ROTOR HUB DESIGN FOR A COMMERCIAL RAMJET HELICOPTER

by

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ABSTRACT

The rotor hub design is considered for a small two-place ramjet helicopter. A design proposal is presented here which embodies a detail design of the rotor hub for a flapping-blade type rotor. The advantages and limitations of this proposal are discussed. Then, the rotor hub and its component parts are designed to meet the load requirements of a helicopter in a rolling pull-out flight condition.

Thesis Supervisor: Professor Otto C. Koppen

Title: Professor of Aeronautical Engineering

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CHAPTER I

THE DESIGN PROBLEM

1.10 General Description of Helicopter

The Hiller Hornet made by Hiller Helicopters, a two-place ram-jet helicopter, is chosen as a typical helicopter for the two-place ram-jet helicopter class. The general specifications and performance data for the Hornet are used. (See Table I.)

The ram-jet engine used in this design proposal is different from the one used in the Hornet. The Hiller ram-jet engine was developed in 1946-47 and is considered out-of-date today. An engine of the required size was taken from the research and development work by F. Martins in a thesis for a B.S. degree in 1954 in the Mechanical Engineering Department.

1.20 Design Requirements

Designing a rotor hub entails three basic problems: first, accommodating an internal means of transferring fuel from the hub to the blade; second, integrating a flapping and/or feathering blade connection from the hub to blade link; third, designing the hub to withstand the large centrifugal forces imposed when one considers a ram-jet type operation. All of which makes the problem interesting, to say the least.

TABLE I

RAM-JET HELICOPTER

SPECIFICATIONS AND PERFORMANCE DATA

General Dimensions	
Rotor blade diameter	24'
Height	7'
Fuselage length	13'11"
Power Plant	
Number	2
Type	ram-jet
Weight	10#/engine
Thrust	45#/engine
Weight	
Empty	390#
Gross	980#
Useful load	530#
Performance	
Normal cruising speed	70 mph
Top speed	80 mph
Ceiling (full gross load)	12,000'
Rate of climb	1100 fpm

1.21 Structural Integrity

The rotor hub and its component parts must be able to receive the limit loads specified without permanent deformation and must be able to withstand the most critical combination of ultimate loads without failure. It must satisfy these requirements without exorbitant weight penalties.

1.22 Accessibility

The rotor hub and its accessories must be available for maintenance and repair purposes.

CHAPTER II

PRELIMINARY DESIGN

2.00 Initially, an analysis will be made in the preliminary design to calculate and design the various rotor blade characteristics in order to establish the loads requirements and some semblance of arrangement. The performance and specifications data from Table I will be used as a basis for the preliminary design.

2.10

The disc area = $Ad = \frac{\pi (D)^2}{4} = \frac{\pi (24)^2}{4} = 453$ sq.ft. $= S_e$

$$\sigma = \frac{bc}{\pi R} = \text{rotor solidity ratio}$$

In general, the most favorable operating conditions will be obtained when the distribution of lift is constant over the rotor blade. It is obvious that to obtain this one must have twist and tapered blades. However, for small helicopters it is economically feasible to use constant chord-constant blade angle type rotor blades. This will be the case for the preliminary design consideration.

Most present-day designs have σ in the range from 0.04 to 0.065. A solidity factor, $\sigma = 0.052$ will be used.

$$\sigma = .052 = \frac{2c}{\pi \cdot 12} = \cdot \frac{c}{\pi 6}$$

$$c = .98' = 11.75''$$

Ram-jets are most efficient at high mach numbers. However, one wants to avoid the drag rise and high centrifugal forces encountered at high tip speeds. In such a case as this, one must make a compromise. The drag rise for a NACA 23012 occurs near .68 mach number. At sea level conditions, this is 750 feet f.p.s. The blade will be designed to rotate with a tip speed of 600 ft/sec which is even a little high for tangential velocities.

$$V_T = 600 = R\Omega$$

$$\Omega = \frac{600}{12} = 50 \text{ rps} = 3000 \text{ rpm} = \text{radians/sec}$$

$$RPM = \frac{3000}{2\pi} = 478 \text{ RPM}$$

Disc loading

$$\frac{T}{\pi R^2} = \frac{980}{453} = 2.16\#/\text{sq.ft.}$$ which is fairly good according to current designs (2.2 - 3.5)

$$\text{Power loading} = \frac{T}{P} = \frac{980}{68} = 14.3\#/\text{HP}$$ which also agrees quite closely with current types (10 - 14#/HP)

$$\text{Figure of Merit} = M = \frac{T}{P}\sqrt{\frac{T}{\pi R^2}}$$

$$= PL\sqrt{\frac{D.L}{\rho}}$$

$$= 14.3\sqrt{\frac{2.16}{.002378}} = 431$$

2.20 Using the results found in the preceding work and verifying the results with PL, TR, and M with current designs, a preliminary design of a rotor blade will be conducted. See Figure 1.

$$CF_m = \text{centrifugal force moment} = 2/3\left[R\ M\Omega^2 R\beta\right]$$

where MR = blade mass

$$CF_m = 2/3 \cdot 12\ \frac{33.25}{32.2} \cdot 12^2 \times 50^2\ \beta$$

$$= 12 \times 20.65 \times 10^3 \beta'^{\#}$$

$$\text{Lift moment} = 3/4\ R \times \text{lift (for no twist, no taper)}$$

$$= 2/3 \cdot R \cdot \frac{980}{2} = 4 \times 980$$

$$= 4 \times 980 = 3{,}920\#$$

$$= \text{coning angle} = \frac{\frac{9}{8}\ \text{blade lift}}{CF}$$

$$= \frac{\frac{9}{8} \times \frac{980}{2}}{2{,}540 \times 12} = \frac{.216}{12} \times (57.3) = 1.333^\circ$$

2.30 Rotor Blade Preliminary Design

The rotor blade is considered and analyzed in order that its weight may be obtained for the load conditions. The rotor blade and ram-jet engine are considered as point masses located at 1/2 R and R respectively.

Using the chord found in section 2.10 and the diameter from Table I, a rotor blade is designed using an NACA 23012 airfoil as a basic plan form. The blade is fabricated; that is, it will consist of a main steel spar, wooden ribs, and

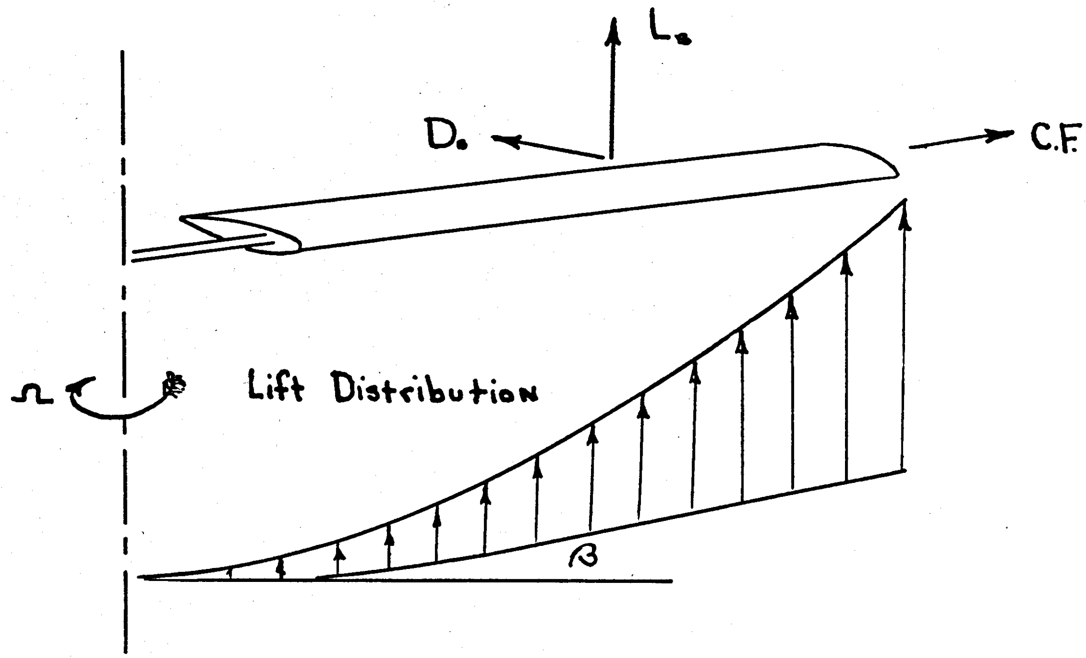

Figure 1: ROTOR BLADE

skin. Birch and birch plywood are used for this purpose. See Figure 2.

$R = 12'$

$c = 11.75'' = .98'$

NACA 23012 Airfoil

Birch wood and ply = 45#/ cu.ft.

Area of cross section = 10.85 sq. in. which was determined graphically.

Volume of a blade = 10.85 x 144 = 1550 cu. in.

The spar is a 1-1/4 OD 4130 steel with a tapered section thickness from .180 - .06. The weight$_{av.}$ of such a span = 1.93#/'.

See Figure 4. Built up blade with the spar at the quarter chord. Half the effective area of the wood cross section is in the leading edge to bring the c.g. to the quarter chord.

Area in leading edge = 3.1 sq. in.

Area behind 1/4 c = 3.1

wt. of wood - $\frac{1550}{144} \times \frac{45}{12} \times \frac{6.2}{10.85} = 23.4^{\#}$

wt. of pipe - $12' \times 1.93 = 23.2^{\#}$

Total weight of blade = $46.6^{\#}$

2.40 Hinge System

The advancing blade of the rotor encounters higher velocities than the retreating blade as the rotor moves forward. Considering first a rigid rotor, it is seen that a sizable rolling moment would be present in forward flight

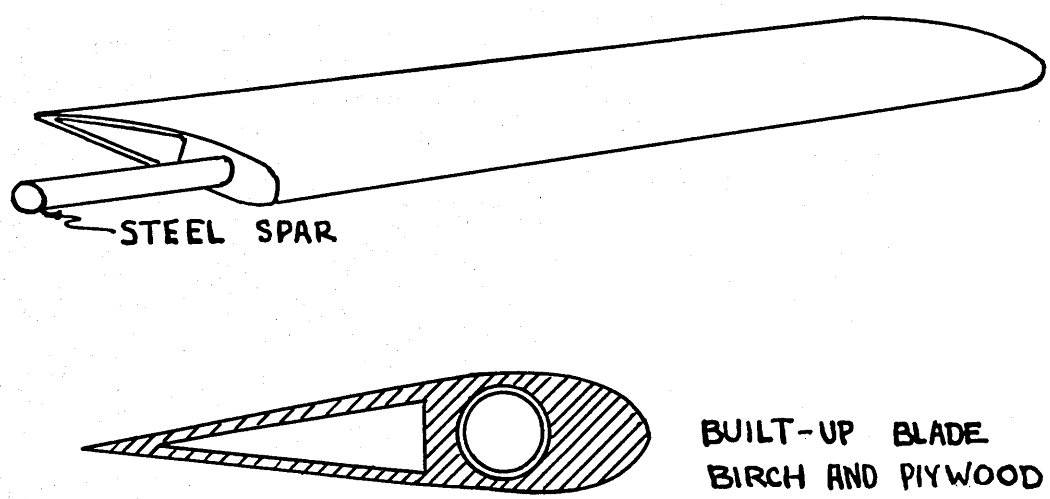

Figure 2: FABRICATED ROTOR BLADE

as a result of the difference in the lift produced on the advancing and retreating blade.

Two standard means are used to overcome the dissymmetry of lift in forward flight. They are:

(1) The blades may be hinged at their roots so that no moments can be transmitted through the hub. Control is achieved by tilting the hub axis until the vector points in the desired direction.

(2) The blades may be rigidly attached to the shaft but cyclically feathered, decreasing the pitch on the advancing side and increasing the pitch on the retreating side so as to equalize the lift around the disk.

The hinged blade method is selected because of its relative ease of construction and simplicity of design as compared to the feathering blade method.

2.41 Flapping Blades

In such an arrangement as Figure 3, high bending stresses which are built up in the blade root are relieved. The blades are free to rotate in any direction about the point of attachment, being held in equilibrium at any given instant by the action of the weight, inertia and aerodynamic forces.

2.42 Forces on a Blade Element

See Figures 4(a) and 4(b).

$$\text{The centrifugal force} = m\Omega^2 r \cos\beta \, dr$$

$$\text{weight} = w\,dr$$

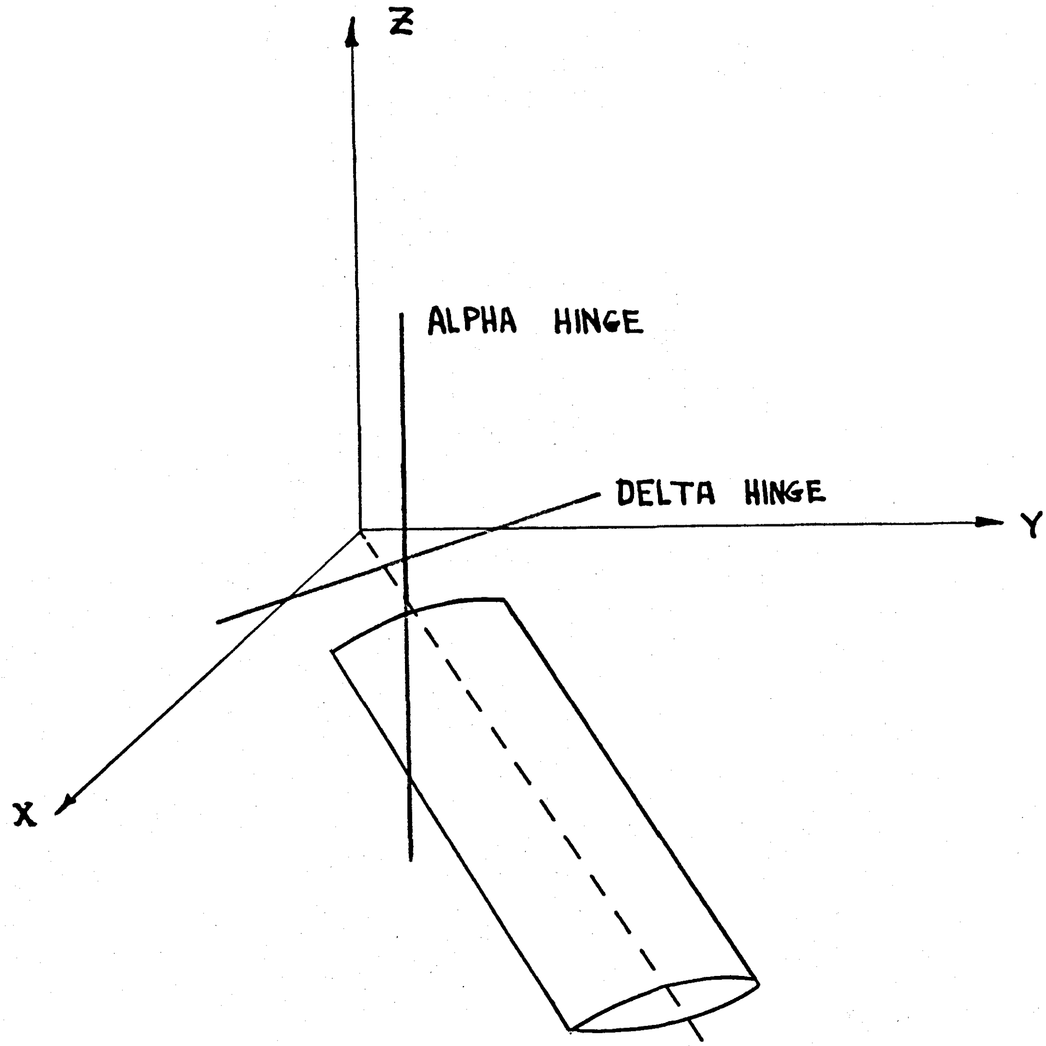

Figure 3: FLAPPING BLADE

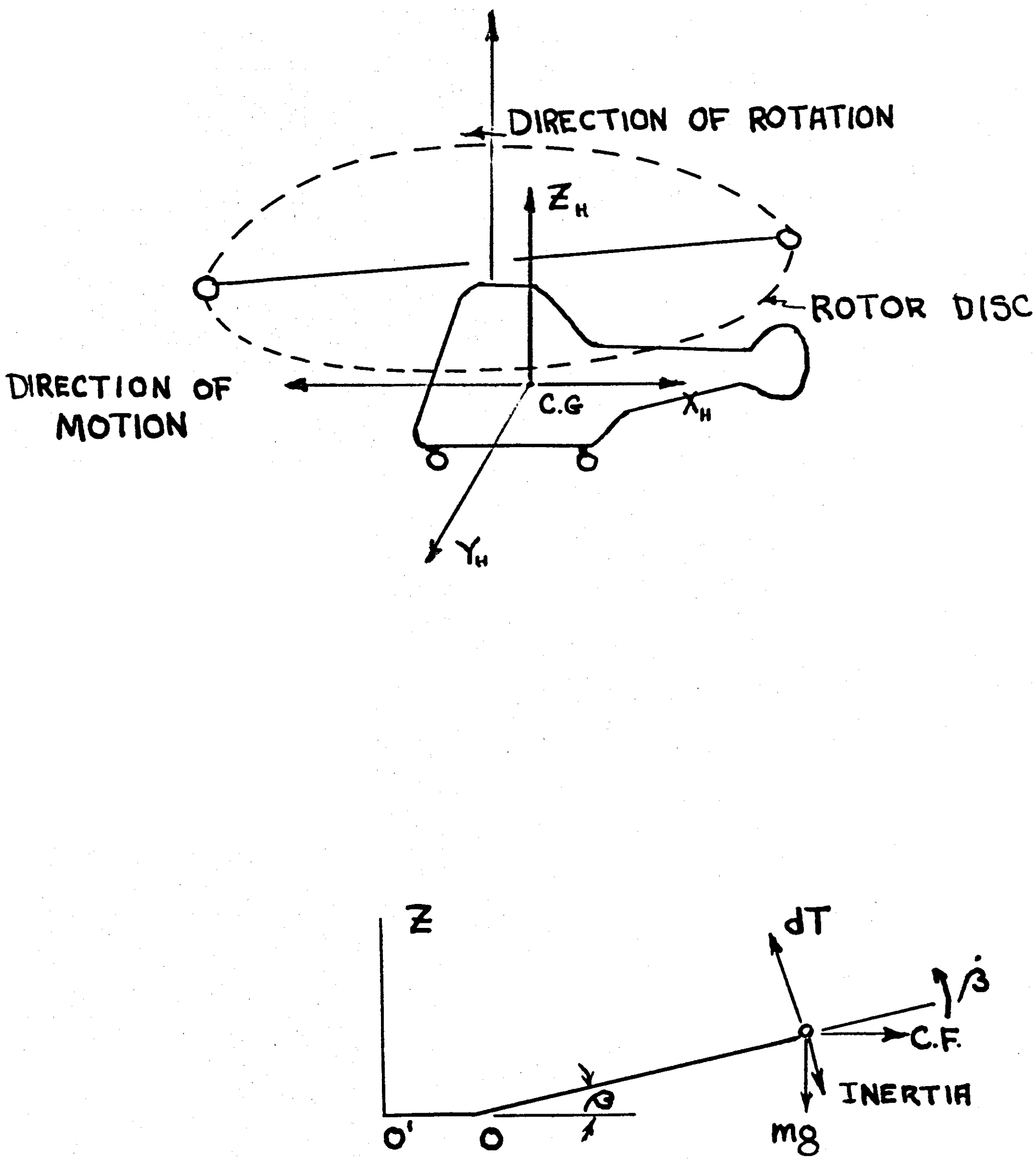

Figure 4: FORCES ON BLADE ELEMENT

inertia $= rm\ddot{\beta}\, dr$

air load $= \frac{1}{2}\rho \cdot C_{\ell} \cdot c_e \cdot U_T 2\; dr$

By attaching the blades to the rotor hub through a nearly horizontal delta (flapping hinge), the rolling and bending moments are reduced. It affects the rotor and rotor blades as follows:

(a) Bending moment acting on the blades in the direction of flapping hinge is eliminated at the root.

(b) Owing to the short length of the supporting arms extending from the rotor to hub to flapping hinges, the rolling moment becomes negligible.

(c) The blades, owing to cyclic variation of the velocity component begin to rise and fall cyclically, that is, to flap.

The sum of the moments must equal zero about the flapping hinge.

$$M_{fl.hg.} = 0 = \Omega^2 \sin\beta \cos\beta \int_0^R mr^2 dr + \beta \int_0^R mr^2 dr \qquad (1)$$

$$- M_T - M_W$$

where M_T = moment due to air loads

M_W = moment due to weight

β = coning angle

β usually is a small angle and the small angle approximation can be made; thus

$$\sin\beta = \beta$$

$$\cos\beta = 1.00$$

but $\int_0^R mr^2 dr$ = moment of inertia of blades

equation (1) may be written

$$(\ddot{\beta} - \beta\Omega^2)\, I - M_T + M_W = 0$$

2.50 Rotor Control in Forward Flight

Control of the helicopter in any flight condition involves the proper orientation of the rotor thrust vector and therefore the tip-path plane in space.

With the ram-jet helicopter where there is no power transmission difficulties through the shaft, a teetering axis type of control in which the whole shaft with the hub rigidly connected is teetered with respect to the fuselage. This type of system is simple to control and is easier to construct, operate and maintain than either the tilting hub or the cyclically varying blade types.

CHAPTER III

DESIGN SOLUTION

3.00 The critical maneuver which determines the design conditions and loads is a rolling pull-out with a load factor of 5. This was determined through discussion with Professor Raymond L. Bisplinghoff of the Aeronautical Engineering Department, considering current design conditions which determine the design loads.

3.10 Load Calculations

First, the rotor blade will be analyzed to see what loads it imposes on the hub. See Figure 5. For the blade, a point mass is concentrated at $\frac{R}{2} = 6'$ to calculate the centrifugal force of the blade.

The rotor is located at R = 12'.

$$C.F. = mR\Omega^2$$

for blade:

$$C.F. = \frac{46.6}{32.2} \times 6 \times 50^2 = 20{,}600\#$$

for ram-jet engine

$$C.F. = \frac{10}{32.2} \times 12 \times 50^2 = 9{,}300\#$$

total C.F. force = 30,000#

A lift force of 490# is also applied to the rotor plus a drag force. However, the lift and drag forces are very

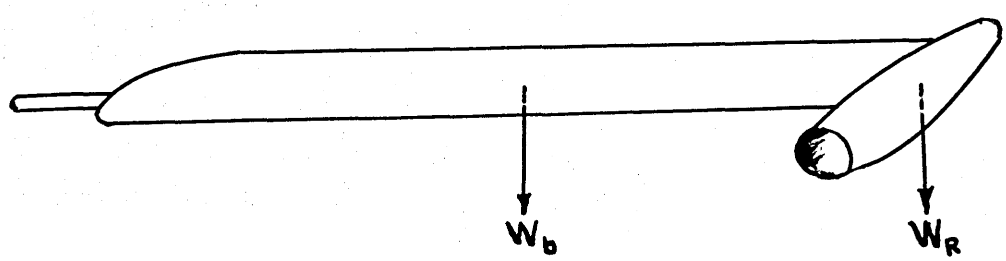

Figure 5: LOAD DISTRIBUTION

small compared to the centrifugal force; therefore they are neglected in the hub design.

3.20 Detail Design of Main Spar in Rotor Blade

As mentioned previously, the blade is constructed on a fabrication design. A steel spar is used to carry the loads with a wooden structure built about it to insure the proper aerodynamic characteristics. The leading edge is solid birch with plywood being used from the quarter chord back to the trailing edge. See Figure 2. The reason a solid leading edge is used is to move the center of gravity of the cross section to be approximately at the quarter chord.

Calculations:

See Figure 6.

The outside diameter of the spar is 1.25".

$P = 30{,}000^{\#} = $ C.F./blade

One wants a Margin of Safety of .15 or better for such a connection. The tensile strength of 4130 steel, which will be used, is given as F_{tu} which is the ultimate tensile strength. However, one is interested in designing for the yield tensile strength which is equal to approximately $\frac{F_{tu}}{1.5}$.

Therefore, calculating the thickness of the wall pipe at the root which would be sufficient to carry the centrifugal loading,

F_{tu} for 4130 steel - 90 Kips heat treated

$A = \frac{P}{f}$ which was derived from $f = \frac{P}{A}$

$$A = \frac{30{,}000 \times 1.5 \times 1.15}{90{,}000} = .575 \text{ sq}''$$

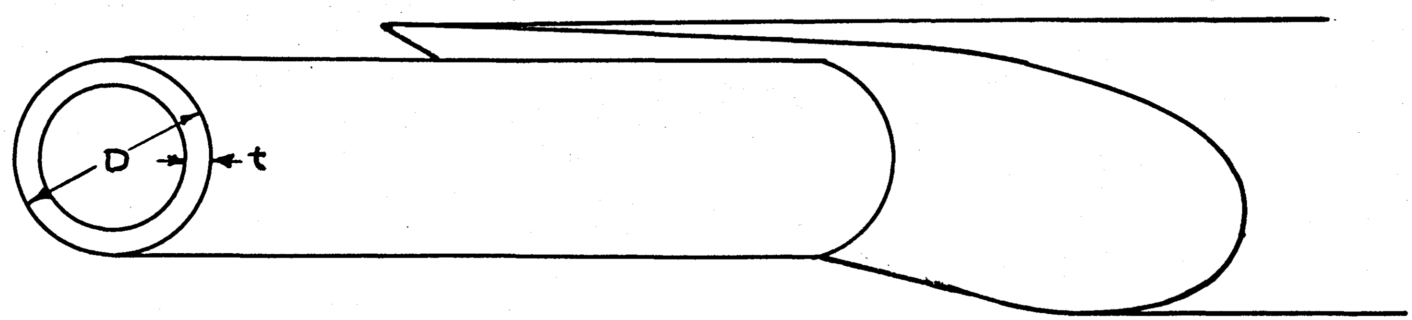

Figure 6: , MAIN SPAR

but $A = \frac{\pi}{4} (D_o^2 - D_i^2)$

where D_o = outer diameter

D_i = inner diameter

$$.575 = \frac{\pi}{4} (1.25^2 - D_i^2)$$

solving $D_i = .91''$

$$t = \text{thickness} = \frac{1.25 - .91}{2} = .17''$$

3.21 Detail Design of Flapping-hinge Shaft Connection

See Figure 7.

P for ultimate = P x 1.5 = 30,000 x 1.5 = 45,000

M.S. = .15

$$f = \frac{P}{A} = \frac{45 \times 1.15}{A}$$

4130 heat-treated steel is also used for this connection.

F_{tu} = 90 Kips

$$A = \frac{45 \times 1.15}{90} = .575 \text{ sq}''$$

The outside diameter of the shaft = 1.5"

$$A = .575 \text{ sq}'' = \frac{\pi}{4} (1.5^2 - D_i^2)$$

solving $D_i = 1.235''$

$$t = \frac{D_o - D_i}{2} = \frac{1.5 - 1.235}{2} = .13''$$

3.22 Bolt Size

P_u = 45,000#

One bolt per hinge will be used. These bolts will be in double shear. Using ANC-5, table 2.611(a) for standard

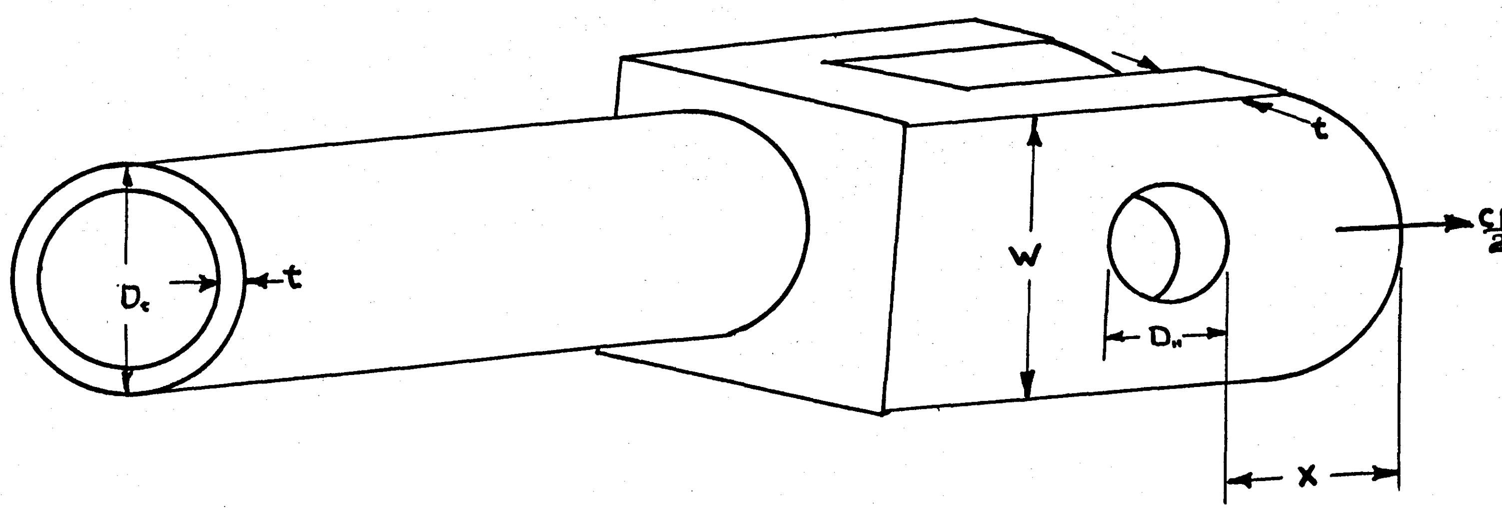

Figure 7: FLAPPING HINGE SHAFT CONNECTION

bolts heat treated to 125 Kips, AN-12 is tried.

$$F_s = 33{,}150 \times 2 = 66{,}300 \text{ for double shear}$$

$$MS = \frac{66{,}300}{45{,}000} - 1 = .47$$

An AN-12, which is a bolt 3/4" in diameter, will be used.

3.23 Wall Thickness

A factor of safety of 2 is used to take care of shock, vibration and rotation in this connection.

Using 4130 AN-QQ-5-685 type,

MIL-T-6731

$F_{bru} = 140$ Kips

(a) Designing for bearing

$$f = n \cdot MS \cdot t \cdot d \cdot F_{bru}$$

$$45{,}000 = 1 \times 2 \times 2 \times 3/4 \times 140{,}000 \times t$$

solving for t

$$t = .324"$$

(b) Designing for tear-out

$$f_s = \frac{P}{2tx}$$

$x = 1"$

$$P = \frac{45{,}000}{2} = 22{,}500\#$$

$t = .324$

$$f_{su} = \frac{22{,}500}{2 \times .324 \times 1} = 34{,}000$$

$f_{su} = 65$ Kips from ANC-5

$$M.S. = \frac{65}{34} - 1 = \underline{\underline{.91}}$$

(c) Designing for tension failure

See Figure 7(b).

$$f_t = \frac{P}{(w-d)t}$$

$$= \frac{22,500}{(2 - 3/4) \times .324} = 55,500 \text{ psi}$$

f_{tu} = 100 Kips from ANC-5

$$\text{M.S.} = \frac{100}{55.5} - 1 = .80$$

3.30 Method of Control

A teetering axis system is selected for the method of control. The rotor shaft is tilted with respect to the fuselage.

A set of two gymbols, perpendicular to each other in the Z plane, is used to allow the rotor shaft to tilt in the X and Y directions. Limits are incorporated in the design to restrict the range of teeter to 7° to the left or to the rear and to a limit of 15° tilt forward.

The bearings which make up the gymbols have friction locks and adjustors which can be tightened to give a specified degree of friction in the control system which enables the flight condition to be maintained and a "hands-off" flight control condition realized.

3.31 Control System Gymbols

The lift is equal to the weight times the load factor for the rolling pull-out condition. Each set of gymbols carries this load.

$$\text{Load} = 1000 \times 5 = 5000\#$$

$$\text{Load ultimate} = 5000 \times 1.5 = 7{,}500\#$$

$$\text{Load/gymbol} = \frac{7{,}500}{2} = 3{,}750\#$$

See Figure 8.

The gymbol shaft is in single shear. Using table 2.6111(a) ANC-5 for AN-5, 5/16" bolt,

$$f_s = 7360 \text{ for heat treated to 125 Kips}$$

$$\text{M.S.} = \frac{7{,}360}{3{,}750} - 1 = .965$$

AN-4 was examined and found to have M.S. < .15. Therefore, 5/16"-diameter gymbol bearings will be used for both the lateral and longitudinal gymbols.

3.32 Thickness of Gymbol Mounts

(a) For bearing see Figure 8. 4130 steel is used for the gymbol mounts.

$$F_{bru} = 140 \text{ Kips}$$

$$f_u = n \cdot \text{M.S.} \cdot t \cdot d \cdot F_{bru}$$

$$f_u = 3{,}750 = 1 \times 1.5 \times t \cdot \frac{5}{16} \times 140{,}000$$

solving for t

$$t = .0573''$$

Actually, in the design a much thicker gymbol mount is employed to support the bearings for the gymbol.

(b) Designing for tear-out - Longitudinal Gymbol Direction Control

$$D = 5/16''$$

$$Pu = 3750$$

$$t = 1''$$

$$y = \frac{11}{32}$$

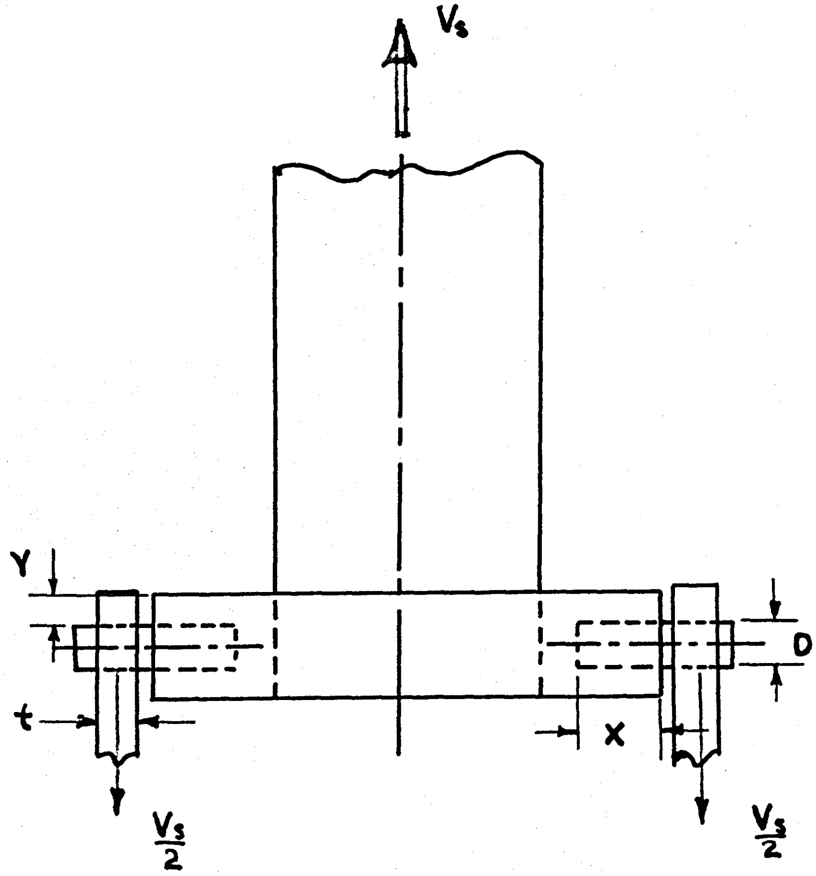

Figure 8: CONTROL SYSTEM GYMBOLS

See Figure 8.

$$f_{su} = \frac{Pu}{D \cdot 2 \cdot x} = \frac{3750}{2 \times 1 \times 1/2} = 3750 \text{ psi}$$

f_s = 35 Kips for 4130

$$M.S. = \frac{35}{3.75} - 1 \ggg 1 \quad \therefore \text{ not critical}$$

(c) Tension failure

$$f_{tu} = \frac{P}{(w-D)t} = \frac{3,750}{(5.5 - .312)} = 720 \text{ psi}$$

$$M.S. = \frac{90}{.720} - 1 \ggg 1$$

1(b) Designing for tear-out - Lateral Gymbol Direction Control

See Figure 8.

$x = 3/4''$

$t = .125''$

$$f_{su} = \frac{Pu}{20x} = \frac{3750}{.75 \times 2 \times .125} = 20,000 \text{ psi}$$

f_{su} = 55,000

$$M.S. = \frac{55}{20} - 1 = 1.75$$

1(c) Tension failure

$$f_t = \frac{P}{(w-D)t}$$

$$= \frac{3,750}{(1.5 - .312)} = 27,500 \text{ psi}$$

f_t = 95,000

$$M.S. = \frac{95,000}{27,500} - 1 = 2.45$$

3.33 Second Stage Gymbol Mount Connection

See Figure 9.

(a) Number of bolts required:

from AN-C5 table 2.6111(a)

two AN-4 1/4" dia. bolts for each connection

$f_s = 3{,}680$

$$M.S. = \frac{3{,}680}{\frac{3{,}750}{2}} - 1 = .965$$

From the previous work, it is obvious that tension failure investigation and tear-out failure investigation are not necessary because these conditions are not critical.

(b) For bearing

$$f_{bu} = \frac{\frac{3{,}750}{2}}{1.15x1.5x.125x.25} = 34{,}800 \text{ psi}$$

$$f_{bu} = 140$$

$$M.S. = \frac{140}{34} - 1 \qquad 1 = 3.04$$

3.34 Gymbol Mount to Shaft Connection

See Figure 8.

The shaft and the gymbol plate are threaded with a standard American 8-pitch thread series -8N ASA B1.1-1949. Size 2-1/2"; basic diameter 2.5000"; lead angle at basic pitch diameter λ, 57'; minor diameter internal threads = 2.3647; stress area = 4.4352 sq".

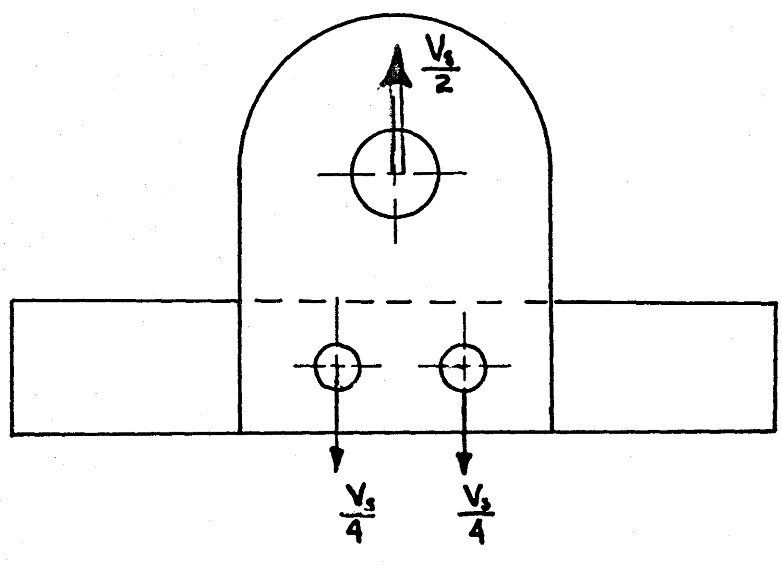

Figure 9

SECOND STAGE GYMBOL MOUNT CONNECTION

3.35 Shaft Wall Thickness

The shaft is made of 4130 steel.

f_{tu} = 90 Kips

$$f_t = \frac{Pu}{A_{eff.}} \quad \text{where } A_{eff.} = \frac{\pi}{4}(D_o^2 - D_i^2)$$

$$A_{eff.} = \frac{Pu}{f_t} = \frac{7,500}{90,000} = .0834$$

$$A_{eff.} = \frac{\pi}{4}(2.5^2 - D_i^2)$$

$$D_i = \sqrt{6.25 - \frac{.20}{\pi}} = 6.25 - .106$$

$$= \sqrt{6.14} = 2.48''$$

$$t = .01''$$

The shaft wall thickness is not critical in tension. The depth of the threads plus a marginal effective wall thickness is the factor which determines the shaft wall thickness.

3.40 Bearings

There are extremely high loads in tension on the rotor spar, hinge and hub which consitute a bearing problem when one realizes that a collective pitch control requires a hinge which must be rotated or pitched.

3.41 Bearings for Hub System

Ball or roller bearings support a loaded moving part on a set of hardened steel rolling elements. These rolling elements usually roll between two smooth hardened steel rings, although in some cases the rolling elements may roll directly on the shaft or housing bore. A cage or a separator is used to keep the rolling elements properly spaced about the bearing pitch circumference.

Standard rolling bearings are manufactured as interchangeable units and can be quickly obtained and inserted as a unit if replacement is necessary. Ball and roller bearings have low friction characteristics of rolling motion, this friction being only slightly greater when starting than when under uniform speed of rotation. Anti-friction bearings, especially roller bearings, are relatively insensitive to moderate overloads or shock loads.

Although in rolling bearings sliding motion is essentially replaced by rolling motion, some sliding does occur; therefore lubrication is necessary.

3.42 Hub-shaft Bearings for Cyclic Control Shaft

First, angular contact ball bearings will be analyzed for application. See Figure 10.

This type of bearing is designed to carry thrust loads or combined loads which are predominantly thrust. They can carry radial loads if two are mounted opposed.

The collective pitch control shaft is .75" in diameter.

The external dimensions and radial capacities of angular contact bearings (SKF Industries Inc., Philadelphia, Pa.) are, using the medium series for a 20 mm bore:

$$d = .787'' \qquad D = 2.047'' \qquad B = .5906''$$

If a radial bearing is loaded by a thrust load under standard conditions, X is a rotation factor which depends on the bearing type and on which ring rotates relative to the

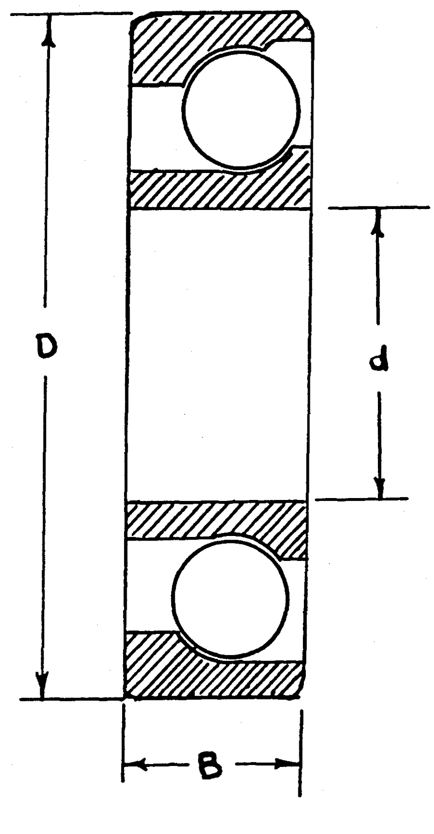

Figure 10: ANGULAR ROLLER BEARINGS

load direction; Y is a thrust factor which depends on the bearing type and on the load. F_R is a radial component of the load. F_a is the thrust component of the load.

For this bearing:

$Y = .48$

Dynamic reference capacity $= 2{,}700\# = C$

$X = 1$

$P_a = XF_a + YF_R$

However, $F_R = 0$

$$P_a = F_a$$

The life in number of rotations may be expressed as a function of the life in hours and speed in rpm.

$$C = \frac{f_h P}{f_n}$$

where C = dynamic reference capacity

P = equivalent load

f_n = speed factor

f_h = life factor

For this application, the arm rotates with the hub at 486 rpm.

At this rpm, f_n can be read off of table 12-69 in Kent's Handbook.

$$f_n = .41$$

but $$f_h = \frac{f_n \times C}{P}$$

$$= \frac{.41 \times 2700}{100}$$

where a maximum of 100# control force is used.

$$f_h = 11.1$$

At this f_h, the rated life is found on scale 2 in section 12-69 Kent's Handbook to be greater than 90,000 hours which is more than adequate.

Two such type and size angular ball bearings are used opposed to each other. Oil bath lubrication will be used which is more convenient than grease lubrication. The bearing may be pressed on the shaft by the use of an arbor press, tapped on the shaft by a mounting tube and hammer, shrunk on the shaft after heating or forced on with mounting nuts.

3.43 Hub-shaft Bearing for Main Load

The tapered roller bearing will be used in this application. This type of bearing is constructed upon the principle of rolling cones, the elements of rollers and raceways converging to a common intersection on the axis of the bearing. True rolling motion is obtained along the cone elements and the bearing will sustain radial or thrust loads or any combination of them.

This is the type of loading that exists between the hub and shaft. One has a large thrust loading from lift and small radial loads due to the small changes in the centrifugal forces which may occur due to accelerations and decelerations of the blades in the plane of rotation. These loads are small but should be considered.

A steep angle, tapered roller bearing is used because the thrust load dominates, although there are appreciable radial loads. Combined with the steep-angle tapered roller bearings, a double-row bearing will be used to provide maximum load carrying capacity in a small space; resist thrust loads from both directions and maintain a precise and rigid mounting. See Figure 11.

For double-row steep-angle tapered roller bearings

$$A = 2.5''$$

$$B = 5.513$$

$$C = 1.4375''$$

$$\text{Radial load} = 9{,}880\#$$

$$\text{Thrust load} = \pm 8{,}690\#$$

$$M.S. = \frac{8{,}690}{7{,}500} - 1 = .16$$

A double-row steep-angle tapered roller bearing, Type TSS, made by Timken Roller Bearing Company, Canton, Ohio, with shaft lock nuts and washers (ASA coarse-thread series, medium fit, class 3). In general, bearings less than 6" outside diameter, operating at speeds less than 1000 rpm without unusual conditions, may be lubricated with a lime or soda base grease of medium consistency. The bearings can be purged with grease when needed.

3.44 Thrust Bearings

At low rotational speeds at which the thrust extension bar will rotate, shaft shoulders or collars which bear against flat bearing rings immersed in a semi-fluid may be used. For hardened steel collars on bronze rings with interrupted service, pressures up to 2,000 psi are permissible. In multi-collar thrust bearings, the pressures are reduced

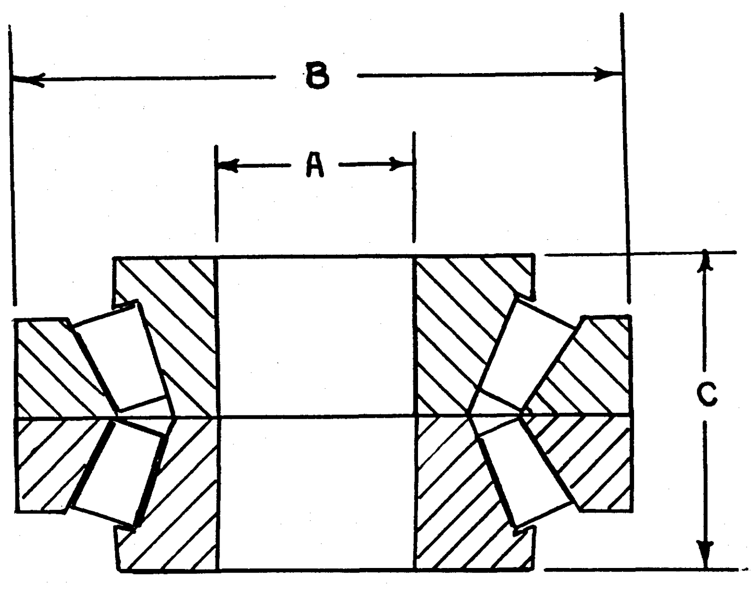

Figure 11: STEEP ANGLE TAPERED BEARINGS

because of the difficulty in distributing the load evenly between several collars.

The fixed-wedge film type of thrust bearing has also been investigated. It is obvious that part of the thrust bearing face must be flat to permit the collar to start and stop under load without undue concentration of pressure. The wedge-shape film in a bearing of this kind is unable to lift its load at the low speeds that one experiences in such a control system. The fixed wedge may be combined with flat surfaces by using alternate bevels and flats with a supply groove at the end of the bevel, in the direction of rotation.

Various methods of carrying thrust loads were analyzed and applied to this problem. Radial ball bearings, angular ball bearings, tapered thrust bearings, needle bearings, spherical bearings and collar bearings were used in the analysis with the result that very large bearings of various types or a great number of bearings in series were required.

The collar bearing described in section 3.44 offered the best result; that is, a small, compact, simple unit.

Calculations:

The thrust collars are formed by grooving the shaft; the outside diameter $D = 1.6 \rightarrow 1.9d$ where d = normal shaft diameter. The thickness of each collar $= w = .13 \rightarrow .16d$ and the distance between collars $= s = 2w \rightarrow 3w$.

number of collars required $= n$

$$= \frac{P_t}{\pi d\ b \cdot p}$$

where P_t = total thrust on bearing

d_i = mean diameter of collar

b = radial width of collar

p = allowable bearing pressure

$d = 2.5''$

$D = 2.0d = 5.0''$

$P = 45{,}000\#$

$d_i = 1.5d = 3.75''$

$b = .5 \times 2.5 = 1.25''$

$p = 2{,}000$

$$n = \frac{45{,}000}{\pi\, 3.75 \times 1.25 \times 2{,}000} = \frac{45}{29.4} = 1.54$$

Two collars will be used.

$$M.S. = \frac{F_t}{f_t} - 1$$

The force which the two collars can withstand

$$F_t = 2 \times 3.75 \times 1.25 \times 2000$$

$$= 2 \times 29.4 \times 10^3 = 58{,}800\#$$

$$M.S. = \frac{58{,}800}{45{,}000} - 1 = \underline{\underline{.31}}$$

3.50 Hub Design

The size of the rotor hub is determined by the size of the multi-collar thrust bearing. The hub will be either cast and machined, forged and machined or machined from a solid piece. If the part is either cast or forged, a larger margin of safety is required.

The rotor hub must also be large enough to accommodate the shaft thrust bearing and fuel cell. Extreme care must be taken in design and construction of the hub. All sharp corners must be replaced with fillets. The hub must be inspected for scratches and cracks by a magna-flux process. Sharp corners, scratches and cracks are stress raisers and will lead to failure under the large centrifugal loading.

The hub is made in two half shells which are bolted together by eight AN-8 bolts heat treated to 125 Kips. The hub itself is heat treated and case hardened to relieve residual stresses obtained through construction and to insure a resistance to scratches from damage.

3.51 Hub Design

See Figure 12.

$$\text{Area} = 6t + (6 - 3/4 - 2t)t + 4(1 - 1/2t)t$$
$$= 15.25t - 4t^2$$

where t = thickness

The area must take the shear flow due to the tensile force of 45,000#.

$$F_{tu} = \frac{45{,}000 \times 1.5}{A}$$

where a M.S. of .5 is used for a forging or casting. Hub is made from 4130 steel.

$$F_{tu} = 90{,}000$$
$$A = .75 \text{ sq}'' = 15.25t - 4t^2$$
$$t = .05$$

which shows the hub thickness is not critical.

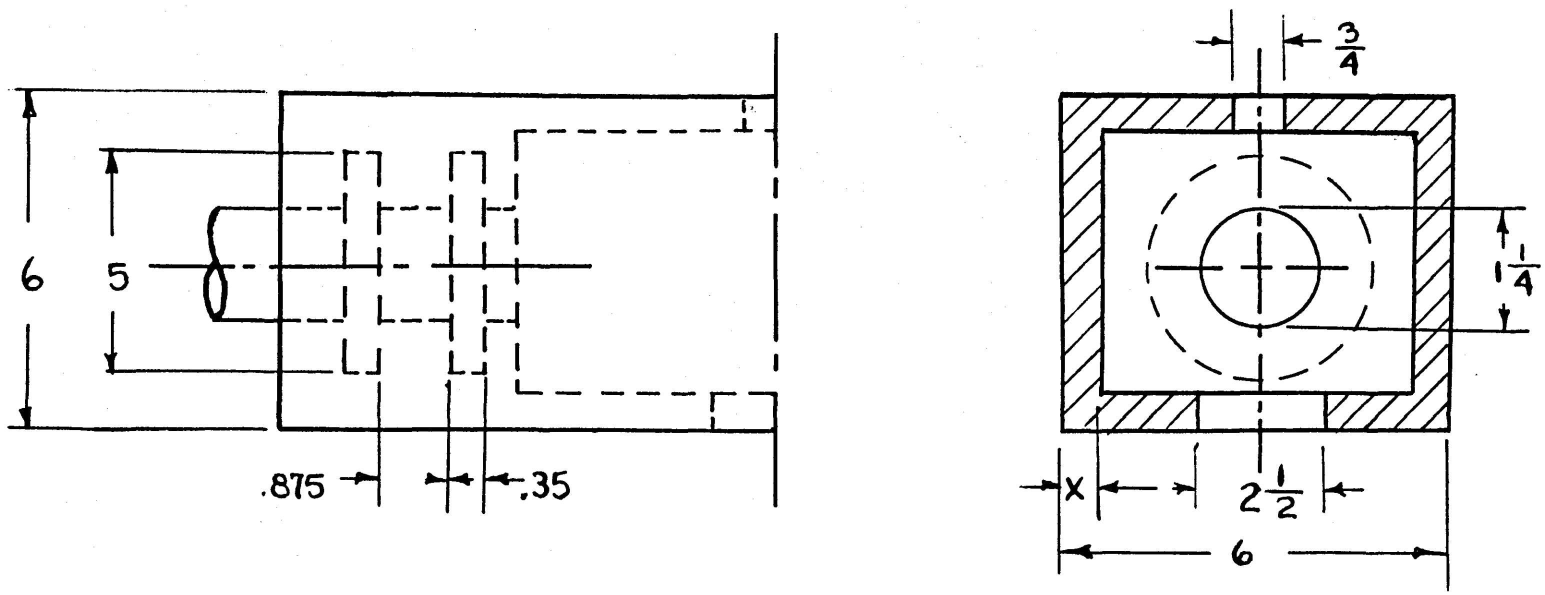

Figure 12: HUB, CROSS-SECTION

3.52 Number of Bolts Required

AN-8 8/16" bolts

$f_s = 14{,}700$ heat treated to 125 Kips

$$n = \frac{45{,}000 \times 1.15}{14{,}700} = 3.52 \text{ bolts}$$

Four bolts will be used on each side.

3.60 Fuel Cell

One great problem encountered is the method of transferring the fuel from the helicopter to the engine which is located at the end of the rotor.

A fuel cell method is selected. It will be inclosed in the rotor hub and will rotate with the hub about the stationary main shaft. The fuel is pumped through the shaft and exhausted into the fuel cell reservoir. The return path is presented by dynamic seals enclosed in the shaft; between hub and collective pitch control rod and at the cell-shaft connection. (See Design Drawing #1.)

The cell is divided into two parts. Each part exhausts its fuel to its respective engine. The fuel is pumped out of the cell down the hollow thrust bearing, through the hollow rotor spar to the engine by the action of the large centrifugal forces.

3.61 Top Seal (Shaft Seal)

See Design Drawing #1. There are three basic types of shaft seals: namely, leaf spring, garter spring and self-sealing. The sealing is compounded to suit the average operating conditions. The primary criterion is surface speed under the sealing lip. At rpm = 486 - 500 surface speed is not a deterring factor and a leaf-spring-type shaft seal will be used.

In order that dynamic seals may have a maximum life, every effort must be made to have and to maintain smooth surfaces over which packing slides. The ideal surface is one machined to a finish of 10 to 15 microinches rms. This can be done by fine grinding or honing. If the surface is smoother than 10 to 15 microinches, lubrication of the packing becomes difficult because the surface is too smooth to hold the oil.

Leather, a universal packing material, is used. It is moulded and impregnated with wax. It has a very long life as compared to rubber. Rubber (synthetic) packings tend to cold bond; therefore, leather packing is better for this application.

3.62 A flange packing is used for the seal between the inner part of the outer shaft and the collective pitch control rod. (See Design Drawing #1.) A leaf spring is used to insure a pressure on the packing lip against the collective pitch control rod.

Size of packing - standard packing

Outside diameter	1-1/2"
Inside diameter	3/4"
Height of packing	1/2"
Thickness of leather	1/8"

A standard retaining ring, 1/16" thick, 2.75" outside diameter, is used.

3.63 Fuel Cell - Shaft Connection

A combination friction and labyrinth packing is used for the seal between the main shaft and the fuel cell. The centrifugal force due to the fuel in the cell will exert a force on the cell and will cause a tight seal and lock in the connection. (See Design Drawing #1.) A garter spring is also used on the bottom section to insure a pressure against the seal during static conditions.

3.70 Collective Pitch Control

The primary flight controls embodied in the rotor consist of cyclic and collective rotor-pitch controls. As mentioned previously, the cyclic pitch control is achieved by direct tilting of the rotor shaft about the gymbol axes. The rotor shaft does not rotate. Collective pitch is achieved by actuating a push-pull shaft inside the rotor shaft which supports the collective pitch cross arms. This action raises and lowers the collective pitch cross arms and changes the pitch of both blades through the series of control rods, pitch links and pitch arms. This system is a basic type of helicopter control and has the advantages of being simple, yet eliminating possible interaction between cyclic and collective pitch changes without the complex compensating linkages employed in a "swash-plate" type of control.

The range of control for the collective pitch system is from -2° angle of attack to $+12^{\circ}$. The -2° limit is used for autorotation purposes.

3.80 The ignition system consists of a small 6-8 volt rechargeable storage battery, switches, slip rings, ignition coils and miniature spark plugs. Ignition is supplied to the engines only while the particular engine is being started.

DETAIL DESIGN DRAWING #1

	Component Parts	Specifications
1	Flapping-rotor blade hinge	DD5
2	Collective pitch lever arm	DD3
3	Two roller bearings for collective pitch control link	3.42
4	Collective pitch control link	DD3
5	Collective pitch control push-pull shaft	(3/4" OD x .049 wall 1015 steel)
6	Leaf spring-type shaft seal	3.62
7	Fuel-cell retainer ring	(standard 1/8"x3" OD)
8	Fuel line	---
9	Fuel cell	DD6
10	Journal Rotor blade thrust bearing	DD5
11	Rotor hub	DD7
12	Retainer ring for dynamic seal	3.61
13	Dynamic Seal (leather packing)	3.61
14	Standard 3/4" bushing for Collective pitch control sod	
15	TSS Timken Standard Tapered-Roller Bearings	3.43
16	Retainer ring for Tapered-Roller Bearings	3.43
17	Standard 3/4" bushing for collective pitch control rod	
18	Second stage gymbol fixture	DD4
19	Control system braces (not designed for; not critical)	---
20	Control system plate (not designed for; not critical)	---

	Component Parts	Specifications
21	Control system stick	(1" OD x .049 wall 4130)
22	Collective pitch linkage bar	DD3
23	Collective pitch linkage bar and lever arm	DD3
24	Shaft washer	3.43
25	Shaft lock-nut	3.43
26	Shaft	DD2
27	First stage gymbol fixture	DD4
28	Collective pitch control arm	DD3

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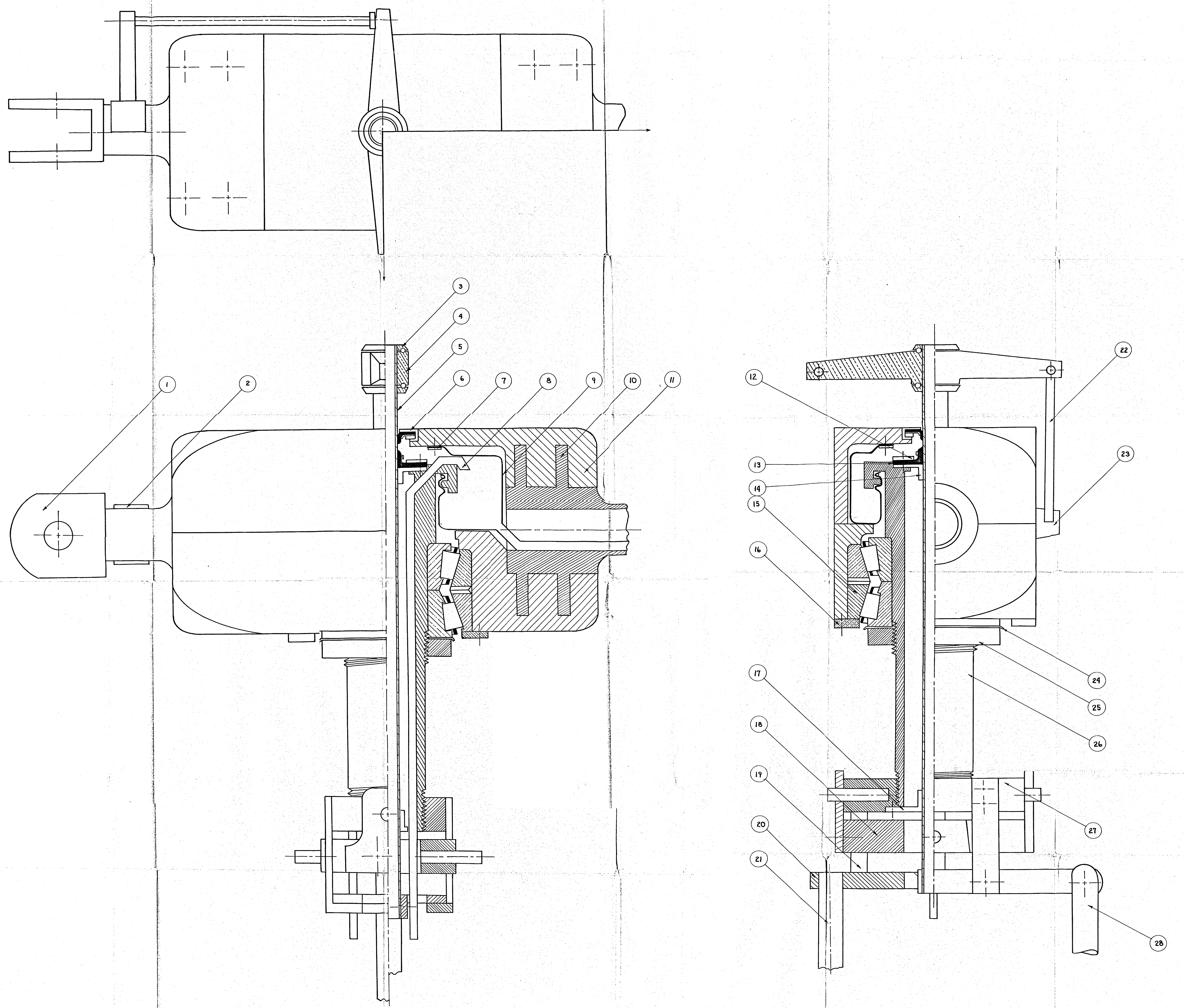

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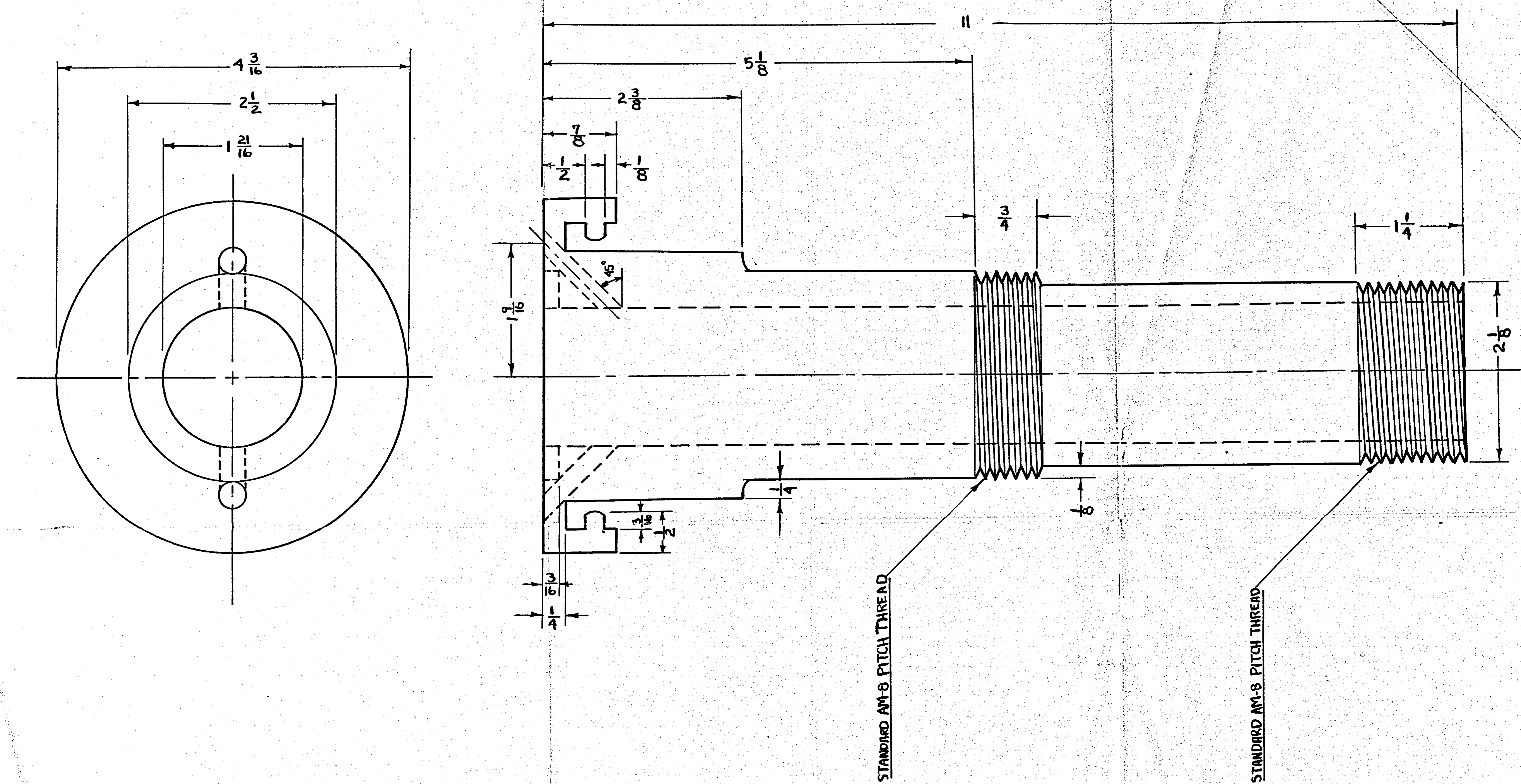
DRAWN BY WM. F MOODY	HELICOPTER HUB ASSEMBLY DR.	DD 1



FULL SCALE	ROTOR SHAFT	MAY 24, 1954 B.S. THESIS
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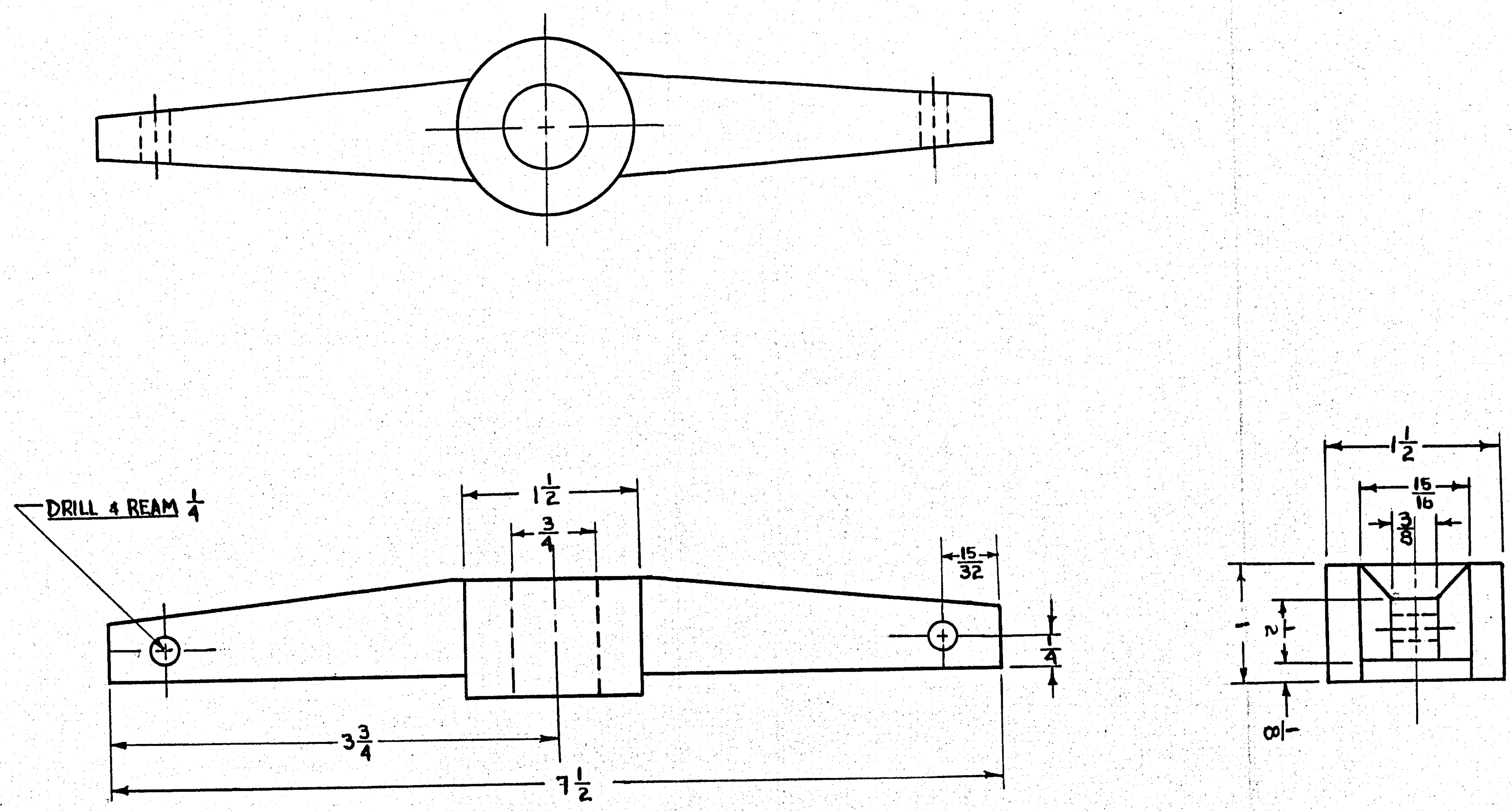

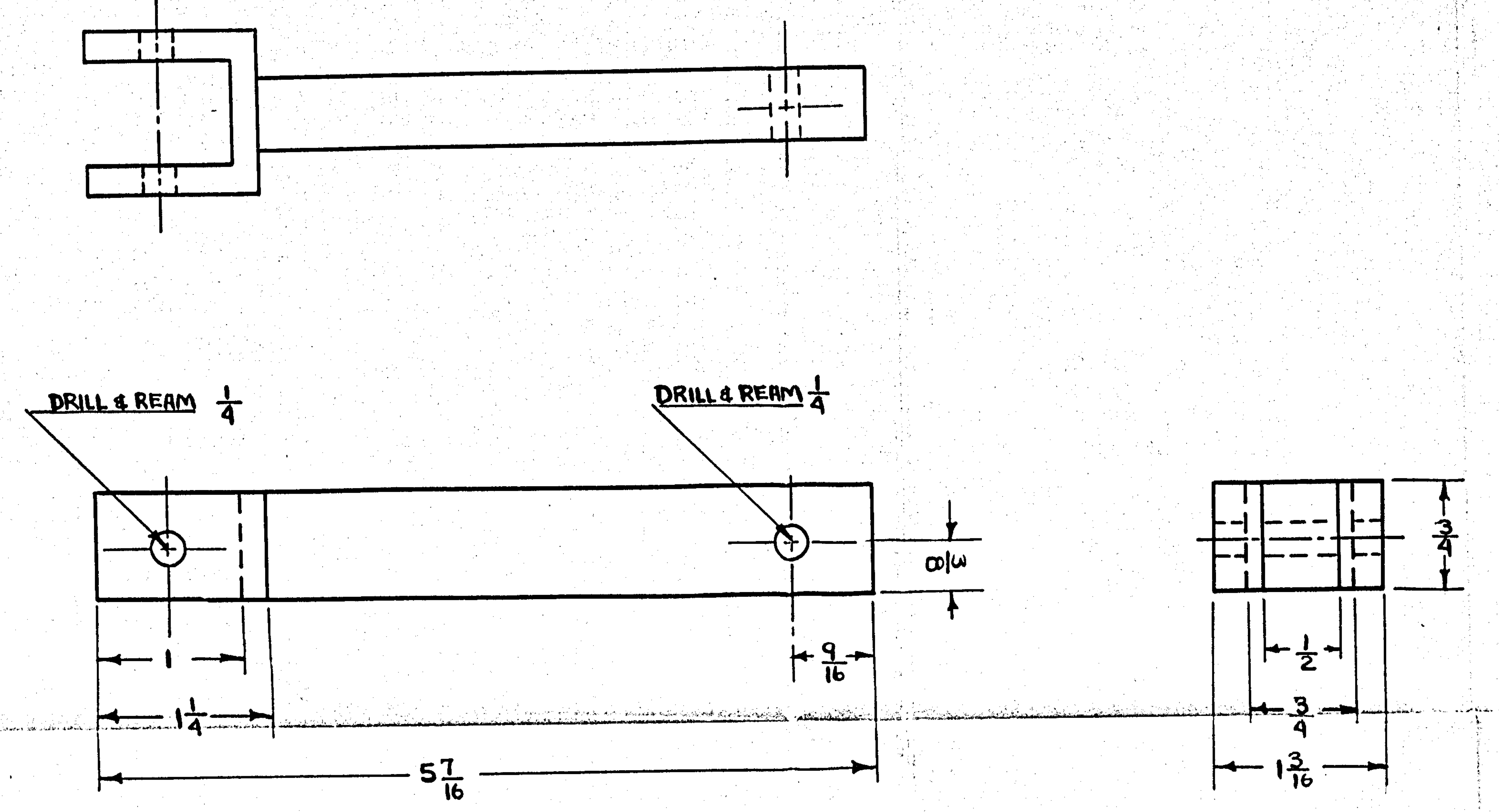

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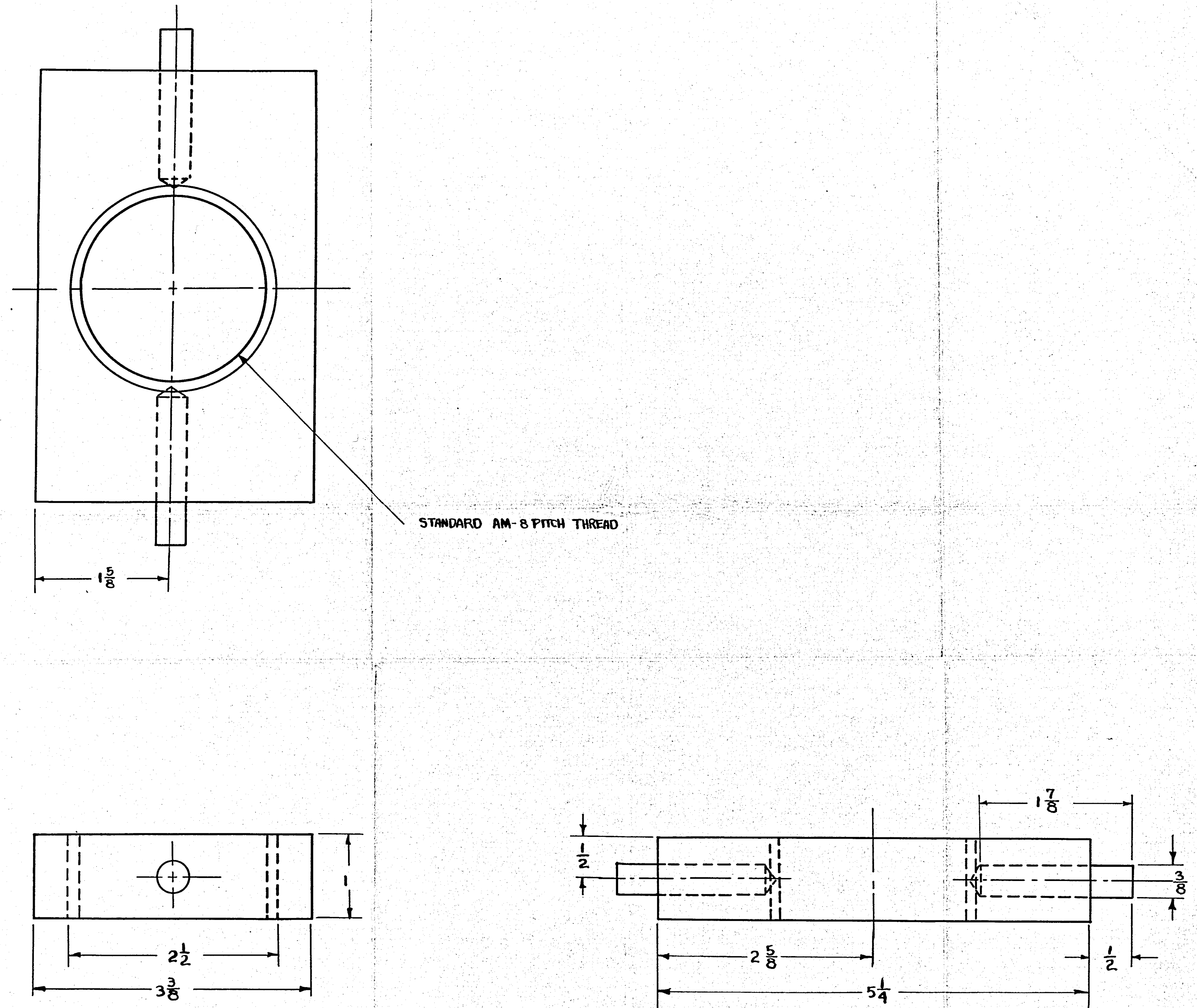

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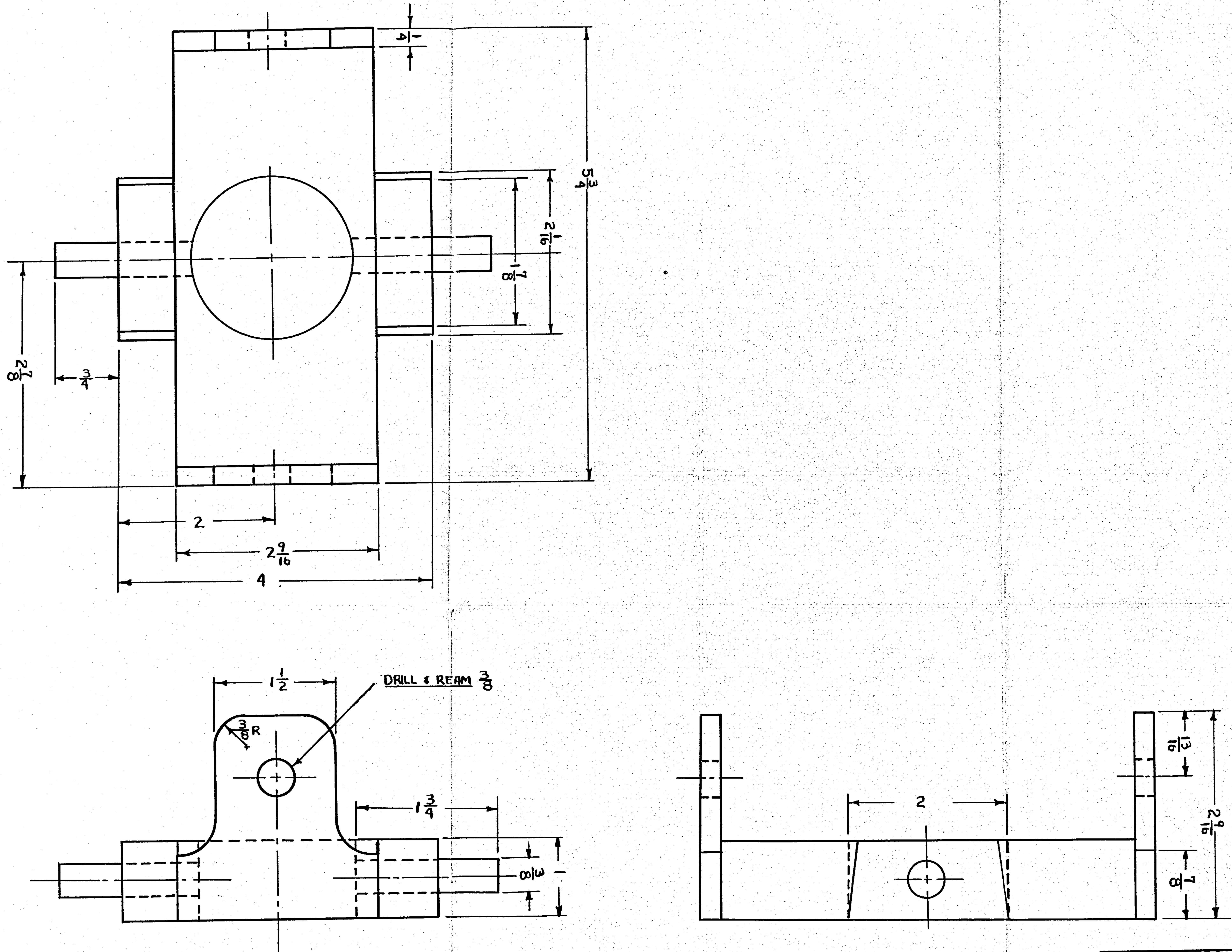

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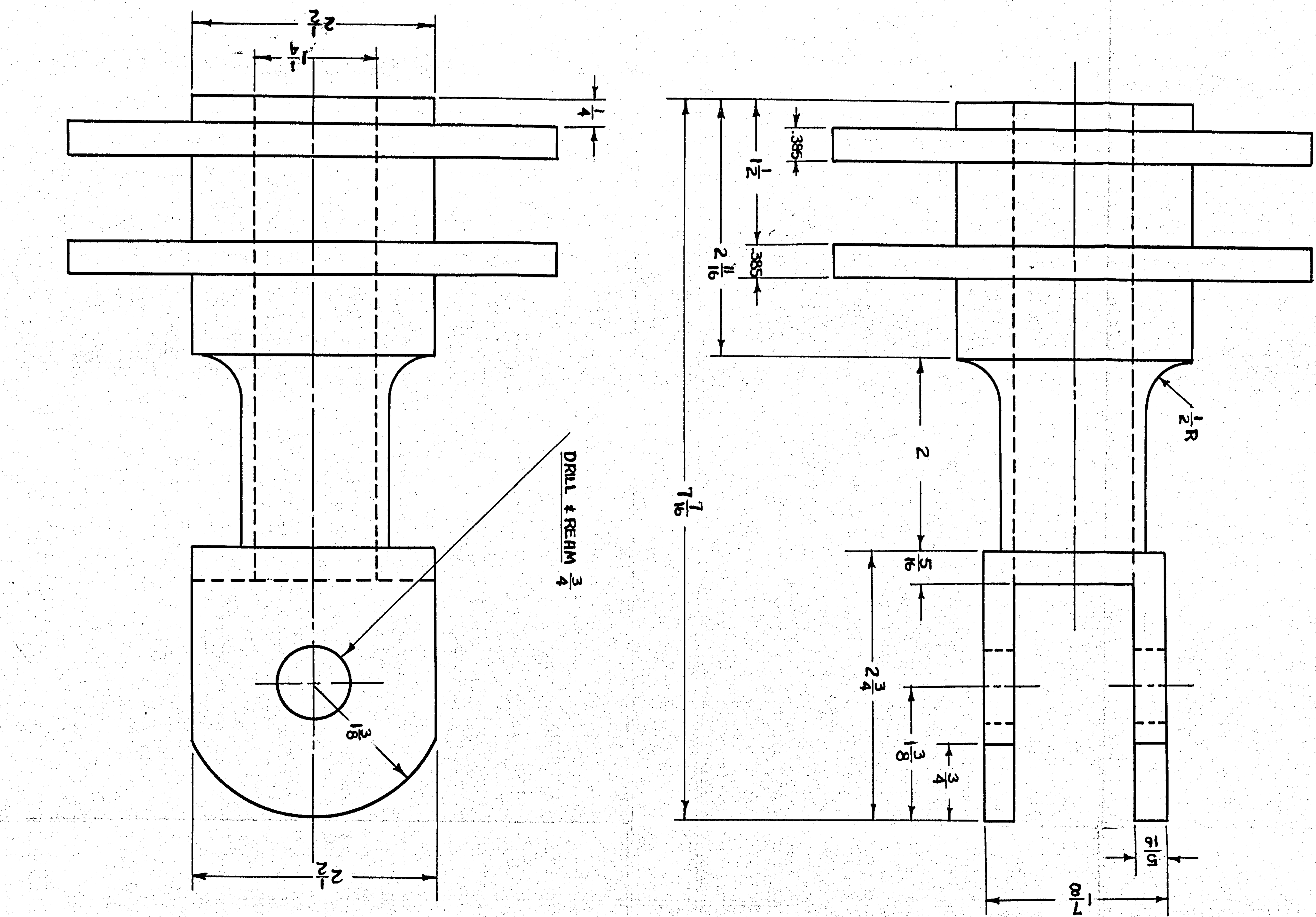

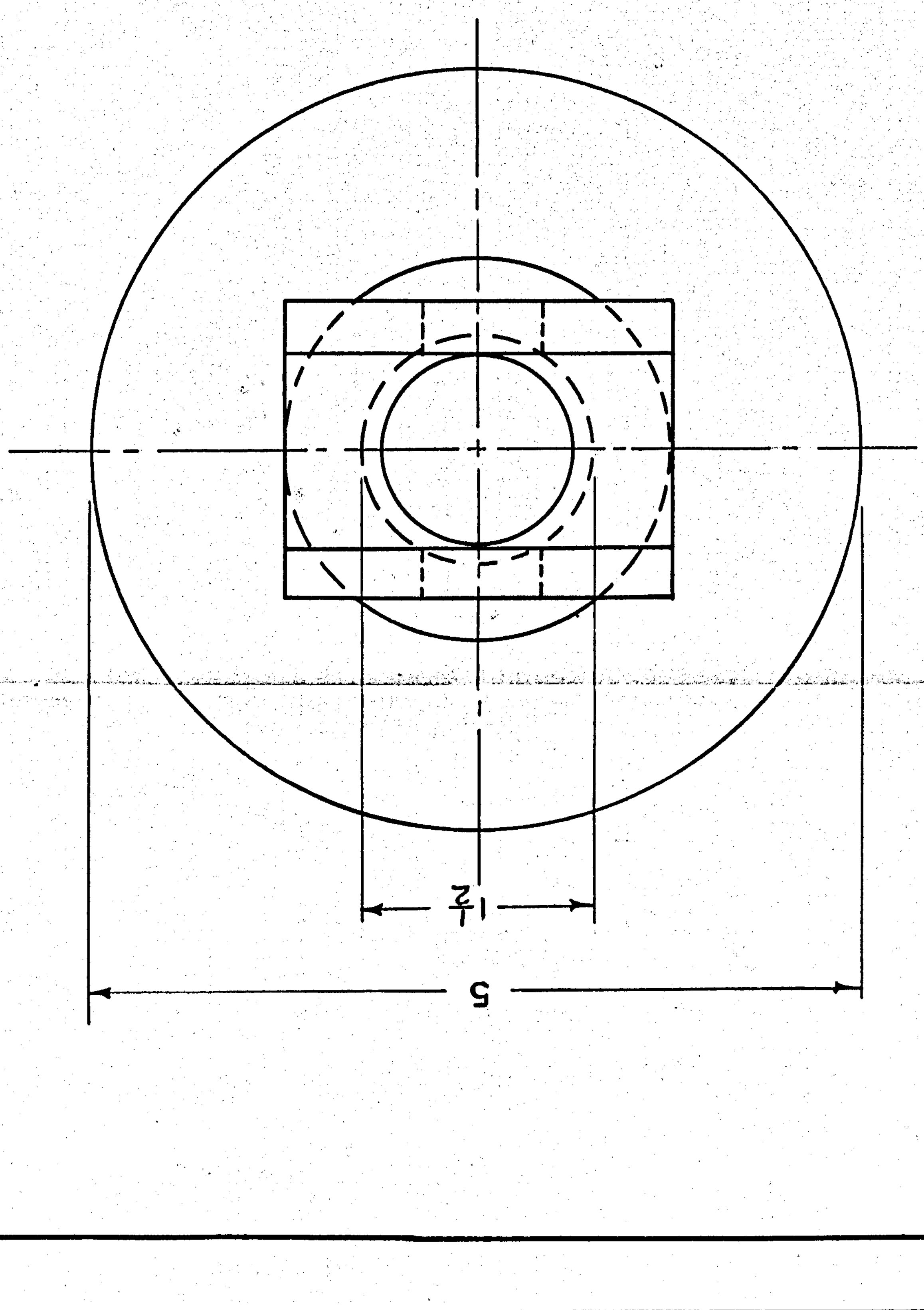

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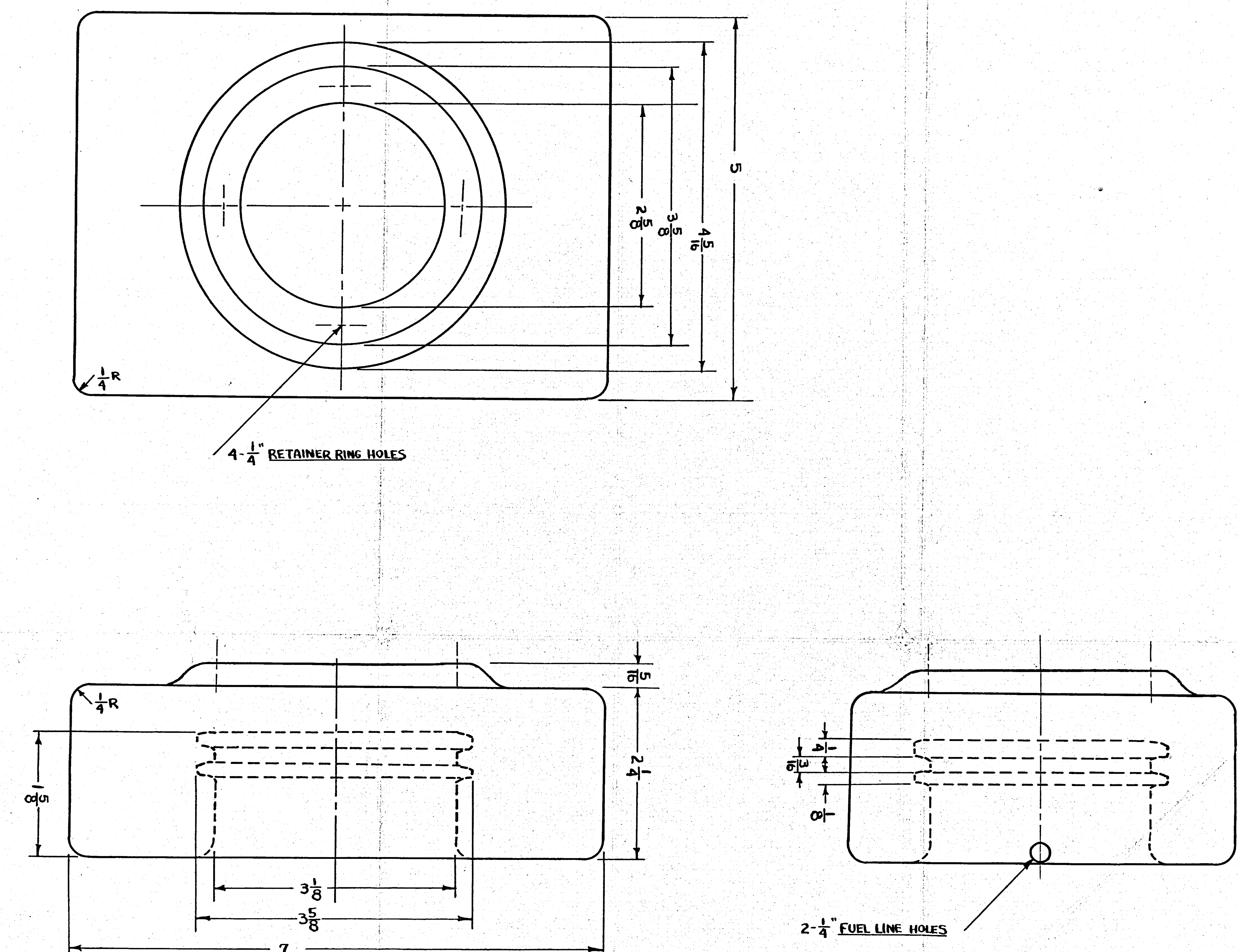

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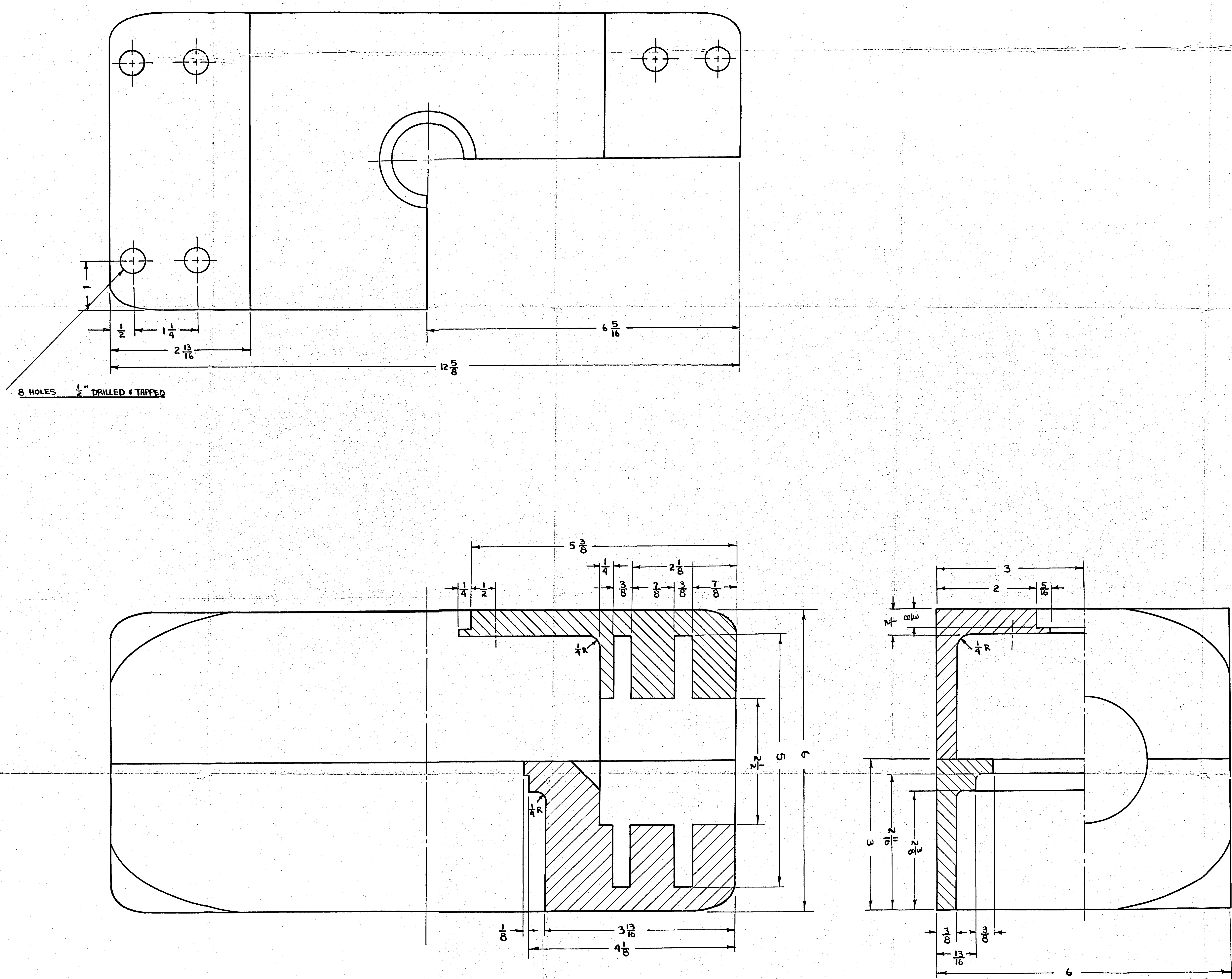

FULL SCALE	ROTOR HUB	MAY 24, 1954 B.S. THESIS
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