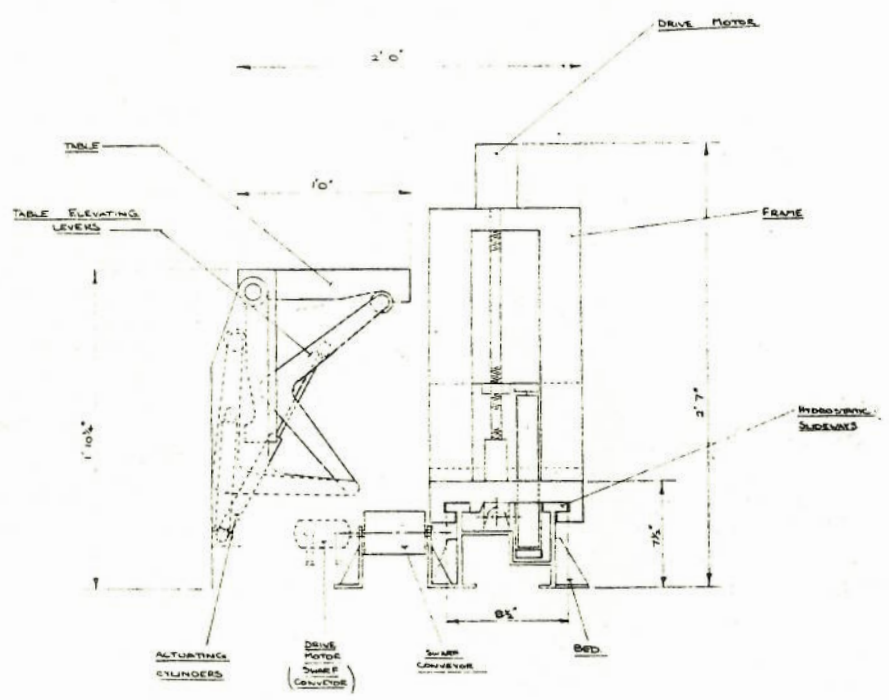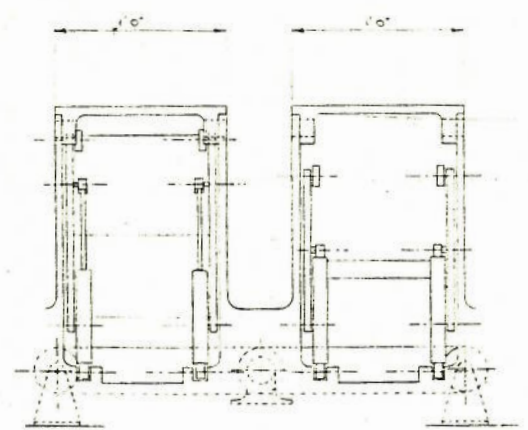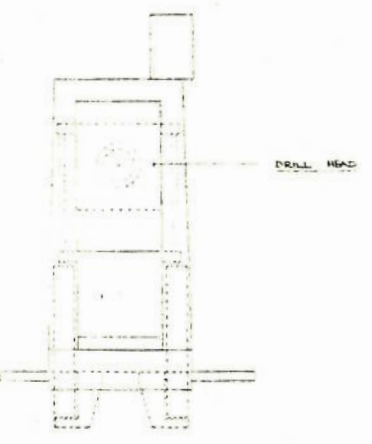
In conclusion the members of the Committee wish to thank Mr. Peter Cooke for the advice and assistance he has given relating to the structural design of the machine.

REFERENCES

1. Loxham J and   The application of hydrostatic bearings to
    Hemp J.   high precision machine tools.
                                 Coll. of Aero. Memo No. JL1306

2. Koenigsberger, F.   Design Principles of Metal Cutting Machine
                                 Tools.

C.A. FIG. 1

TABLE RAISED
(LOADING POSITION)

TABLE IN VERTICAL PLANE
(MACHINING POSITION)

DRILL HEAD

DRIVE MOTOR

TABLE

FRAME

TABLE ELEVATING
LEVERS

HYDROSTATIC
SLIDEWAYS

ACTUATING
CYLINDERS

DRIVE
MOTOR
(SWARF
CONVEYOR)

SWARF
CONVEYOR

BED

APPENDIX I

THE COLLEGE OF AERONAUTICS, CRANFIELD

DEPARTMENT OF PRODUCTION AND INDUSTRIAL ADMINISTRATION

DRILLING SYSTEM DESIGN PROJECT

FRAME DESIGN COMMITTEE REPORT NO. 2

WORKTABLE DESIGN PART 2

W. Morrison & R.S. Sutcliffe

1. INTRODUCTION

Part 1 of this report outlined the general proposals concerning the design
of the worktable. Fig. 3 of that part showed the tables being raised upwards
from a horizontal (loading) position. A subsequent decision has been taken to
arrange for the table to lower to the vertical (drilling) position. This has
two advantages: the table can be made to rest against a firm support during
the drilling operation, and the overall height of the drilling column can be
reduced by 30 per cent. The following sections describe some detailed aspects
of the proposed design for the worktable, the means by which it is operated,
and the general arrangement of the whole machine.

2. WORKTABLE RAISING AND LOWERING MECHANISM

The general arrangement of the machine (Fig. 1) shows each table being
moved using two connected boomerang shaped levers, and actuated by a
hydraulic cylinder on each lever. This configuration was chosen so that all
moving parts were kept clear of the operators working area. It has the
added advantage that the boomerang lever acts as a mechanical support for the
table when it is in the horizontal position. It is also a safety device in
the event of a hydraulic failure.

The two levers are connected by a plate which also acts as a swarf
collection tray during drilling. The swarf is then tipped onto a conveyor
belt when the lever is raised to the loading position.

The loaded worktable when lowered is locked against the main frame
by a hydraulic lock which then gives a signal to the hydraulic circuit
operating the drilling head.

3. TO DETERMINE THE REQUIRED SIZE OF HYDRAULIC CYLINDER

Fig.2 shows the force system to be considered for the determination of
the size of hydraulic cylinder required to operate the loaded table. W_1

is the effective workpiece weight, which is calculated to include the
worktable.

3.1. Effective workpiece weight

$$\text{Actual maximum workpiece weight} = 0.25 \times 18^2 \times 12$$

$$= 324 \times 3$$

$$= 972 \text{ lbf.}$$

$$\text{Worktable weight} = 0.25 \times 24^2 \times 1.5$$

$$= 144 \times 1.5$$

$$= 216 \text{ lbf.}$$

$$\text{Worktable weight referred to centre of gravity of workpiece} = \frac{216 \times 1.5}{9}$$

$$= 36 \text{ lbf}$$

Total effective workpiece weight = 1008 lbf.

Assume a total weight of 1200 lbf.

3.2. To find the cylinder load variation as the table is raised.

Again from Fig. 2, and assuming that all the load is supported by one
cylinder.

$$W_1 r_1 = W_2 r_2 \qquad \& \qquad W_3 r_4 = W_2 r_3$$

so that $W_3 = \dfrac{W_1 r_1 r_3}{r_2 r_4}$

The term $\left(\dfrac{r_1 r_3}{r_2 r_4}\right)$ may be computed for different values of table angle θ,
and plotted graphically for the chosen configuration, this graph is shown
in Fig. 3. It can be seen that there is a maximum at a table angle of
about 45° from the vertical. At this position the single cylinder load is
4.33 times the effective workpiece load i.e. 5200 lbf. In the actual
design there are two cylinders therefore each one supports 2600 lbf.

3.3. Choice of cylinder size.

Two factors must be considered; the hydraulic system pressure and the

Euler crippling load.

For the latter, the Euler crippling load must be greater than the actual load; so that :-

$$W < \frac{\pi^2 E I}{l^2}$$

For a round rod $\quad I = \dfrac{d_p^4}{64}$,

where d_p is the piston rod diameter.

$$\therefore W = \frac{\pi^3 E d_p^4}{64\, l^2}$$

For steel $E = 30 \cdot 10^6$ lbf/in^2, $\quad l = 15$ in & $W = 2600$ lbf.

$$d_p^4 = \frac{2600 \cdot 64 \cdot 15^2}{\pi^3 \cdot 30 \cdot 10^6}$$

$$d_p^4 = 0.040 \text{ in}^4$$

$$d_p = 0.447 \text{ in}$$

If a safety factor of 2+ is used $d_p \doteq 1$ in.

If the system hydraulic pressure is 2000 lbf/in^2 the cylinder area should be $\dfrac{2600}{2000} = 1.3$ in^2

If d_c = cylinder diameter

$$\frac{d_c^2}{4} = 1.3$$

from which $d_c = 1.29$ in.

From the above analysis it would appear that cylinder piston rod is the dimension of major importance and this should be not less than 1 in. The cylinder diameter is then governed by availability and the system hydraulic pressure.

4. THE HYDRAULIC CIRCUIT

Fig. 4 is a circuit diagram for the hydraulic operations of the worktables, and the interlocking features associated with the movement of the drilling column.

5.2.   Boomerang lever

Fig. 5 shows the forces acting on the boomerang lever at the maximum condition. These may be resolved normally and tangentially to the particular section of the lever.

The main forces to be considered are those normal to the beam section. The maximum bending moment may be found (as shown on the force diagram), and from

$$\frac{M}{f_t} = \frac{I}{y} = Z$$

$$Z = \frac{9900}{15000} = 0.66$$

Z for a rectangular section is $\frac{b_d{}^2}{16}$

so assuming d = 4 in, b = 0.66 in.

5.3.   Boomerang lever hinge pin.

Each pin has a shearing force of

$$\sqrt{3910^2 + 649^2}$$

$$= \quad \sqrt{1529 + 42 \times 10^2}$$

$$\sqrt{15.71 \times 10^3}$$

$$= \quad 3963 \text{ lbf.}$$

With a safety factor of 3 hinge pin diameter d_b

$$d_b \geqslant \sqrt{\frac{4}{11} \, \frac{11889}{11200}}$$

or approximately $\sqrt{\frac{4}{11}}$ $=$ $\sqrt{1.272}$

$= \quad 1.269$

say   1.25 in

5.4.   Worktable frame support for drilling wads.

In this case the bending moment is assumed to be entirely due to the proportion of the drilling force transmitted through the table hinge pin   i.e. 1000 lbf at 10 in radius. Depending upon the final design

and method of construction of the main frame, this bending moment will be easily accommodated.

6.   CONCLUSIONS

The basic design parameters for the worktable configuration have been evaluated. A general arrangement has been suggested, and the hydraulic circuitry for the operation of the worktable has been given.

These proposals have been made based on the general concept of the drilling machine as at present: they are readily adapted to changes in design.
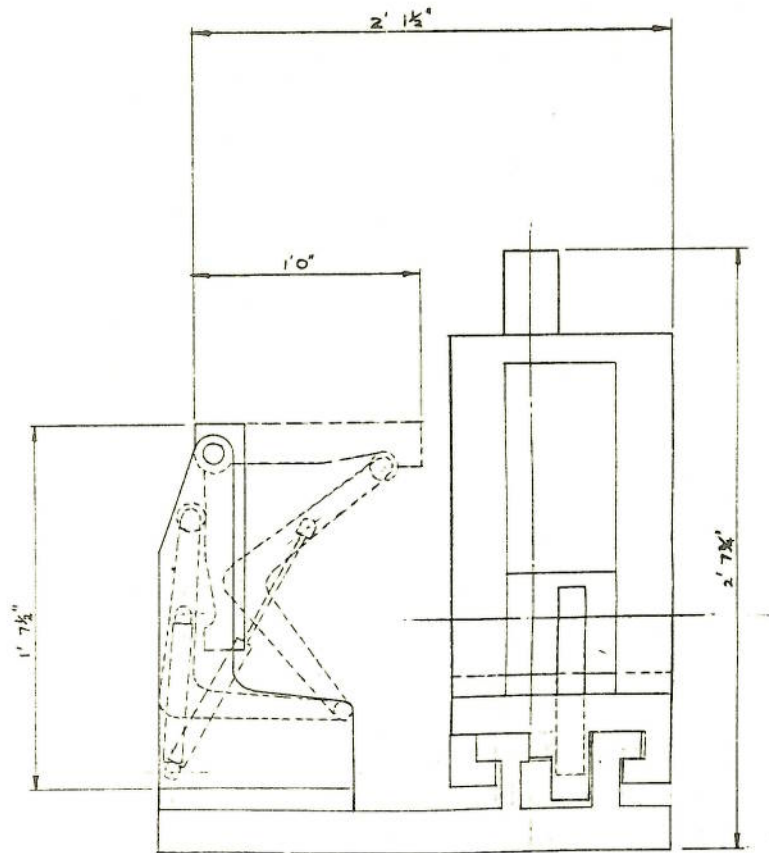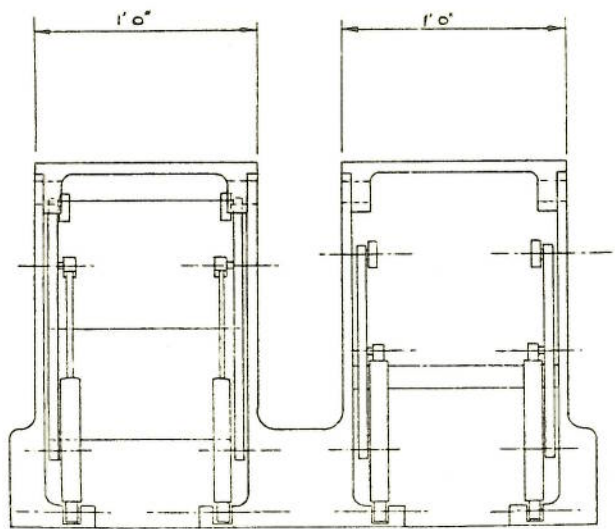
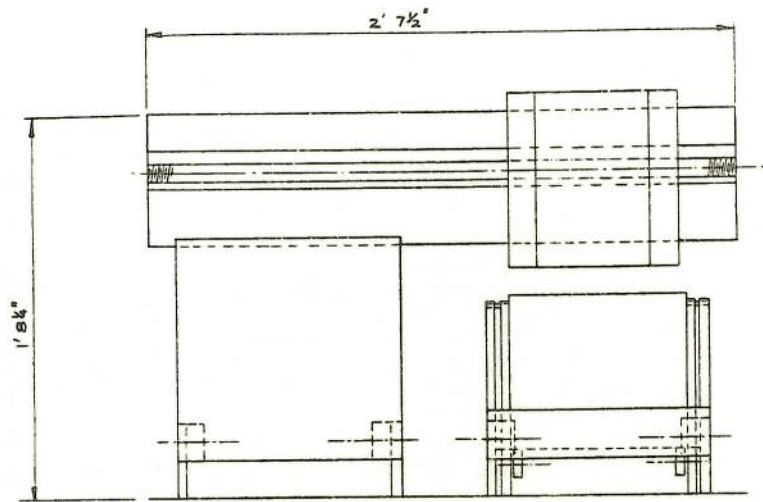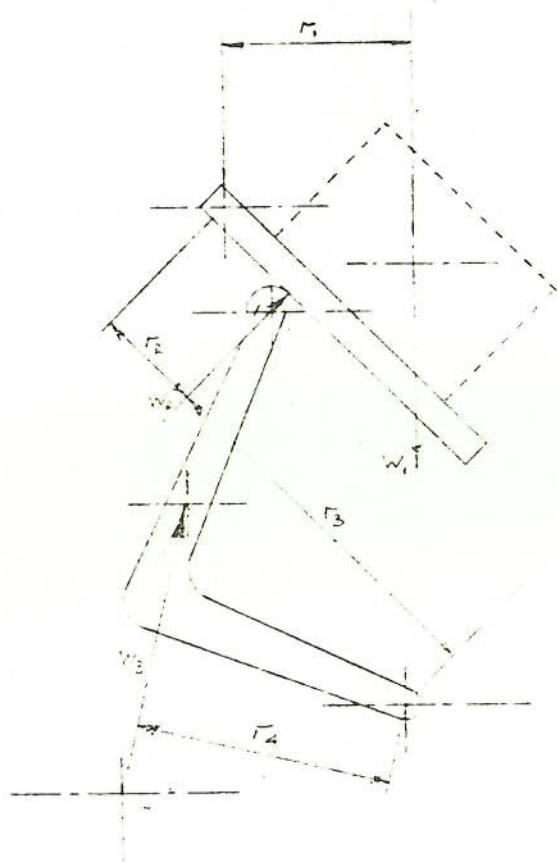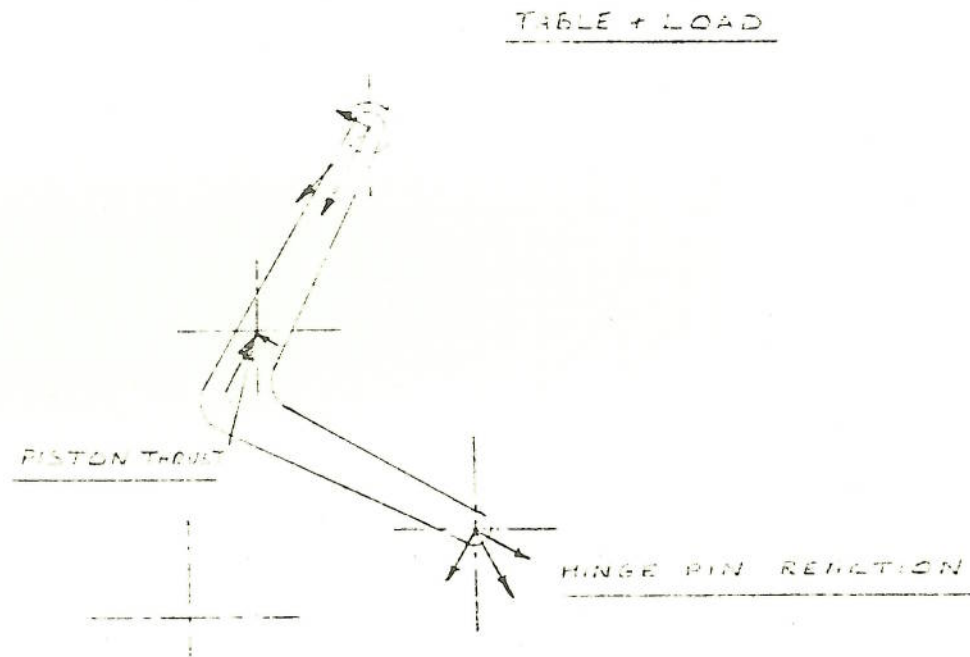FIG. 1. PROPOSED GENERAL ARRANGEMENT OF DRILLING MACHINE.

FIG 2

TABLE + LOAD

PISTON THRUST

HINGE PIN REACTION

FIG 5

Fig 3

SINGLE RAM FORCE / WORKPIECE WEIGHT

TABLE ANGLE FROM VERTICAL , degrees

DRILLING HEAD
STATION A
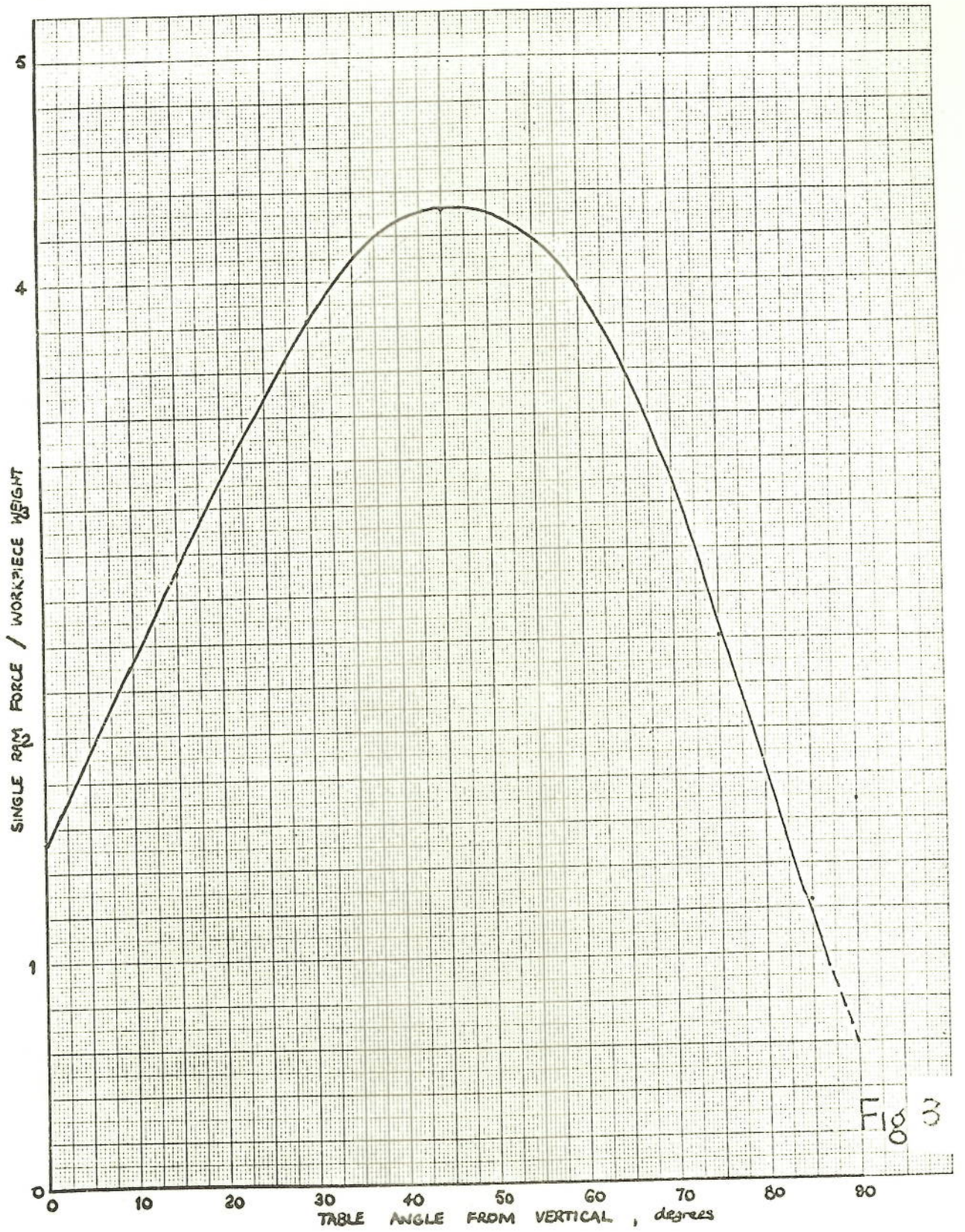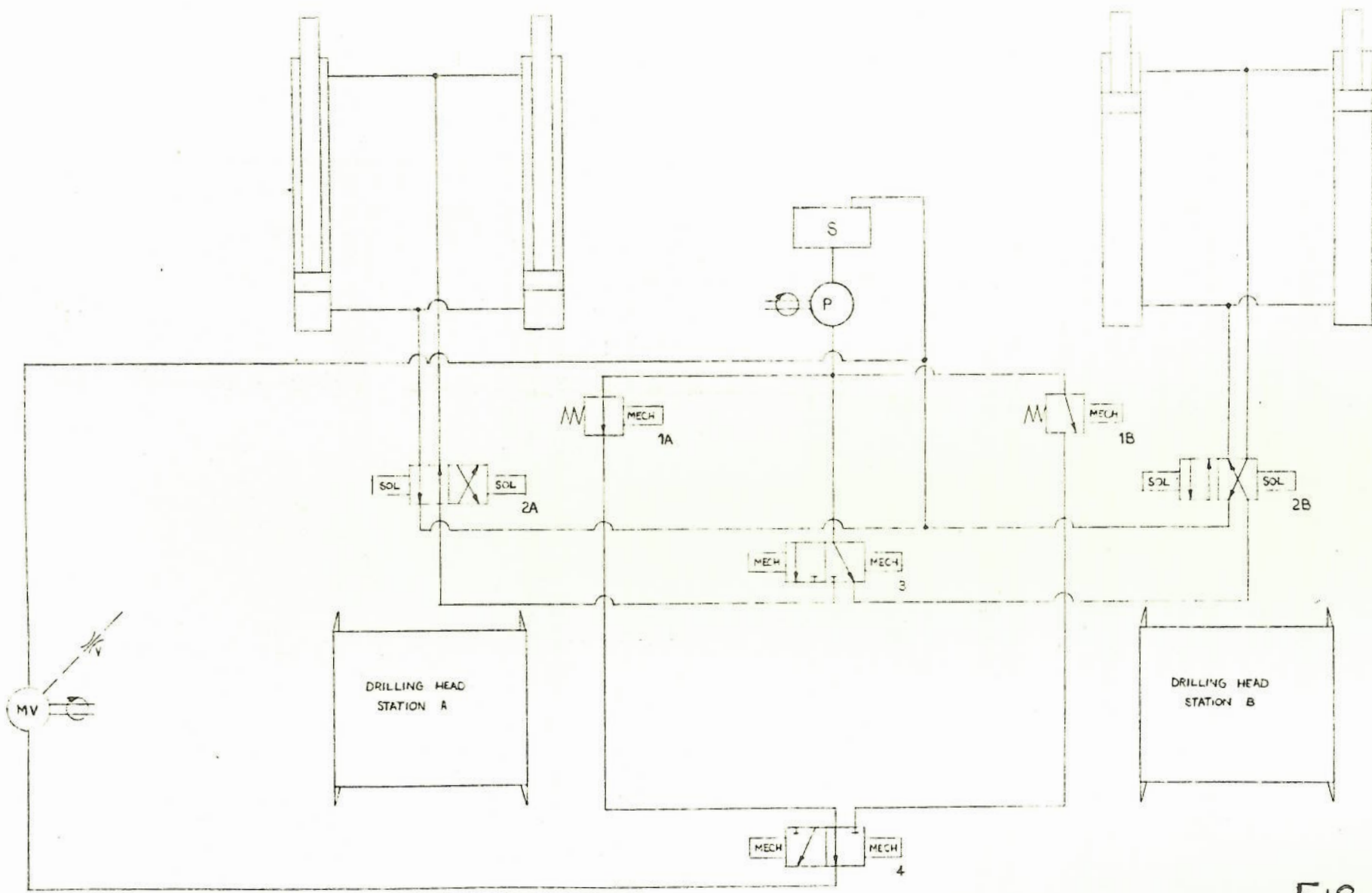
DRILLING HEAD
STATION B

FIG. 4

APPENDIX II

Deflections of Drill Head Support Columns due to
Thrust exerted by Drill

The expected thrust exerted by a 1in diameter drill whilst drilling carbon steel, up to 38 tonf/in^2, at a feedrate of 0.035 in/rev has been calculated by Koenigsberger (ref. 2) to be 4000 lbf.

It is assumed that the four columns are equally loaded due to this thrust, and that it has its greatest effect when acting at a distance 24 in above the base of the column.

The deflection at this point can be determined by considering the column to be a cantilever with a point load acting at its free end.
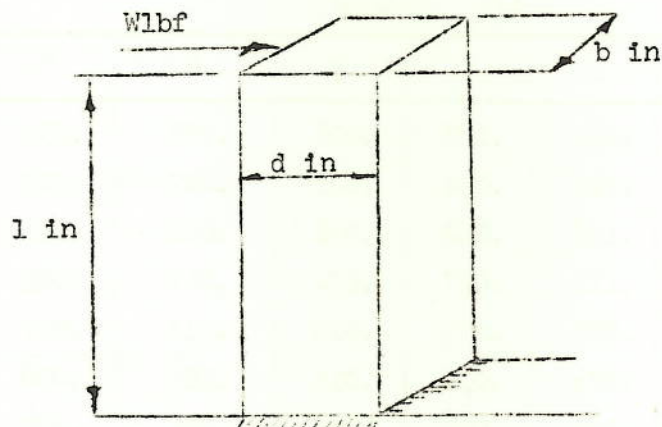
Taking into consideration deflection due to plane bending and shear, and assuming the column to be solid, the equation for deflection at the free end due to force W lb.f becomes :

$$= \frac{Wl^3}{3EI} + \frac{Wl}{Cbd}$$

where
W = 1000 lbf.
l = 24 in.
C = 12.10^6 lbf/in^2.
E = 30.10^6 lbf/in^2.
I = $\frac{bd^3}{12}$ in^4.

I VALUES (in^4)

b/d	1	2	3	4	5	6	7	8
1	.0833	.6664	2.25	5.33	10.40	17.99	28.57	42.65
2	.1666	1.3300	4.50	10.66	20.80	35.98	57.14	85.30
3	.2499	1.999	6.75	15.99	31.20	53.97	85.71	127.95
4	.3332	2.666	9.00	21.32	41.60	71.96	114.28	170.60
5	.4165	3.332	11.25	26.65	52.00	89.95	142.85	213.25
6	.4998	3.998	13.50	31.98	62.40	107.94	171.42	255.90
7	.5831	4.665	15.75	37.31	72.80	125.93	199.99	298.55
8	.6664	5.331	18.00	42.64	83.20	143.92	228.56	341.20

1/I VALUES

b/d	1	2	3	4	5	6	7	8
1	12.00	1.50	.444	.187	.096	.055	.034	.023
2	6.00	.752	.222	.094	.048	.027	.017	.012
3	4.00	.500	.148	.062	.032	.018	.011	.008
4	3.00	.375	.111	.047	.024	.014	.009	.006
5	2.40	.300	.089	.037	.019	.011	.007	.005
6	2.00	.250	.074	.031	.016	.009	.006	.004
7	1.70	.215	.064	.027	.014	.008	.005	.0033
8	1.50	.187	.055	.023	.012	.007	.0044	.003

DEFLECTION δ VALUES (in)

b/d	1	2	3	4	5	6	7	8
1	1.868	.2420	.0762	.0347	.0195	.0124	.0085	.0064
2	.934	.1215	.0381	.0174	.0097	.0061	.0043	.0033
3	.622	.0808	.0254	.0115	.0065	.0040	.0028	.0019
4	.467	.0606	.0190	.0087	.0048	.0031	.0025	.0017
5	.373	.0485	.0152	.0068	.0038	.0024	.0018	.0014
6	.311	.0404	.0127	.0057	.0032	.0020	.0015	.0011
7	.265	.0347	.0109	.0050	.0028	.0018	.0012	.0009
8	.234	.0302	.0094	.0042	.0024	.0015	.0011	.0008

Preliminary Hydrostatic Bearing Area Calculations

The bearings to support the main drill head structure are considered in this appendix. The total bearing thrust of the rear pads, i.e. those furthest away from the work tables, is found by consideration of moments to be 6000 lbf. If this thrust is assumed to be spread over three pads, each pad must support 2000 lbf. For air, referring to fig. A2.1

$$A_p = 20 - 1 + \frac{3}{4} = 19.75 \text{ in}^2$$

$$T_p = P_c.A_p = 200 \times 20 = 4000 \text{ lbf}$$

$$T_c = P_c \left[\frac{3(2)}{2 \text{ loge} \frac{1.5}{0.5}} - \frac{3}{4} \right]$$

$$= 200 \left[\frac{3}{1.1} - 0.75 \right]$$

$$= 400 \text{ lbf.}$$

$$T_s = 200 (10) = 2000 \cdot \text{lbf}$$

$$T_a = T_s + T_c$$

$$= 2400 \text{ lbf}$$

$$T = T_a + T_p = 6400 \text{ lbf}$$

$$A_c = \frac{6400}{200} = 32 \text{ in}^2 \text{ which is excessive for the available bearing}$$
surface area.

For oil : $W = 2000 \text{ lbf}$

$$A_c = \frac{2000}{200} = 10 \text{ in}^2$$

Hence a pad approximately 5 in x 2 in may be used. The final configuration for this pad is left until the final details of the machine bed and drill head column have been designated.

This equation reduces to

$$\delta = \frac{1}{I}\left[\frac{Wl^3}{3E} + \frac{Wld^2}{IRC}\right]$$

$$= \frac{1}{I}\left[.1538 + \frac{x^2}{6000}\right]$$

By varying the two column parameters d and b, values of δ and I were calculated, the maximum size of column for practical purposes was set as 8 in^2. A deflection of 0.001 in at a point 24 in above the base would give a maximum positional error of 0.000 5 in on a component drilled on this machine.

From the tables of δ and I values, a column 7 in x 8 in and having an I value of approximately 300 in would satisfy this condition. However, four columns, each of these dimensions, would have a combined weight of approximately 1400 lbf: a weight that is considered excessive by the Committee.

By combining two columns on each side of the drill head and working on an I value of 600 in^4 a hollow box section could be used, resulting in a lighter structure.

The following section, for example, has an I value of 700 in^4 and if
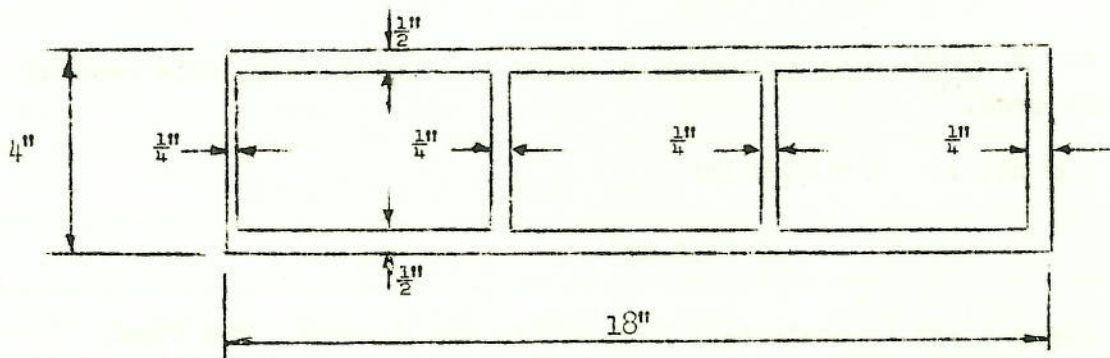
used on the machine would give a structure weighing 252 lbf, which is 18% of the weight of the four solid column structure first discussed. In conclusion the committee recommends that the support columns should be of welded steel box section type.