

"DESIGN AND CONSTRUCTION  
OF  
"A TWENTY-FIVE HORSE-POWER GASOLINE TRACTION-ENGINE"  
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Introduction.

Object: To design and construct at the shops of the Kansas State Agricultural College, a gasoline traction engine that shall fulfill the following requirements;

The engine shall have but one set of gearing to be used for forward and reverse motion, dispensing with speed changing gearing, together with its expensive first cost and maintenance, noise, low efficiency and weight. The gearing shall be on stiff shafts and rigid centers, having no radius rods nor stub gearing. All gears shall be of crucible cast steel, and shall be of ample size to stand rough usage and wear. Master or bull gears shall have roller bearings. Differential gears shall be of the spur gear type and shall be placed on a counter shaft. The bull gears shall be attached to the drive wheels by means of ample spring cushion links to take off sudden strains on the gearing and frame.

The crank shaft shall be given as steady a torque as possible by using four quadruple opposed cylinders. These cylinders shall be large enough to give the crank shaft as great a torque as is given to the crank shaft of steam engine of the same nominal rating.

The governor shall be constructed so that it can be set and regulated to run the engine under complete control at any speed from 50 R. P. M. to 220 R. P. M., as desired.

The cylinders shall be horizontal and shall be located as low as possible in order that they may be covered with cooling water when water in the tank is low.

The engine shall be easily reversed and shall be fitted with two friction clutches, one for the traction gearing



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and one for the belt pulley.

All levers controlling the engine and the steering wheel shall be of easy access and easily operated. The engine shall be made self starting from rest by means of compressed air. The combustion chamber of one cylinder shall be provided with a small outlet pipe and check valve to supply the compression tank. This tank shall be of sufficient capacity to do whistling, lift plows, start engine, and fill gasoline tank and cooling water tanks.

There shall be a winding drum located on the intermediate gear cross shaft for the purpose of pulling a load over soft ground where the engine cannot get sufficient footing. In using this drum the engine is to run forward, unrolling the cable at the same time. The load is to be pulled up to the engine by winding the cable and hooked on again if on good ground.

The frame work shall be constructed of structural steel and shall be designed to give ample rigidity and strength. Couplers shall be placed at each end of the engine. The coupler at the front shall consist of a ring and pin of sufficient size. The coupler at rear shall be of sufficient strength to draw the load and shall also be arranged so that the load can be disconnected while pulling, thus rendering it unnecessary to give slack before uncoupling can be accomplished.

The engine proper and the platform shall be covered with a large top to protect the working parts of the engine from the elements and also to protect the operator from the sun.



As a basis from which to work in designing and constructing this engine I procured the gearing and mounting wheels of a 25 H. P. Huber steam traction engine. Since an ordinary 25 H. P. nominal rating steam traction engine will develop in a brake test from 50 to 70 H. P., this gearing can be assumed to be capable of transmitting 50 to 70 H. P. also. I procured four cylinders which the Buckeye Engine & Foundry Company use with their 10 and 12 H. P. single cylinder gasoline engines. These cylinders when used with gasoline engines running at 250 R. P. M. will develop from 14 to 16 B. H. P. each. Four of these cylinders will give from 56 to 64 B. H. P. which shows that they are well adapted to the gearing of a 25 H. P. nominal rating steam traction engine.

Cylinders: My reason for selecting a four cylinder engine is that a four cylinder, four cycle engine gives the same number of power impulses per revolution as a single cylinder steam engine, and the four cylinder quadruple opposed type engine is almost perfectly balanced and needs no counterweights.

The shape of the combustion chamber of these cylinders is not the ideal spherical, but approaches that form as nearly as possible without having a difficult pattern to cast and machine. The bore of the cylinder is 8-1/2", stroke 11", 200 cu. in. being allowed for the compression space. Using the adiabatic equation,

$$p_1 v_1^n = p_2 v_2^n$$

where p = lbs. pressure per sq. in.  
 v = volume in cu. in.  
 and n = 1.33

for a mixture of gasoline vapor and air, and solving for p<sub>2</sub> gives the compression as 97 lbs. per sq. in. absolute. This corresponds



to 82.3 gage pressure which is a little higher than the general practice, but in considering the fact that the engine is governed by the amount of the charge and that a light load will be carried most of the time, the high compression will have no serious effect. By having a large body of cooling water, a good head and free circulation the cylinders can be kept cool and not liable to back fire.

In the head or combustion chamber the cylinder walls are 3/4" thick and vary from 3/4" to 1/2" in the wall surrounding the piston. Assuming the explosion pressure to be approximately 300 lbs. per sq. in. at maximum, the pressure tending to burst a hoop of metal 8-1/2" diameter and 1" wide will be about 2550 lbs. At maximum pressure the piston will be at the inner end of the stroke leaving the 3/4" thick wall of combustion chamber exposed to a pressure of about 2550 lbs. The ring of metal resisting the pressure has a cross section of 1-1/2 sq. in., submitting it to a tensile stress of 1700 lbs. per sq. in., giving a factor of safety of about 10 for cast iron.

The cylinder head is secured by six stud bolts. These bolts must be of sufficient size that they may be screwed up tight that when an explosion occurs they will not spring enough to loosen the asbestos gasket enough to be blown out. In my experience with 3/4" stud bolts I have always feared to draw hard on them with a wrench on account of danger of snapping off or stretching. By using six 7/8" diameter bolts, I have a cross sectional area of 2.52 sq. in. at root of thread. This area with a total end pressure of 17100 lbs. gives a tensile stress of about 6700 lbs. per sq. in. or about 1/8 to 1/9 of the maximum strength.



To fasten cylinder to base plate I used eight 1" stud bolts having a spring washer under a hexagon nut. These bolts are subjected to a shearing action which is equal to the reaction against the piston head amounting to about 17100 lbs. These bolts have a cross section of 6.3 sq. inches of steel which causes a shearing stress per square inch equal to 2700 lbs. If all of the bolts were made to fit the holes snugly, there would be a factor of safety of about 20, but since it is not likely that they will not all fit snugly in the holes, the factor of safety will probably be reduced to about 10.

Valves: The poppet or mushroom valve is generally accepted as the most practical valve for a gas engine; it has the advantages of having no friction on its seat as with the slide valve, also in requiring no stuffing boxes, and having a high efficiency. The main objection is the noise caused by opening the cylinder so quickly that the escaping gases set the air in vibration. High efficiency in poppet valves is secured by having large valves and quick action thereby having the advantage of utilizing the expansion of the gases to the end of the working stroke, and a free exhaust without great back pressure. The large valves that could be placed in the head were 2-3/8" inlet and 2-3/4" diameter exhaust.

By placing them side by side in the head the exhaust valve is greatly cooled by the incoming charge. The travel of the valves should be enough that the cylindrical opening around the valve when at its maximum distance will equal area of valve. Following this rule gives a lift of 5/8" for exhaust valve. In high speed engines pounding on seat of valve on return to seat,



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is remedied by design of cam on closing. As quick opening of valves is desired and as the cams run both ways, the practice of designing cams for slow closing cannot be followed. Using a light closing spring and a stiff bumper spring materially reduces the pounding of valves on their seats. The sizes of springs must be determined by experiment. The inlet valve has automatic action.

Cams: In designing exhaust valve cams, enough steel must be left around the shaft; in this case  $1/2$ " will do.  $1-3/8$ " shaft will make a base circle of  $1-7/8$ " diameter, by adding the  $1/2$ " for metal. To obtain the angle between the tangents of opening and closing, allow  $10^\circ$  for advance,  $90^\circ$  for opening,  $5^\circ$  for exhaust gases to come to atmospheric pressure and about  $5^\circ$  on each side of cam for back lash in cam mechanism, making  $115^\circ$  between tangents of opening and closing.

Carburetor and Intake Piping. Practice has demonstrated that a 2" intake pipe and a 2" standard carburetor will be large enough for this size of engine. A Schebler automatic carburetor with balance piston throttle is to be used in connection with a cam shaft centrifugal governor together with a throttle lever.

Gasoline tank is placed above carburetor to obtain a gravity feed. The carburetor is fitted to a  $2-1/2$ " pipe (water jacketed with a 4" pipe) going to either end of engine and branching 2" pipe to each cylinder. Said water jacket serves two purposes; First and most important, to vaporize all gasoline that may cling to the pipe and to keep both gasoline and water of the atmosphere from condensing on inside of intake piping and running to lowest cylinders. Second, to help cool the circulating water or to condense steam that may form.  $2-1/2$ " pipe was used



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for two reasons; first, to give more heating surface; second, to lessen resistance to reversal of direction of flow in charge and also to give a gas bag effect to carburetor, causing a steady flow through the same.

Exhaust piping. Practice has shown that a 2-1/2" pipe will serve this size cylinder. The exhaust outlets are so connected up as to give a very good muffler effect and also support canopy top. Area of cross section of pipe equals 4.9 sq. in. Extended length of pipe equals 34 ft. 4 in. Volume of piping equals 2020 cu. in. or 2-1/2 times the total volume of gases in one cylinder at end of stroke. By connecting all exhaust piping together with ells, tees, and cross, the whole system will be under nearly uniform pressure before gases escape at the outlet at the cross. This reduces the pressure by cooling and expansion until exhaust is nearly noiseless.

Water system and cooling tank. Weight of engine without water is about nine tons, which is from 3 to 6 tons lighter than the same rating steam engine. A tank that will hold a ton of water would not be too heavy and will perhaps be needed at times to hold front end of engine down, depending somewhat on height of hitch and steepness of grade. A tank 28" wide will fit on the front end of frame very nicely and yet not obstruct road view very much. It is not possible to put on a tank of sufficient radiating surface to take care of all the heat from cylinders without a great deal of expense. Therefore for this engine, a large tank having considerable height is used with a cover so that steam will be retained and very little water lost by evaporation. The tank is made of 12 gage black steel and riveted with 1/4"



rivets, 1" pitch. Top is dished so that it can be riveted from the outside and for convenience in filling, as all water spilled drops into the tank. A rectangular hole 11" x 15" having a loose fitting cover is left in top for manhole.

Connected to bottom of tank are four 1-1/4" pipes, each leading to a cylinder. These pipes must be of the same size so that they will not be clogged with sediment, as they lay horizontally their full length. Each pipe is provided with a three way cock at end of horizontal run near tank so that each cylinder can be drained separately and circulation tested.

It is necessary to have two outlets to each cylinder otherwise there would be an air trap over the cylinder or behind the valves. Two 3/4" pipes lead from outlets of each cylinder to underside of 4" pipe water jacket of intake pipe. Four 3/4" pipes connected into two 1-1/4" pipes carry water and steam from water jacket into cooling water tank. These pipes are all pitched so they will drain back through cylinders.

Lubrication of cylinders: The oil is fed to cylinder by an ordinary gas engine lubricator. The piston is provided with an extra ring at the crank end. This ring keeps the leakage gas from blowing all the oil off the piston and cylinder walls and uses the pressure of that gas to force oil through the groove near piston pin to lubricate same.

Crank shaft and bearings: The crank shaft has two cranks set at 180° to each other, two connecting rods being attached to each crank pin. The experience of others has shown that it pays to have a crank shaft of ample size; First, so that it will not be liable to fracture; Second, that it will not



spring and cause binding, heating and unnecessary wear on bearings. Three bearings can be had on this shaft but the middle one must be divided in the center to admit worm valve gear. To make engine symmetrical and convenient, the two front cylinders are set as close together as possible and the rear cylinders set apart to admit cam shaft.

The cylinders, the longitudinal and cross beams of I beam frame, the crank shaft, clearance of belt wheel and traction (compensation gear especially), must each be considered in the general design so that all parts will have their necessary clearance without crowding some other part.

The usual practice has been to use a 3-1/2" crank, with a 3-1/2" pin for this size of cylinder. The bronzes from the company were designed for a 3-1/2" pin and I decided to have a 3-5/8" diameter crank at ends, 3-3/4" diameter at central bearings, and arms 2-1/2" x 4-1/2". These dimensions give 11.25 sq. inches cross section in arms compared with 10.4 sq. inches in the crank, giving nearly uniform strength of shaft from end to end, making it stiff enough so that the crank pins can be machined in an ordinary lathe. These dimensions give very high factors of safety, from shear at time of ignition 50; arms at greatest bending moment a factor of 15 to 20; shaft at greatest twisting moment a factor of 15 to 20, but practice has shown that they cannot be safely cut down. Allowing for each connecting rod 3" of pin, and 1" for central worm gear, leaves 18" in length for crank shaft bearings, divided into two end bearings of 6-1/4" each and two center bearings of 2-3/4" each. This is none too much bearing surface for a babbitted bearing and will have to be adjusted more



often than if each of them were about one-half longer. However, this is all the space that can be allowed on this engine without interfering with some other parts. The best babbitt on the market must be used and care taken not to babbitt bearing when crank is sprung from weight of fly wheels.

Bearings: In the design of crank boxes, strength, rigidity and plenty of bearing surface are the main points to observe. The most practical crank box is a two part bearing split at an angle of 45° to the base. A box 6-1/4" in length makes the bearing pressure about 375 lbs. per square inch of projected area which is considered reasonable.

The bases of the bearings fit snugly between the fore and rear cylinder bases thus adding greatly to their resistance to sliding. Each end bearing is fastened to frame with eight 1" stud bolts, all having spring washers underneath hexagon nuts.

Bearing caps are machined to fit in place on bearing and are held in place by four 1" stud bolts, hexagon nuts and lock nuts. The maximum tensile stress per square inch on these stud bolts is 2700 lbs. or a factor of safety of 20 for steel bolts. All bearing caps must have an oil groove parallel with the shaft. All caps are tapped 3/8" pipe size for spring compression grease cups.

Connecting rods: The connecting rods are of cast steel of I beam section, nearly parallel from end to end. Cross sectional area is 2.5 sq. in. Maximum compressive stress per sq. inch equals 6800 lbs., giving a factor of safety of about 15.

Flywheels: The flywheels of a four cylinder throttling gas engine need be but little heavier than those of a single



cylinder steam engine, for the impulses come at the same intervals. The gas engine has a little heavier impulse at the beginning of the stroke and a little more retardation at the end of stroke, due to compression. For convenience I used the steam engine flywheels and clutches. One flywheel is keyed solid to the shaft and one clutch arm keyed to the shaft. When reversing engine and backing into the belt, one flywheel is idle.

Gearing and countershafts: The gearing is crucible cast steel because of its great strength and ability to withstand shocks and sudden strains; It also wears well if kept lubricated and protected from dirt. By making the number of teeth in any one gear not a multiple of any other gear in the same train, the tendency is to wear more evenly. The clutch pinion has 17 teeth, intermediate gear 63 and compensation gear 63 teeth. The bull pinion has 16 teeth and bell gear 73 teeth. The ratio of the gears and size of same, has been determined by gears of field use and will likely undergo no radical change for some time.

Strength: For clutch pinion height of teeth equals 1.1875".  $M = fz$ , where  $M$  = moment in inch pounds, also  
 $M = 3100 \times 1.1875$  when transmitting a torque of 70 H.P.  
 $f$  = fibre stress in lbs. per sq. in.  
 $z$  = section modulus = .445

Substituting,  $3100 \times 1.1875 = f.445$

$f = 8272$ , giving a factor of safety of about 9. All the teeth in the first train of gearing are of the same size and of equal strength. The teeth of the bull gearing are of the same size as first train of gearing. However, there are two sets to take the strain but the strain is 3.63 times as great as before. Therefore, the factor of safety instead of being 9 times



is  $2 \times 9 \div 3.63$  or about 5. These gears run at .275 of the speed of the first train, hence the wear will be nearly the same.

Cross shafting: The shafting should be large enough in diameter so that plenty of bearing surface can be obtained without too great a length of box. Another very important requirement is stiffness so that the gearing will not be thrown out of line, thereby causing a liability of breakage or cutting of shaft in bearings. The cross shaft must be large because of the great distance of box from bull pinion. This distance cannot be reduced without putting compensating gear between bearings which would make matters still worse. Therefore the designers have put the compensating gear in as small a length as possible and increased size of cross shaft. The shaft used was 3-1/2" in diameter.

Distance from center to center of bull gears = 2 ft. 4 in.

Distance from center of bull pinion to center of box =  
13-1/2"

Upward lift on end shaft when transmitting 70 H. P. = 5600 lbs. Equating moments using projection of shaft as a cantilever beam, we have  $M = 5600 \times 13-1/2" = fz$ , where

M = moment in inch lbs.

f = fibre stress in lbs. per sq. in.

z = section modulus =  $.0982D^3$  for round bars

From which,  $M = 5600 \times 13-1/2 = f \times 4.225$

Solving  $f = 18000$  lbs. per sq. in., giving a factor of safety of about 4.

Computing deflection  $d = \frac{WL^3}{3EI}$  where

d = deflection in inches

W = weight in lbs.

L = length in inches



E = modulus of elasticity

I = moment of inertia

$$\text{substituting, } d = \frac{5600 \times (13-1/2)^3}{3 \times 30,000,000 \times .0491 \times (13-1/2)^4} = .023''$$

which is the deflection in 13-1/2". In the width of gear the deflection equals  $\frac{3-1/2}{13-1/2} \times .023'' = .0059''$  which would soon be worn down in the gears. Each bearing has a length of 12", giving a maximum pressure of 130 lbs. per sq. in of projected area. The bearing is tapped 3/8" pipe size below and above for spring cup hard oilers. Each cross shaft boxing is forced up against frame when going ahead but pulls down when engine travels backward. The load on box equals the load on each bull pinion or 5600 lbs. The boxing is fastened on by four 1" bolts, altogether having a sectional area at root of thread of 2.2 sq. in. This makes the maximum fibre stress about 2500 lbs. per sq. in. or a factor of safety of about 20.

Cushion links: Each bull gear has five projecting pins each 1-1/4" diameter and cast in wheel at a radius of 18". The cushion links are attached to these pins and the probable failure is by shear. At a 21" radius the bull gear will transmit a maximum load of 5600 lbs., or at a 18" radius 6500 lbs. Cross section of five 1-1/4" pins = 6.15 sq. in., giving a shearing stress of 1060 lbs. per sq. in. This shows that one pin could take all the strain and then have a good factor of safety.

In each link is a 1" bolt that is put under a tensile stress. Area cross section at bottom threads is .55 sq. in., which under a maximum stress of  $\frac{6500}{5}$  lbs. or 2360 lbs. per sq. in. gives more strength than is really needed.

$$\text{Maximum load on each of the five springs} = \frac{6500}{5} = 1300 \text{ lbs.}$$



Using Wilson Hartnell's formula,  $d = \frac{5.584 W n D}{C s^4}$

Where  $d$  = deflection in inches

$W$  = weight in lbs.

$n$  = number of turns of spring

$D$  = pitch of spring

$C$  = torsional modulus of elasticity

$s$  = side of stock in inches

The load of 1300 lbs. just lacks .01" of closing each coil.

Spring is under compression and would stand 4% more load before closing.

Distribution of weight: When crossing bridges it is always desired to have the weight of the engine equally distributed between the front and rear wheels. There must be enough weight ahead of rear axle to hold the front end down when pulling a load hitched on high in the rear.

Computing the greatest possible pull we have, pressure on bull gear teeth = 11200 lbs. This is at radius of 21" and radius of rear wheels is 32". This gives a possible pull of 7350 lbs. and a moment about rear axles of  $\frac{32 \times 7350}{12} = 19500$  ft. lbs. For simplicity let cross shaft and gearing balance platform and disregarding front wheels and axle, we have (when tank is full of water) a moment of 46750 ft. lbs. Deducting the moment of 19500 ft. lbs. leaves 27250 ft. lbs., this acting through a distance of 10 ft. (wheel base) gives a load of 2725 lbs. on front wheels.

No figures can be intelligently used in computing width of tire and height of drive wheels. Practice has determined width of tire and height of wheel. For farm work packing of the soil must be avoided. It causes the ground to dry out as deep



as packed and nothing but wet weather and frost will break the soil up again. A good self cleaning cleat is necessary, as well as detachable cleats or mud hooks that will go down from 4" to 6" and get good footing if possible in wet plowed fields, mud holes, sand, etc.

It makes a great difference where counter shaft is placed on an engine when the pressure on the rear hub bearing is considered. When the tractive effort is at maximum (7350 lbs.) the pressure on counter shaft bearings is 11200 lbs. If counter shaft is placed directly over rear axles, the backward pressure on rear wheel hub bearings is (7350 + 11200 lbs.) and downward pressure is weight of engine minus rear wheels and load left on front wheels. When counter shaft is on a level with rear axle as in this engine, the backward pressure on rear wheel hub bearings is only the tractive effort. If bull gear was rigid to rear wheel, then the lift on counter shaft or pressure on teeth is greater than amount of engine carried on rear wheels and rear wheel hub pressure is an upward pressure. In this case when pulling 7350 lbs., about 14000 lbs. of engine is carried on rear axle and an uplift on teeth of bull gear of 11200 lbs. leaving only 2800 lbs. to be carried by hub bearings. This would be the case where the 14000 lbs. was directly over the rear axle but engine is nearly an evenly distributed load from end to end. This leaves only 7000 lbs. for rear hubs and there must be an uplift of 4200 lbs. In this engine the bull gear is on a separate bearing of its own, giving a rigid center to gearing train. The downward pressure on this bearing must equal the upward pressure on counter shaft. There being two of these bearings 3-1/2" long and 10-1/2" diameter, we have a pressure per square inch of



projected area of,  $\frac{(5600)}{3-1/2 \times 10-1/2} = 150$  lbs. This is plenty pressure for a bearing that is difficult to keep lubricated. The backward pressure on rear hub bearing is equal to 7350 lbs. and downward pressure equals 14000 lbs. These act at right angles and their resultant is 15800 lbs. This is equally distributed between 4 bearings, 6" in length and 6-1/2" in diameter, giving a pressure per square inch of projected area of 104 lbs.

Mounting springs: Mounting springs are not an absolute necessity, however, they are a great improvement to an engine when proportioned and placed. On a very hard road or on stony roads, springs will absorb nearly all of the shock and in all cases lessen the shock to both engine and operator. The engine is carried from supports in the center of the rear wheel hubs. There are two compression coil springs in each wheel. Stock 3/4" square, pitch 1-1/4", pitch diameter 3-1/4". By Wilson Hartnell's formula given on former page;

$$d = \frac{5.584 W n D^3}{C s^4}$$

$$n = 4.4 \text{ turns} \quad D = 3-1/4"$$

$$s = 3/4"$$

$$W = 3500 \text{ lbs.}$$

$$C = 12,000,000$$

$$d = \text{deflection} = 0.95"$$

The distance allowed in hub for play of spring is 1-1/8". These springs were made for a steam engine which would be from 10 to 20% heavier. When not pulling, the weight on springs is about 1/2 that of loaded so that deflection would be about 1/2".

Rear Axle: The load on rear axle at rest comes on two points, one under each longitudinal I beam of frame. These points are 14" from center. Moment anywhere between A and B =



$$W_1L_1 = W_2L_2 \text{ (L in inches)} = 3500 \times 25-1/2 = 89000 = M$$

Axle is 3" square and a fairly high carbon steel.

$$M = fz \qquad z = 1/6 B^3 \qquad B = 3"$$

$$89000 = f \cdot 27/6$$

$$f = 19700 \text{ A factor of safety of}$$

about 4.

Deflection.

$$d = \frac{WL^3}{3EI}$$

$$W = 3500 \text{ lbs.}$$

$$L = 25-1/2"$$

$$E = 30,000,000$$

$$I = 1/12 B^4 = \frac{3^4}{12} = 27/4$$

This gives a deflection of 0.095", when engine is idle. When engine is pulling the conditions are different. Greatest moment = WL = 32500, giving a maximum fibre stress of 18600 lbs. per square inch against 19700 when standing idle. There is an additional moment at A when under load. A moment of 3675 times 11-3/4, at right angles to weight moment. This is a little more than half the supporting moment. This puts a little more than a half more strain on shaft and cuts factor of safety down 1/3, or 3-1/4. Extensions are aimed to be screwed to ends of rear axle so that a hitch can be made outside of wheels and take strain off both axle and center of draw bar.

Front Axle: It is difficult to tell just what will be required of a front axle. For instance, when an engine is crossing a ditch, front wheels drop into a hole, computing the downward strength first, we have the following conditions; a 3" sq. steel axle, trussed with two 3/4" rods. Computing maximum moments about center;



$M = fz = 60,000 \frac{1}{6} B^3 = 270,000$ . 2-3/4" rods at bottom of thread will stand 36000 lbs. tension. This amount of tension in this truss will give a moment about center of 261,000 in. lbs. The thrust of truss is taken by a flat bar 5/8" x 2-1/2" placed on top of axle. The sum of moments = 270,000 + 261,000 = 531,000 in. lbs. This moment at a distance of 41 inches requires a supporting force of 13000 lbs. 13000 lbs. can be placed on each front wheel or 26000 in center of axle. Normally the maximum load is 7000. Thus it gives a factor of safety of about 3-1/2 or engine could drop about 3 ft. suddenly and not bend axle.

When front wheels are run against an embankment of any sort, there is a tendency to bend the ends backward. The moment of bar has already been computed as 270,000 in. lbs. There is but a single truss rod in this case and its maximum rise is 8" but 7-1/2" would be safer to use in computation. This gives us the maximum resisting moment of rod, 1/2 that of the two rods in the first condition; viz, 130,500 in. lbs.; adding, 270,000 + 130,500 gives 400,500 inch pounds; this acting through 41" supports or pushes 9750 lbs. or 19500 for both wheels. This is 2.65 times what engine can pull.

Front end carriage spring: Front end is supported by a single spring 3-1/2" long, 4" outside diameter, 3-1/8" pitch diameter and 7/8" square stock, 1-3/8" pitch. Using Wilson Hartnell's formula; with no load, distance between coils = 1-3/8" - 7/8" = 1/2" which is the maximum deflection allowable.

$$d = \frac{5.584 W D^3}{C s 4} = \frac{5.584 \times 7000 \times 3.125^3}{12,000,000 \times (7/8)^4} = 0.238"$$

So this spring will carry twice the load before coming together.



Engine frame proper: For dimensions, a ten inch standard I beam works in better than any other shape or size. These I beams constitute only a small part of the weight of the engine and a small increase in weight greatly increases the strength and rigidity. Using two longitudinal and four cross I beams, makes a good frame to begin with. The engine or center of weight of same is directly over mid point for length of I beam. Computing for deflection will consider entire weight of engine as concentrated at center and neglecting weight of beams themselves. Then engine is idle moving on road under no load or standing still, the condition is as follows:

$$\text{Deflection} = \frac{WL^3}{48EI} = \frac{6000 \times (120)^3}{48 \times 30,000,000 \times 122.1} = 0.059''$$

This is not enough deflection to hinder working of cylinders nor is it liable to take up an up and down excessive vibration. For strength, the maximum moment will be at center.

$$M = fz \quad M = 60 \times 3000 = f \frac{I}{y} = \frac{122.7 \times f}{5}$$

$$M = 180,000 = f \times 24.5$$

$$f = 7370, \text{ giving a factor of safety of about } 9.$$

When pulling, the length of beam will be shortened up to cross shaft and load lightened by up lift so that the strain will be taken off beams. When pulling backward there will be a moment on beam at counter shaft of  $5600 \times 24 + 3000 \times 24 = 216500$  in. lbs.  $M = 216500 = fz$ ,  $z = 24.4$ ,  $f = 8900$  lbs. per sq. in. or a factor of safety of about 7. To prevent all racking of frame and to give cylinder bases and crank boxes a rigid support, a sheet of boiler plate 41" x 60" x 5/8" is riveted to longitudinal and cross I beams of frame. Cylinder bases, and crank boxes are bolted through the plate and frame wherever possible. This plate



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prevents all racking and gives rigidity to twisting. To the rear of this plate the I beams are braced by two diagonal  $3/4$ " x 2" braces to prevent side bending effect on I beams when on slanting ground. Weight at front end is carried by a 5" Z bar,  $1/2$ " metal.

$M = 14 \times 3500 = 49000 - fz$ ;  $z = 7.68$ ;  $f = 6400$ , a factor of safety 10. This Z bar is riveted to a 1" x 6" fifth wheel plate. This plate is subjected to a bending moment of about  $3500 \times 3 = 10500 = M = fz$ .  $z = 1$  in this case.  $f = 10500$ , giving a factor of safety of about 6. I beams, engine bed plate, Z bars and fifth wheel plate, are all riveted together in one mass with  $3/4$ " rivets, driven until cold to insure tightness.

Platform: The hitch must be below the platform and should not be lower than 24" for general purposes. If platform is high the operator can see ahead better as he can lean out over the drive wheels. The handiest place to attach hitch or coupling on this engine is on to angle irons running backward from under side of rear axle. This is 30" high. By using a solid sheet metal platform no horizontal bracing is needed. It is not worth while to put on less than  $1/4$ " metal; thinner metal would soon be too weak from rusting. The size of platform is 4 ft. x 8 ft., leaving 13 inches of standing room behind each drive wheel so that the operator can get a better view ahead without dismounting. Rear edge of platform is stiffened by a 3" angle iron.

The regular spring coupler or draw bar is bolted to a plate, which plate is bolted to same angle irons as platform but in front of rear axle. From this coupler a swinging draw bar reaches to rear of platform, having on end of same a patented automatic coupler. A number of equidistant holes are drilled



through platform and stirrup so that pins can be put on either side of draw bar to hold it stationary if so desired. A foot trip for automatic coupler is placed near rear axle where operator can give it a kick and loosen load at any desired moment whether pulling or not.

Winding drum: For convenience in pulling load up to engine again, after having to leave same in a mud hole or in passing over a place where engine drive wheels cannot get a foothold, a drum and cable devise is used. Engine can be run ahead on to good ground, or stopped in the mud and blocked or chunked and load can be drawn up to engine again and process repeated.

A 3-1/2" shaft is put into intermediate gear instead of putting said gear on a stud bearing as usual. On this shaft a drum is fitted loosely; end of drum forming one-half of a jaw clutch which is driven by the other half keyed solid to shaft. This drum must be smaller in diameter than the pitch diameter of bull gears or cable will travel faster than engine itself. Again, if it is made over 9" the cable would wear on cross I beam. There was room for a 7-1/2" drum 12" long and clutch to drive same on shaft. The breaking strength of a 5/8" steel cable is about 14 tons. Pull of drum may possibly be as high as 6 tons but is not likely to be over 3 tons. A 60 ft. cable can be wound on a 7-1/2" drum 12" long and not over wind.

It is necessary to have a guide for the cable. The best guide would be an ordinary sheave on a heavy shaft to hold cable down and a flanged sheave on a light shaft below to keep cable from getting off top sheave. The shaft is held at each end; supports 15" apart. The shaft is considered as loaded maximum when sheave is in center; the resultant of the two forces



of the cable on each side of sheave is 2914 lbs. The resultant of the parallelogram of forces in this case equals twice the sine of half the angle (21°) or 10-1/2° into the load. 8000 lbs. = 2914 lbs.

$M = fz, M = 7-1/2 \times \frac{2914}{2} = f 0.0982D^3 = 23000 \times 0.0982 \times (1-11/16)^3.$   $D = 1-11/16"$ . 8000 lbs. is the maximum load cable is likely to have and a fibre stress of 23000 lbs. per sq. in. gives a factor of safety of 3, which is large enough. I used a 1" shaft for protecting sheave. There is no room for a larger diameter of sheave pulley than 7" and this will answer the purpose.

Couplers: The front hitch is a malleable casting with clevis and also a large bull ring, the whole forming an extension to the fifth wheel plates. The bolt in the bull ring clevis is 3/4" diameter and is subject to single shear in two places. Area = 1.12 sq. in. Maximum steady pull = 7350 lbs. This gives a shearing stress of 6750 lbs. per sq. in.

Rear coupler: The rear hitch is equipped with a spring fixed to act either in pulling or pushing; spring being under compression in either direction. The spring is a helical spring 8-3/4" long, 3-1/2" pitch diameter, 3/4" stock and 1-1/4" pitch. From Hartnell's formula it will take 9000 lbs. to close spring, and maximum draft of engine is 7350 lbs.

$$I = \frac{5.584 W D^3}{C S^4}$$
$$.375 = \frac{5.584 W (3-1/2)^3}{12000000 (3/4)^4}$$
$$W = 9000$$

This spring takes the sudden strains off coupler, all connecting



parts of engine and load, when starting and stopping and when pulling over rough roads and the like.

Steering Mechanism: It is very important that an easy and effective steering mechanism be put on an engine. Not many computations can be made with any degree of certainty. Practice has proved about how fast or how far the front axle can be truned to one turn of the hand wheel. A mechanism that is well smoothed up and lubricated will work easily. If the worm wheel is allowed to become dry it will work very hard. In some engines the guide chains are provided with springs and in others (the Huber for instance) the springs are on the worm shaft. It does not take such heavy springs on the worm shaft and they are a little easier put on.

System of Reversing: The regular order of events in the four cycle gas engine is intake, compression, firing and exhaust. To reverse, the ignition must come early enough on the compression stroke to bring everything to a stand still before the engine comes to dead center. In other words, the work done on part of the compression stroke must be as great as the kinetic energy of all the mechanism of the engine at begining of compression stroke. If the engine stops just about at the end of the compression stroke, it has all the energy of the working stroke to pick up speed in the opposite direction and to compress in another cylinder. Has the engine gone on in the first direction at the end of the compression stroke, it would have fired and went on with the working stroke and then exhausted. However, the engine has gone backward one stroke and must have the exhaust valve opened at end of this first backward stroke to relieve cylinder of gases. Had the engine gone ahead the cam would



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have opened the exhaust valve at the proper time; as it is, the cam has gone backward  $90^\circ$  and is  $90^\circ$  ahead for backward direction at end of compression stroke that reversed engine. Now if we can set it backward  $90^\circ$  more by rotating cam shaft independent of engine, we will have cam thrown back  $270^\circ$  and ready to open exhaust valve at end of first reversed stroke. But now our timer has been thrown around  $270^\circ$  and we want it at the same relative place it was. This is rectified by a 4 pole switch and five terminals. By shifting all the poles, coil No. 4 becomes No. 1 and so on. Now when engine reverses, the cylinder fires in a reversed order but the timer takes care of that as it reverses in direction with cam shaft. Timer adjusting lever takes care of lead of spark. The reverse lever has a little crank attached to its mechanism that throws the 4 pole switch, the order of events is as follows; when wanting to reverse, throw out clutch that drives belt pulley. (this diminishes the momentum to be overcome by what might be called back firing when we reverse); throw off current and set timer adjusting lever to light ignition for opposite direction. This gives spark what we might call double lead. When speed of engine falls to about 50 R. P. M. throwing on the current will reverse direction of engine and during this half revolution, the reverse lever is thrown to opposite position and timer lever put up to central position as far as engine will permit without knocking.

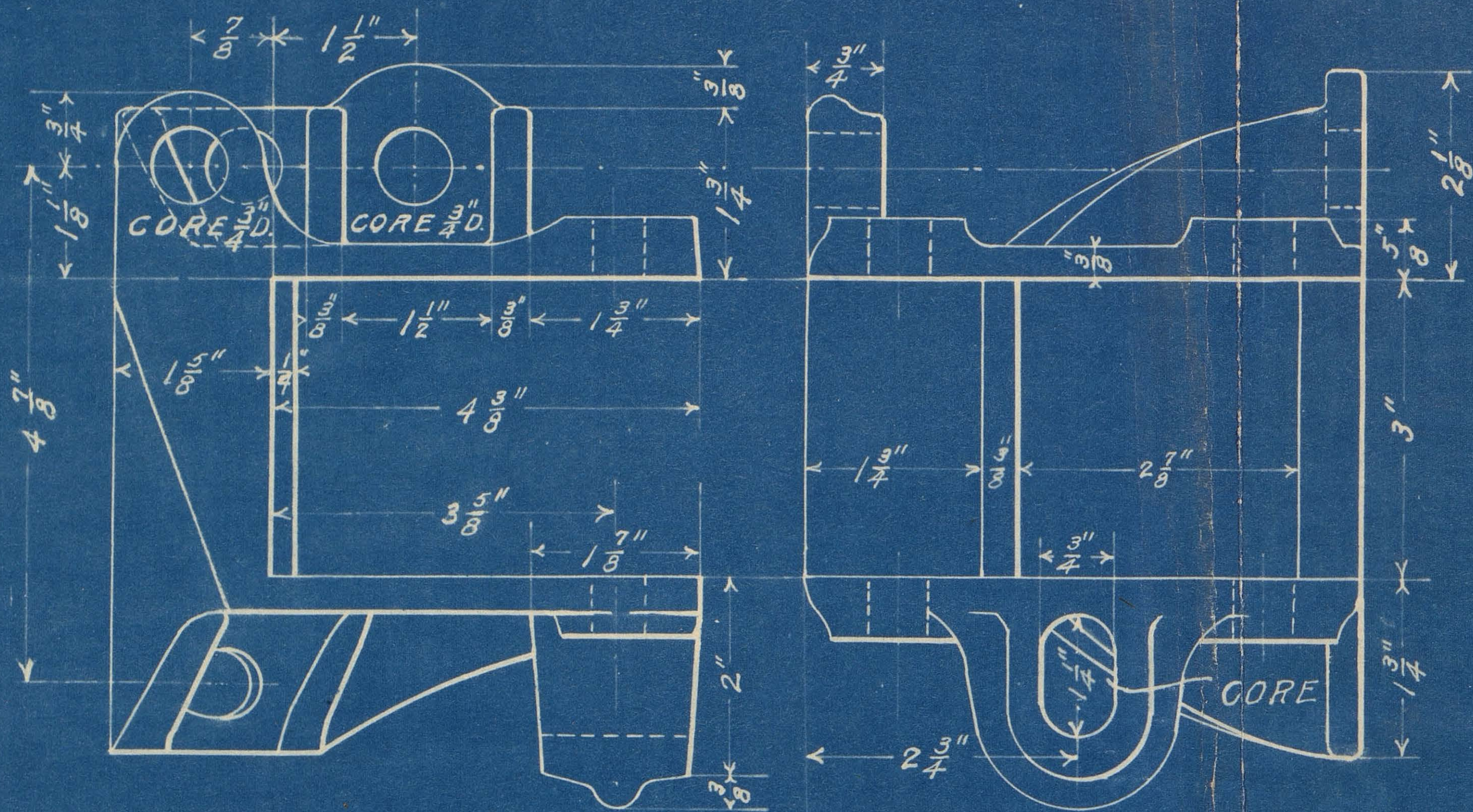
The rotating of cam shaft  $90^\circ$  relative to motion it would otherwise have, is accomplished by a train of bevel gears. Beginning with the crank shaft we have a two to one worm gear. Said worm gear drives a tube around cam shaft. This tube drives one bevel gear and another bevel gear keyed to cam shaft is



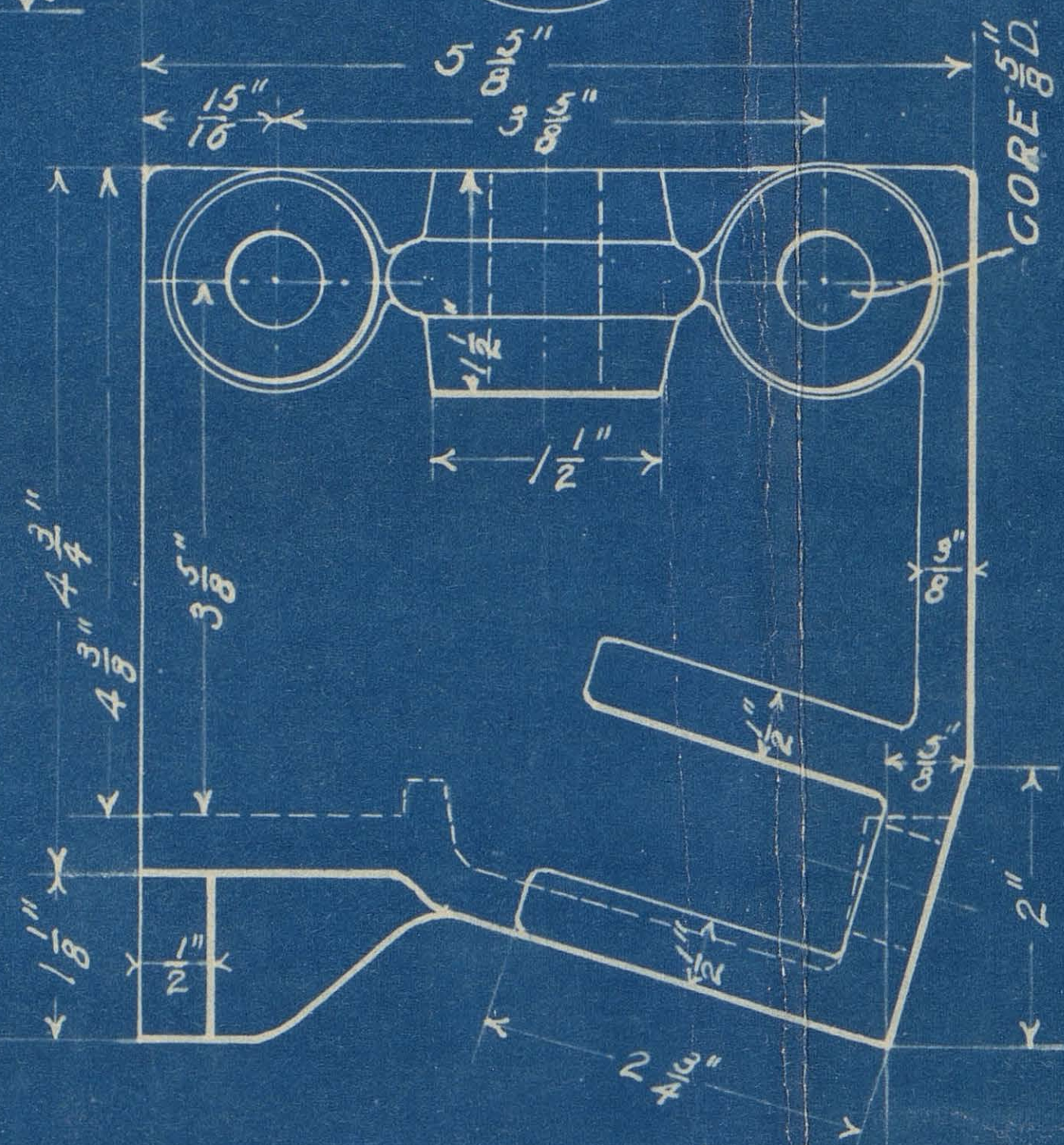
driven by an intermediate bevel on a radical shaft supported on a box on cam shaft. The reverse lever rotates this radical shaft through an angle of  $45^\circ$ , thereby rotating the cam shaft twice the angle. The reverse lever can be arranged to rotate the radical shaft about the cam shaft automatically during the first  $180^\circ$  rotation of crank shaft in opposite direction. This would be a decided advantage as far as manipulating levers is concerned, but the added complication of mechanism would perhaps counter balance the advantage gained.

A small fly wheel on cam shaft makes a decided improvement in the running of the same. Without the fly wheel when each exhaust valve closed it would rotate back lash out of gearing and just at that instant another cam would demand power to open next exhaust valve and this would bring gears back to a pulling or working position. Without the fly wheel all the work would be done by just a few teeth and with the fly wheel there would be more of a steady pull and even wear on teeth of gears.

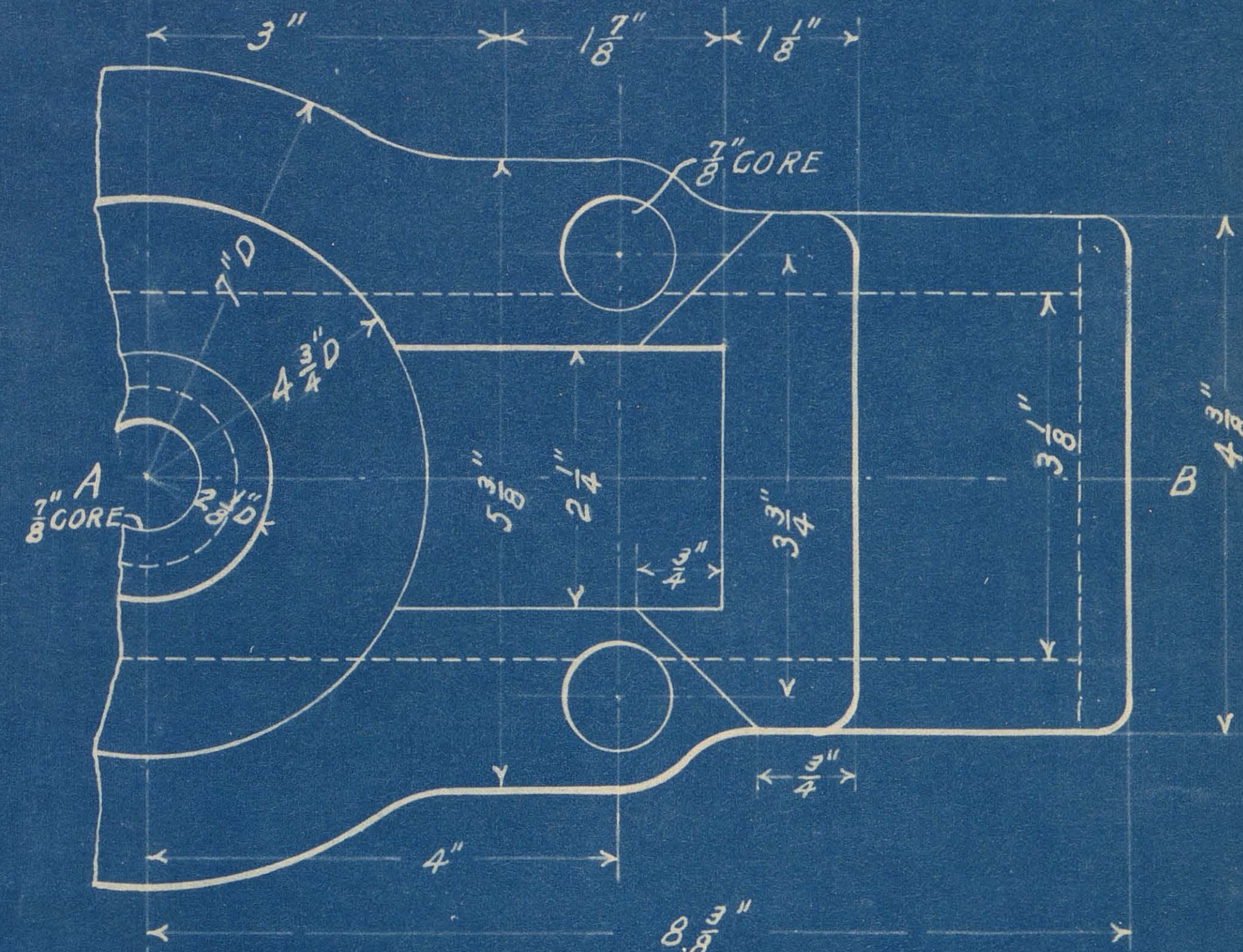




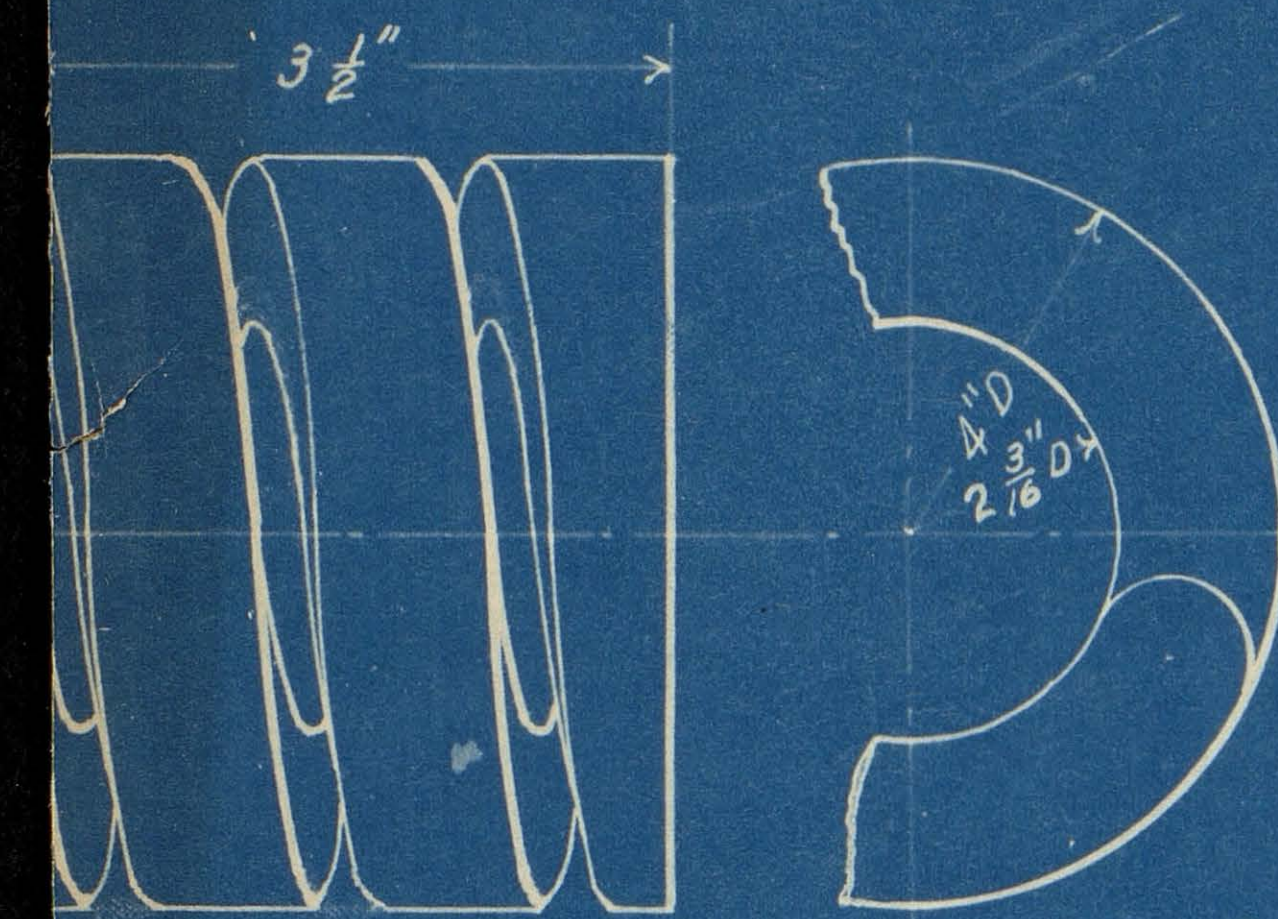
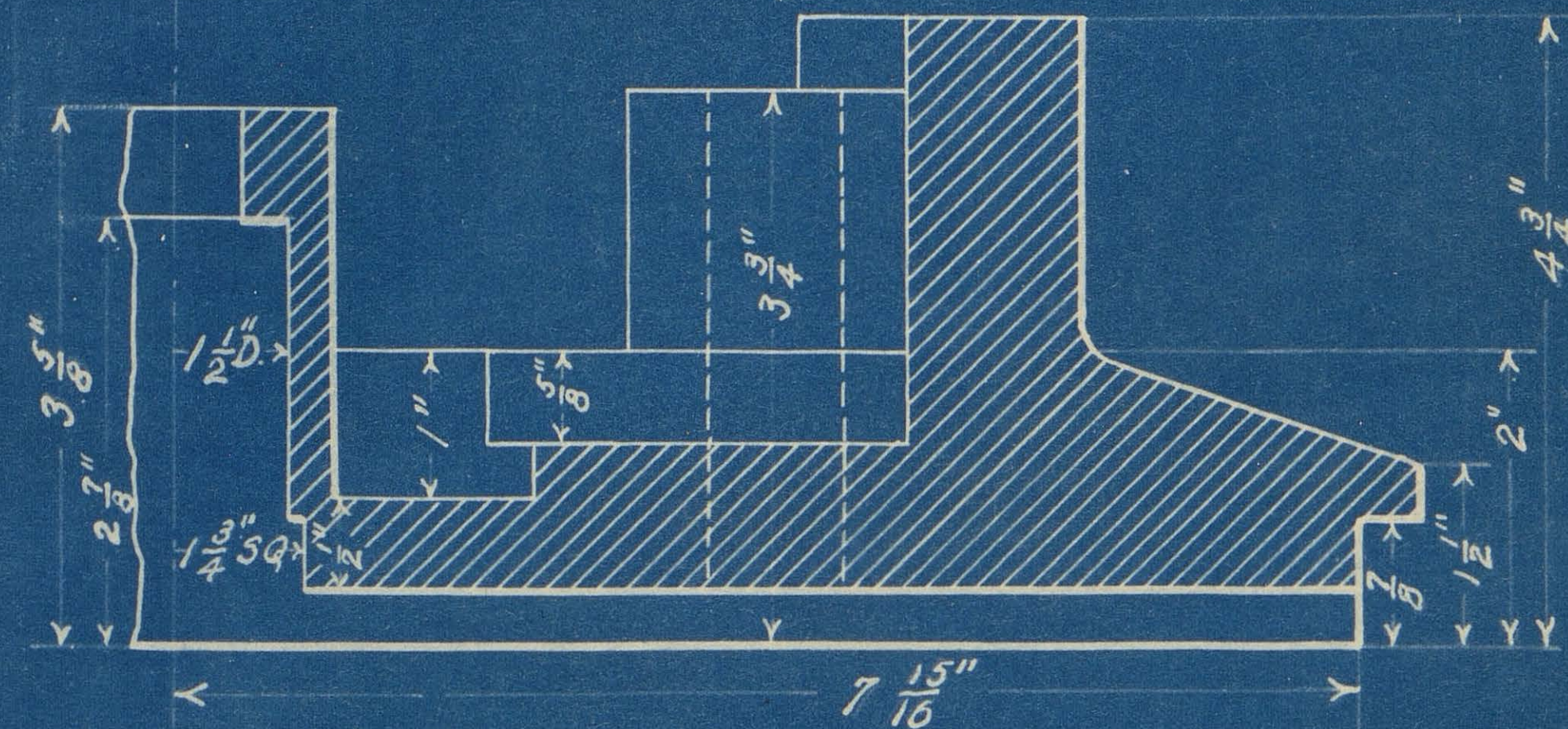
**Z182**  
**FRONT AXLE TRUSS ROD**  
**BRACKET**  
 REQ'D: 1-RIGHT-1-LEFT  
 GRAY IRON NOT MACH'D



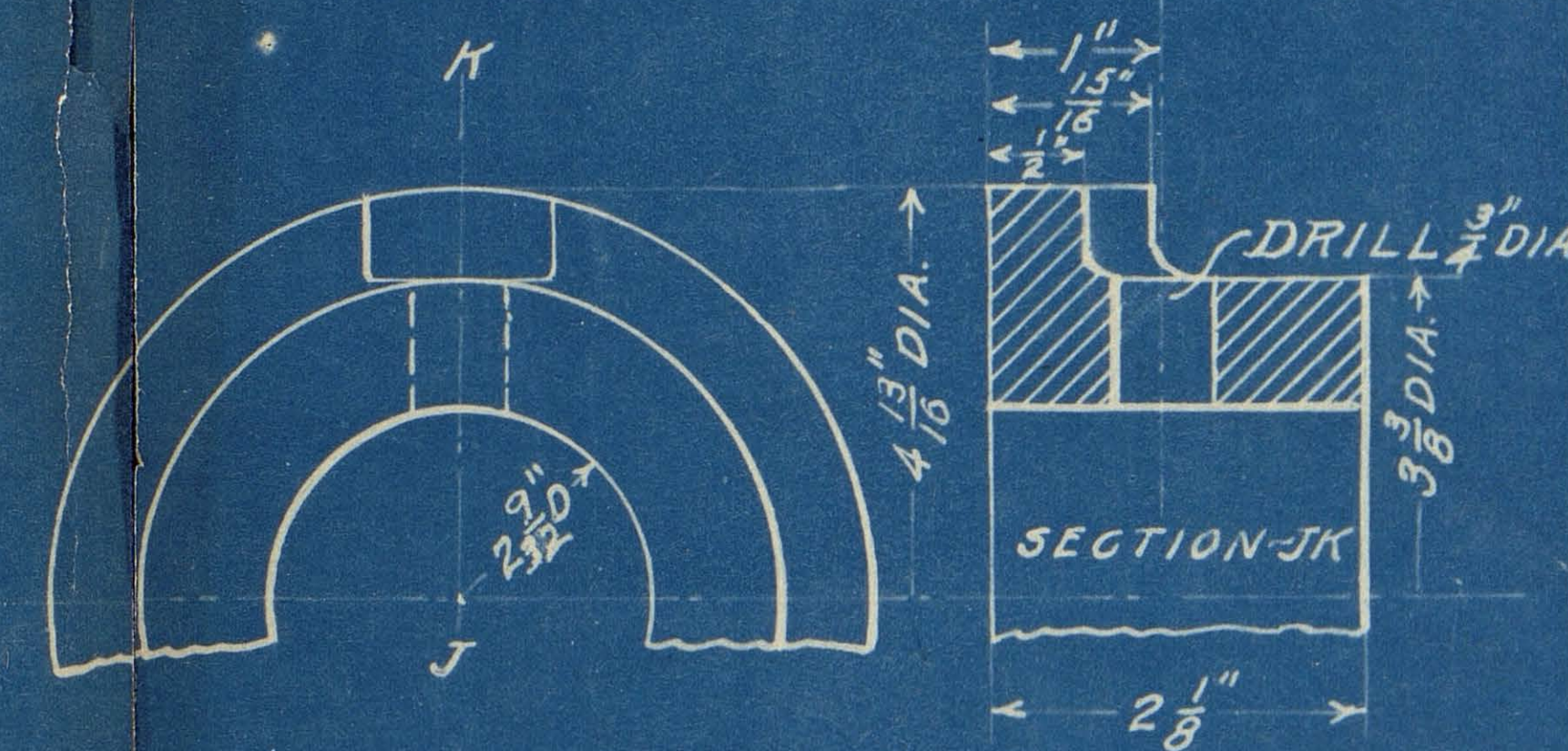
**ZZ21**  
**FRONT AXLE SPRING CAST'G**  
 1 REQ'D NOT MACH'D



SECTION ON AB



**Z227**  
**FRONT WHEEL AXLE CAP**  
 2 REQ'D GRAY IRON



**Z227**  
**FRONT WHEEL AXLE CAP**  
 2 REQ'D GRAY IRON

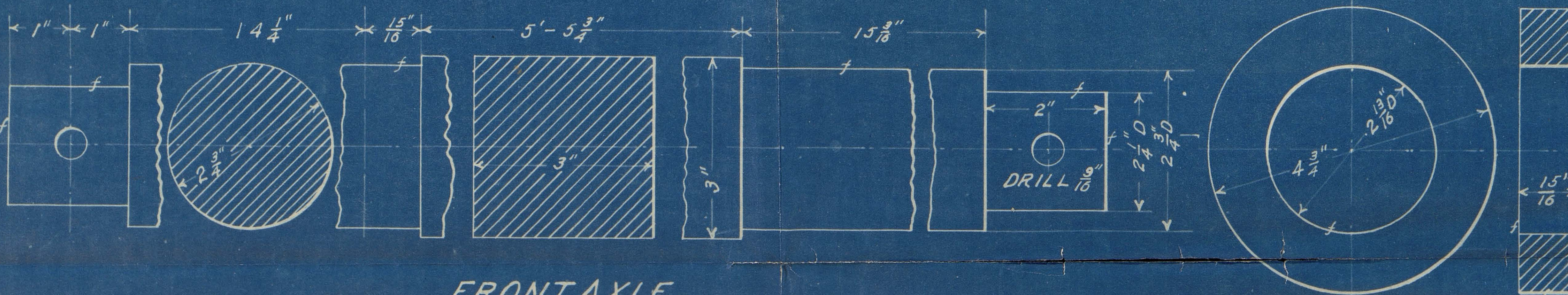
**DETAILS OF**  
**25 HP-GASOLINE**  
**TRACTION ENGINE**

SCALE: 6"=1FT. f MEANS FINISH

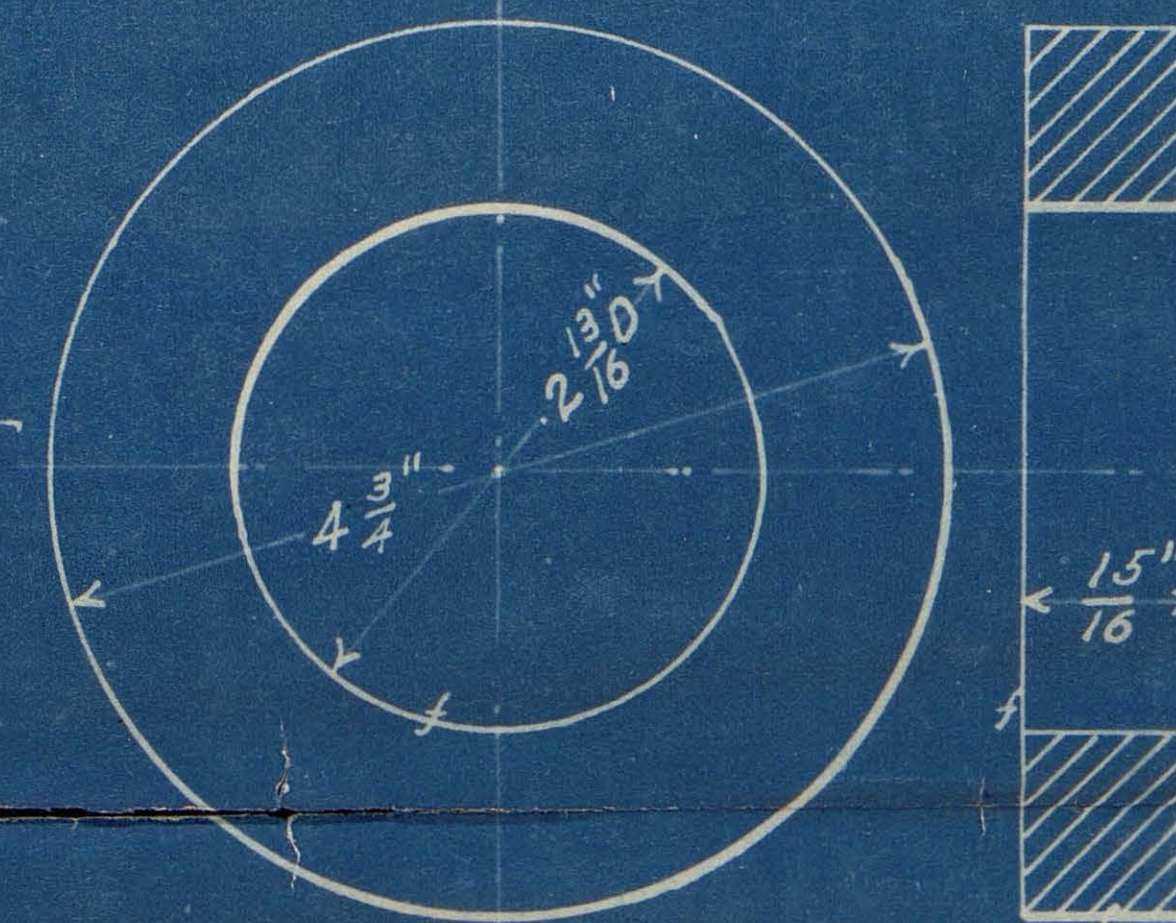
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DEPARTMENT OF  
**MECHANICAL ENGINEERING**  
**K.S.A.C.**

MANHATTAN — KANSAS  
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 DATE 4-3-'05

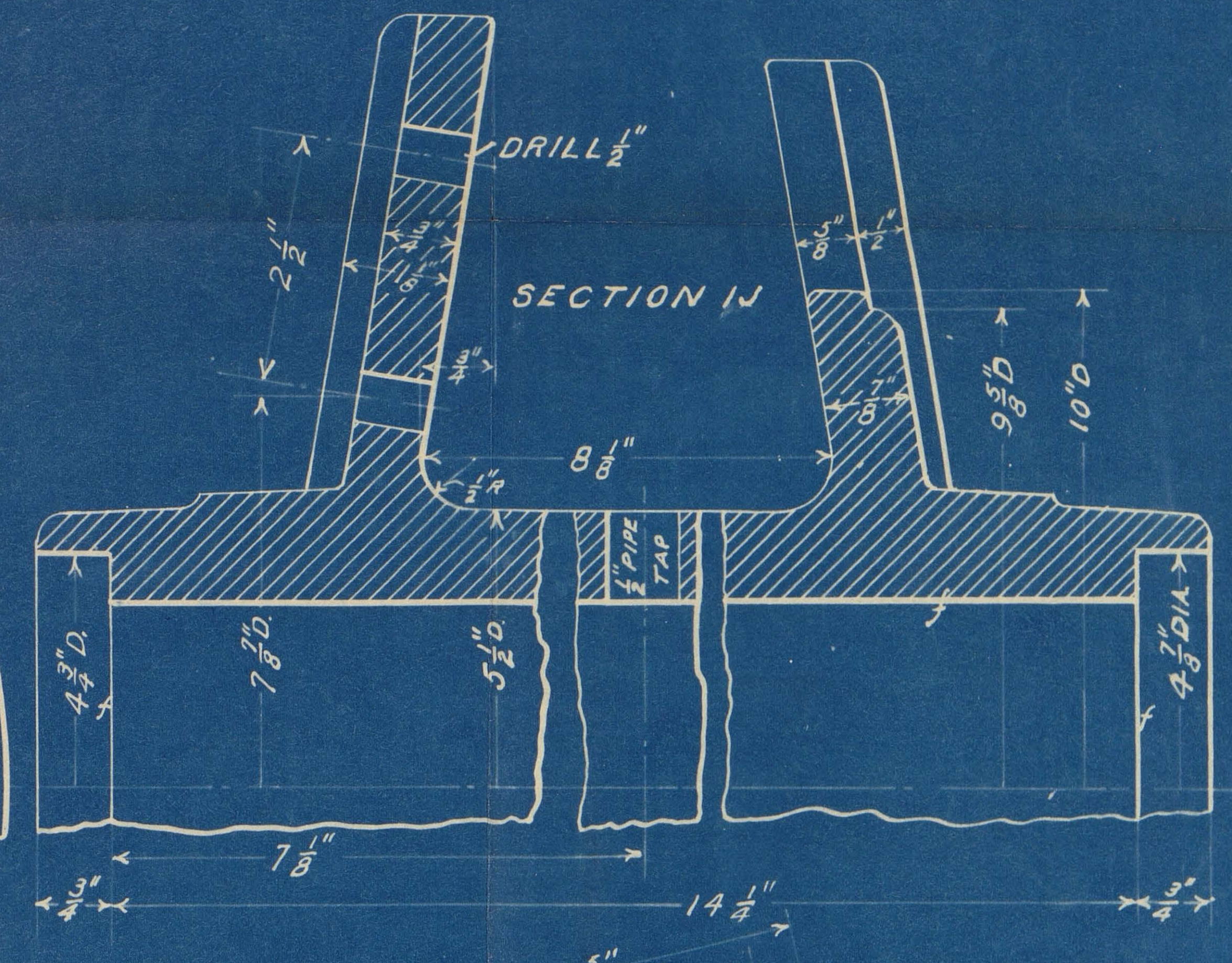
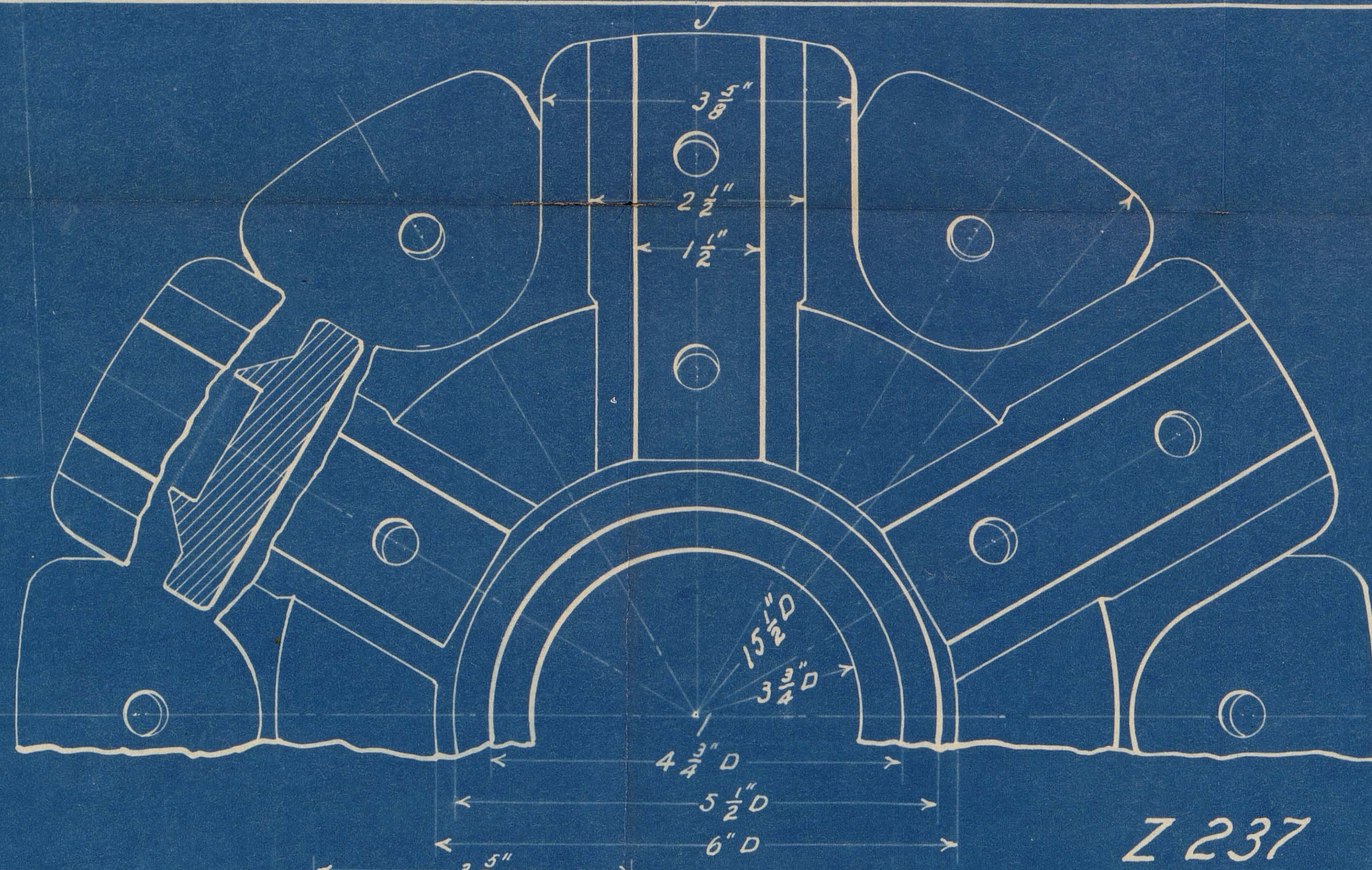


**FRONT AXLE**  
 MILD STEEL-1-REQ'D

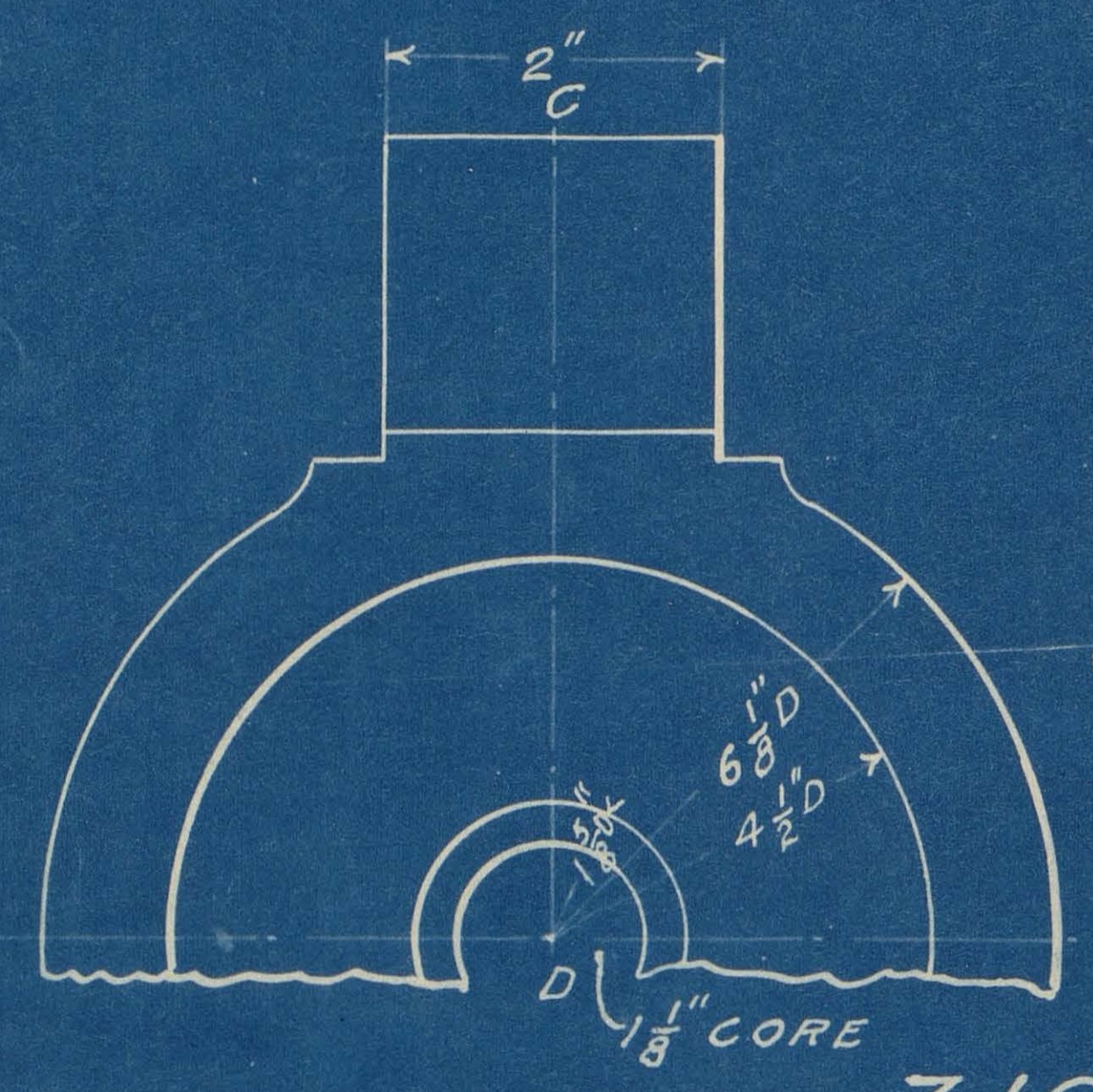


**COLLAR**  
 2 REQ'D W'RT IRON.

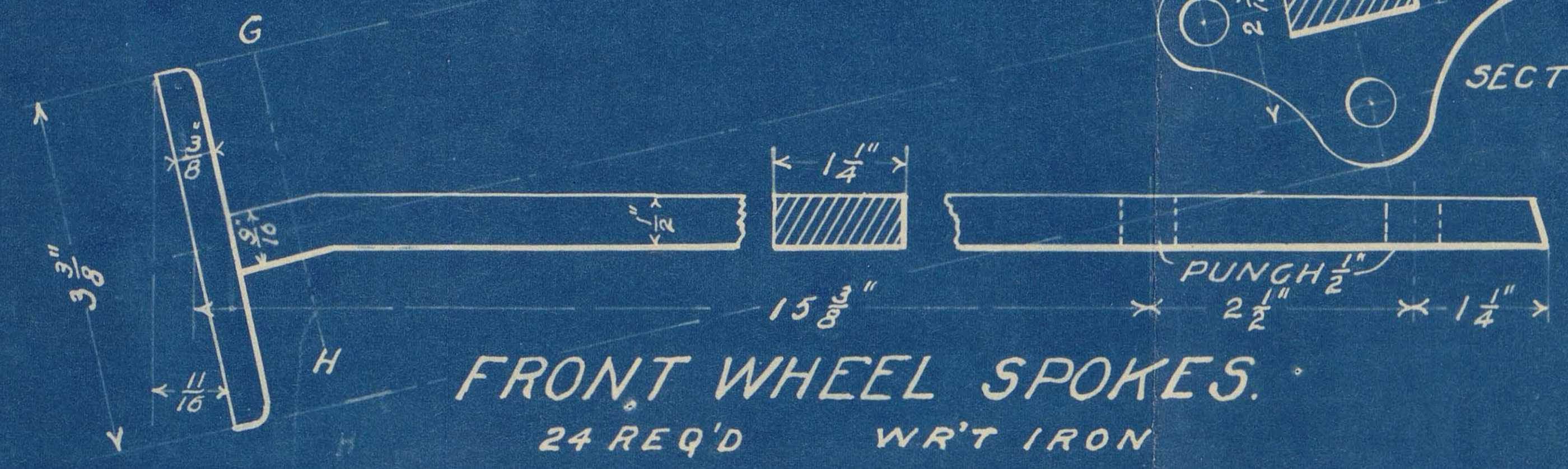
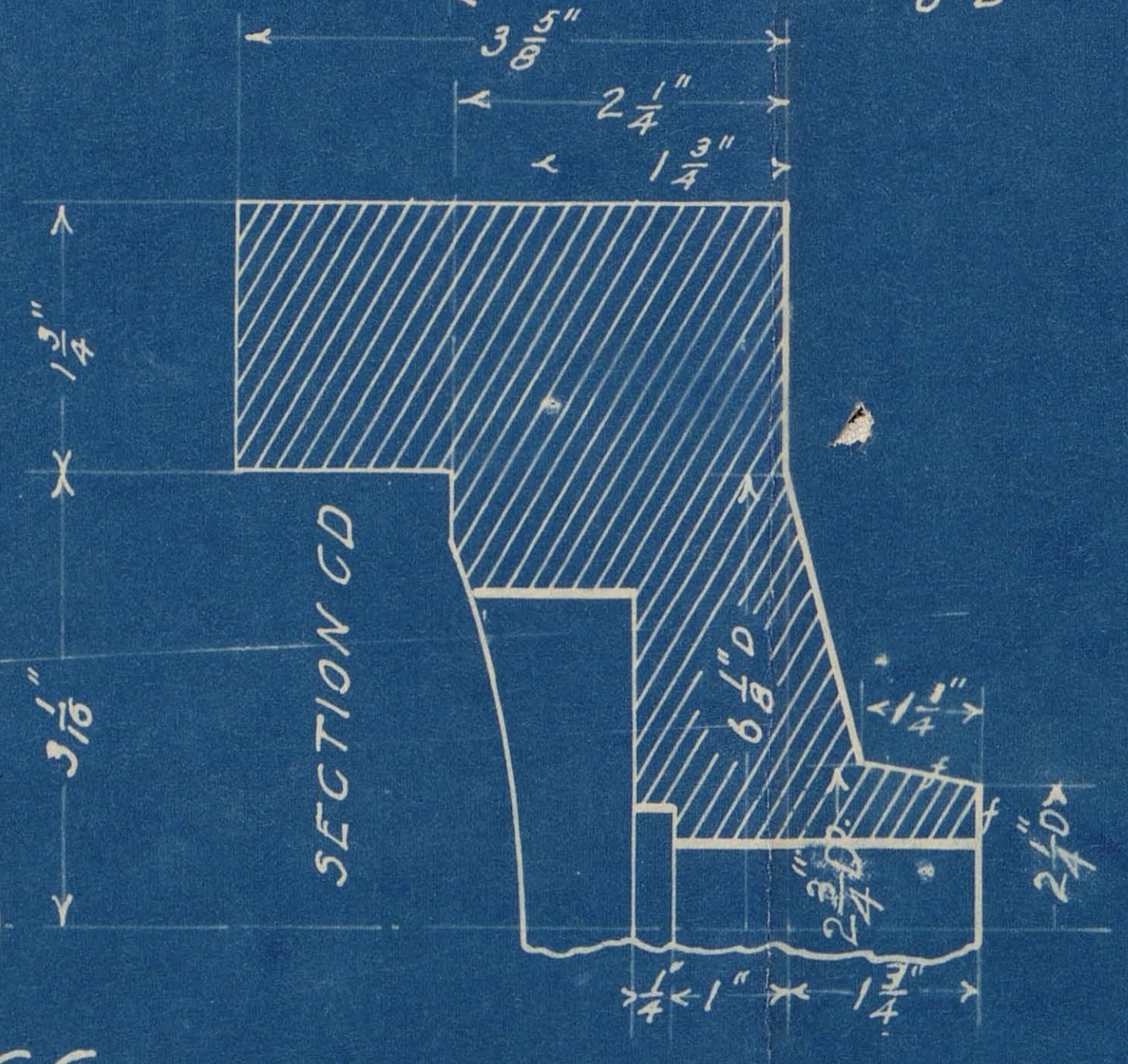




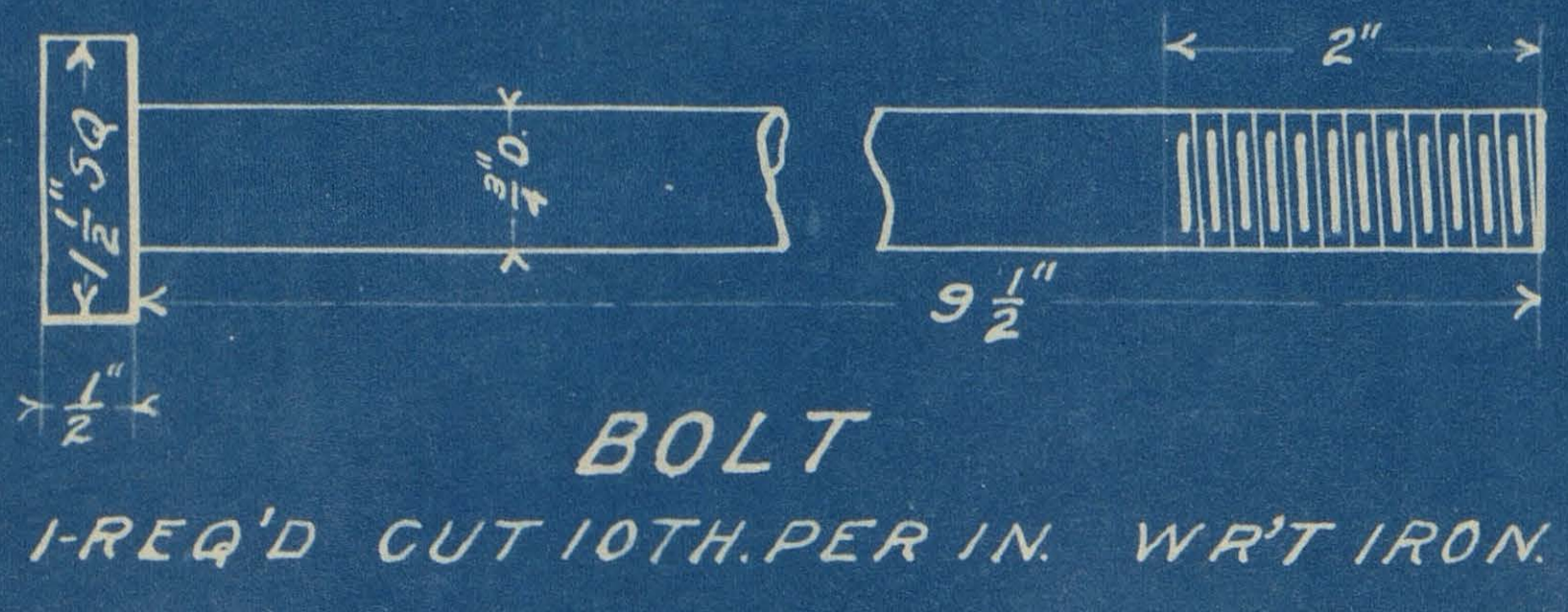
**Z 237**  
**FRONT WHEEL HUB**  
 2 REQ'D GRAY IRON



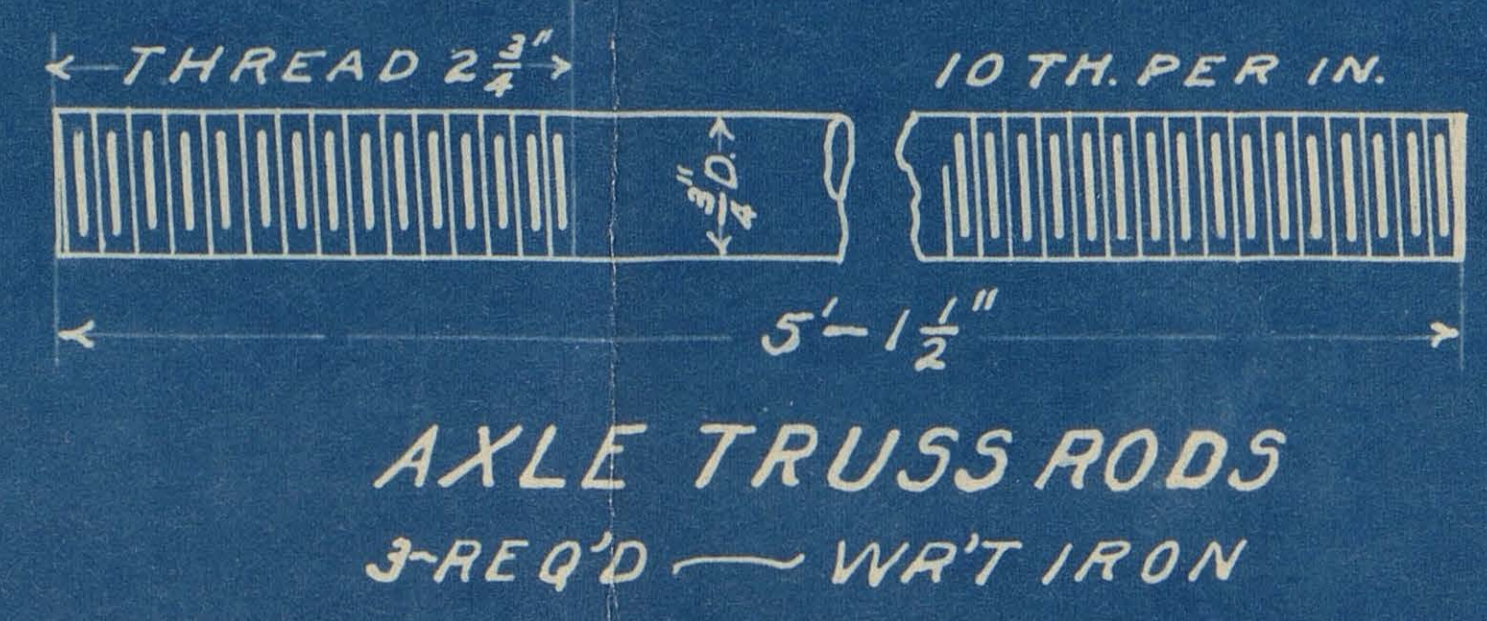
**Z 166**  
**FRONT AXLE SPRING CAP**  
 1 REQ'D GRAY IRON



**FRONT WHEEL SPOKES.**  
 24 REQ'D WRT IRON



**BOLT**  
 1-REQ'D CUT 10TH. PER IN. WRT IRON.



**AXLE TRUSS RODS**  
 3-REQ'D WRT IRON

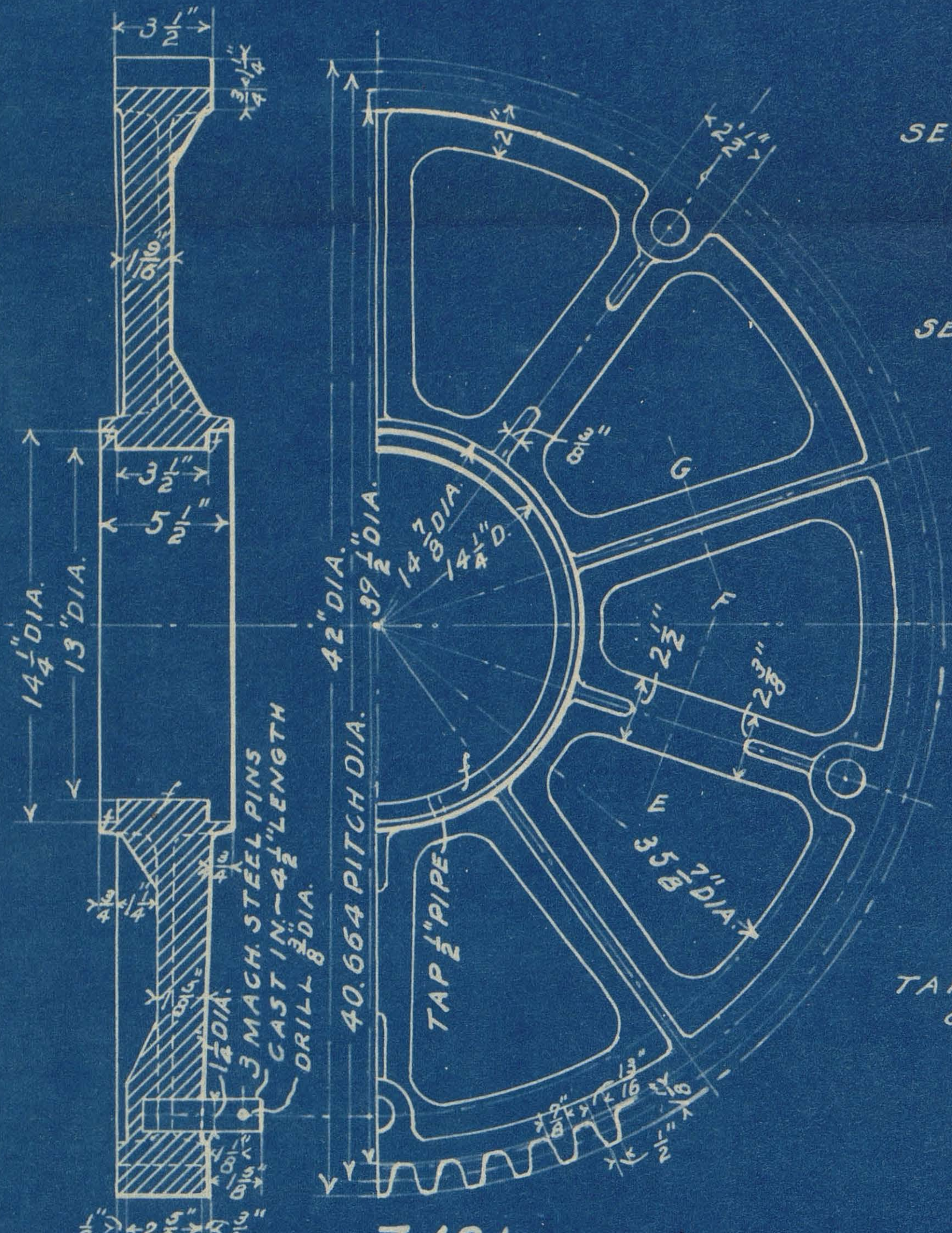
**DETAILS OF 25 HP**  
**GASOLINE TRACTION ENGINE**  
 SCALE: 6"=1 FT. f MEANS FINISH

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 DATE - 4-6-05



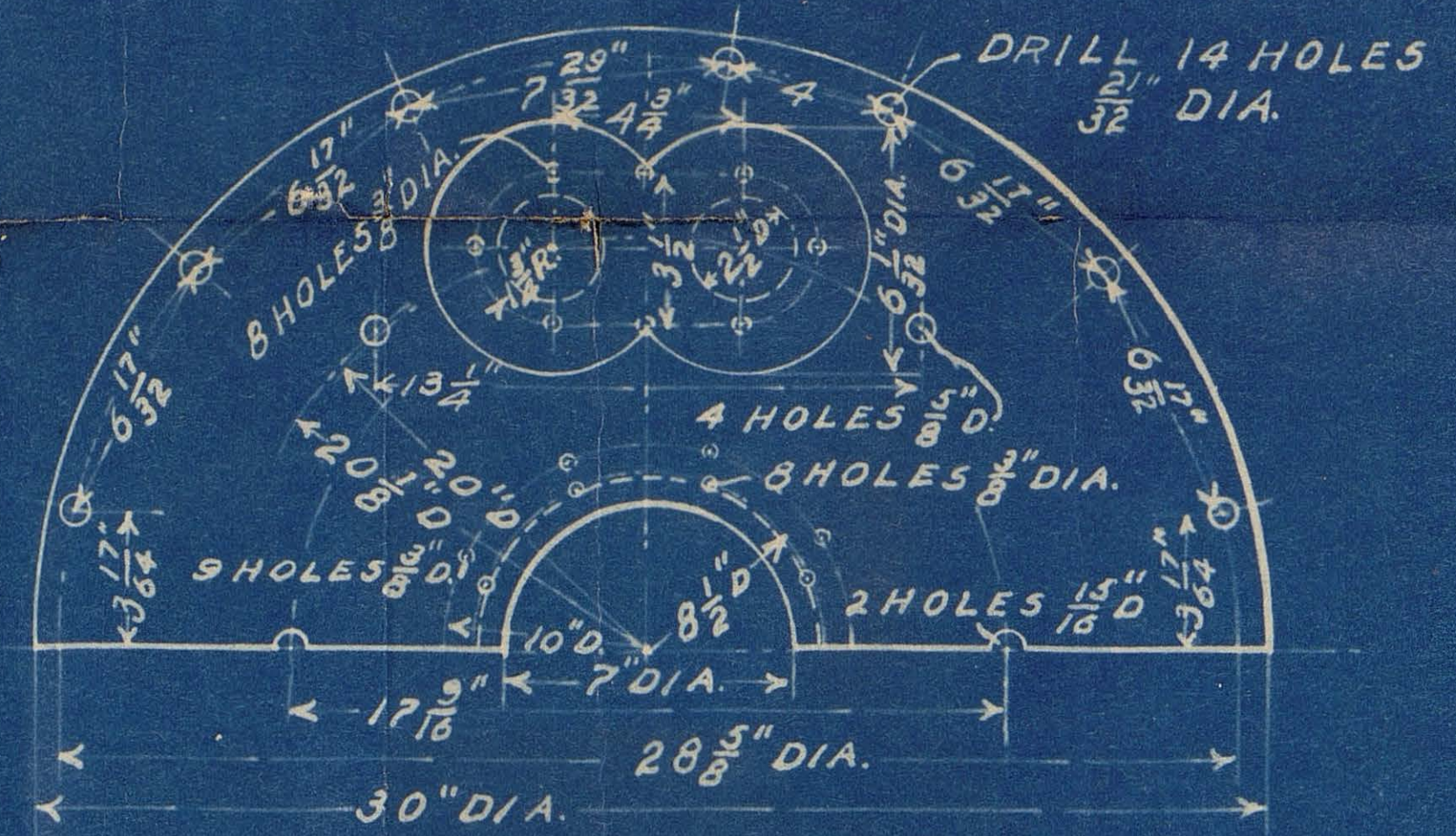
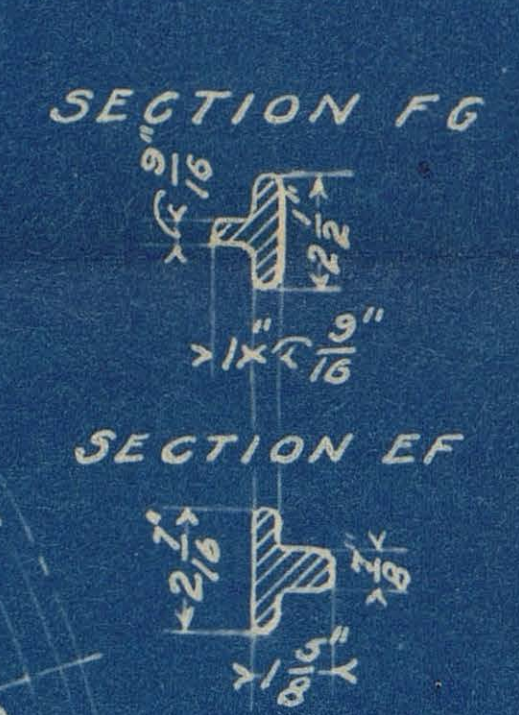




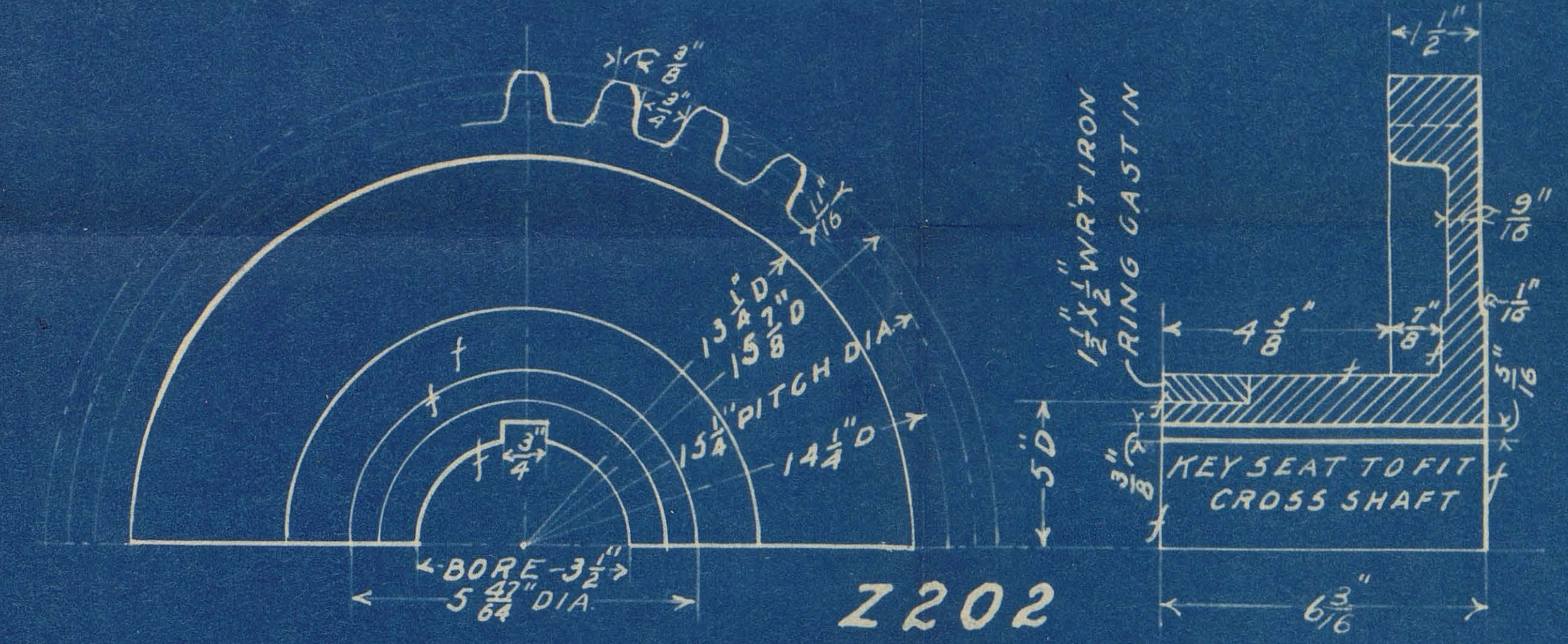


**Z191**  
**BULL WHEEL**

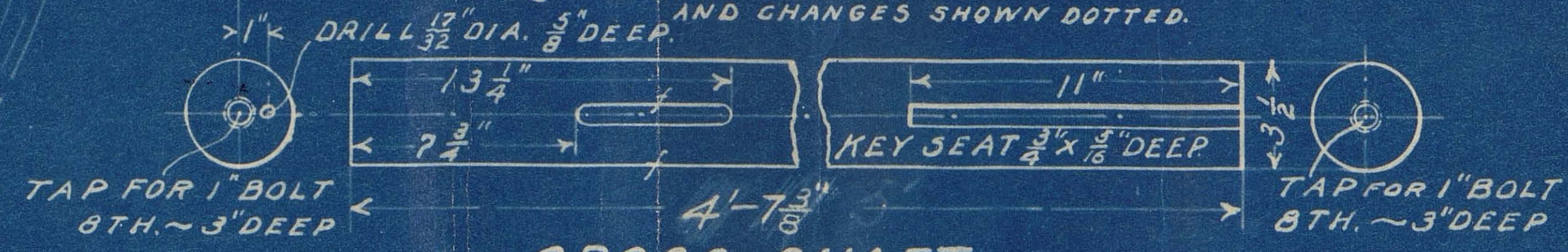
2 REQ'D ~ 73 TEETH ~ CAST STEEL  
CIRC. PITCH = 1.75" DIAMETRAL P. = .557  
NOTE: USE 52 ROLLERS 3 7/16" X 3/4" DIA.  
SCALE: 1 1/2" = 1 FT.



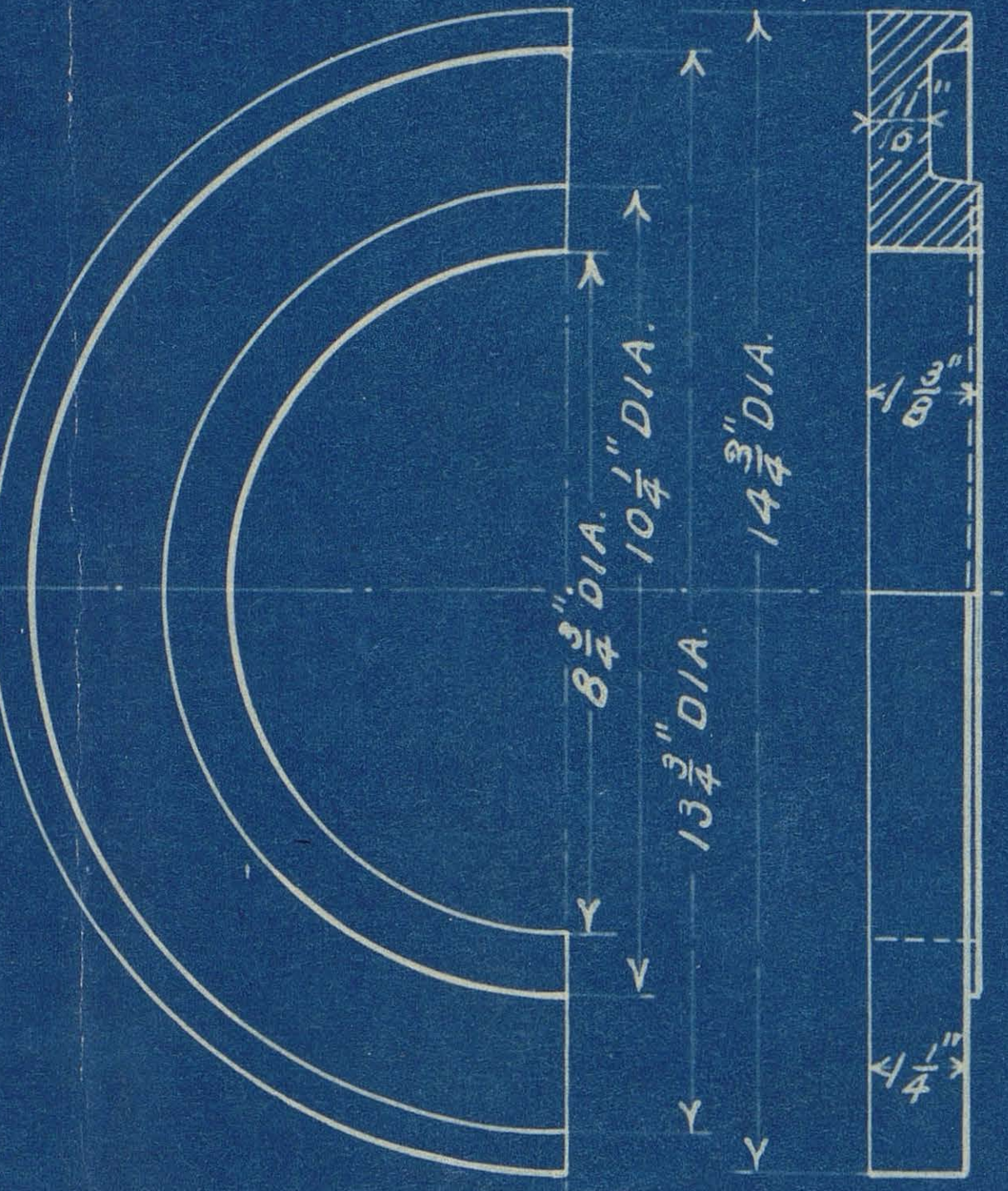
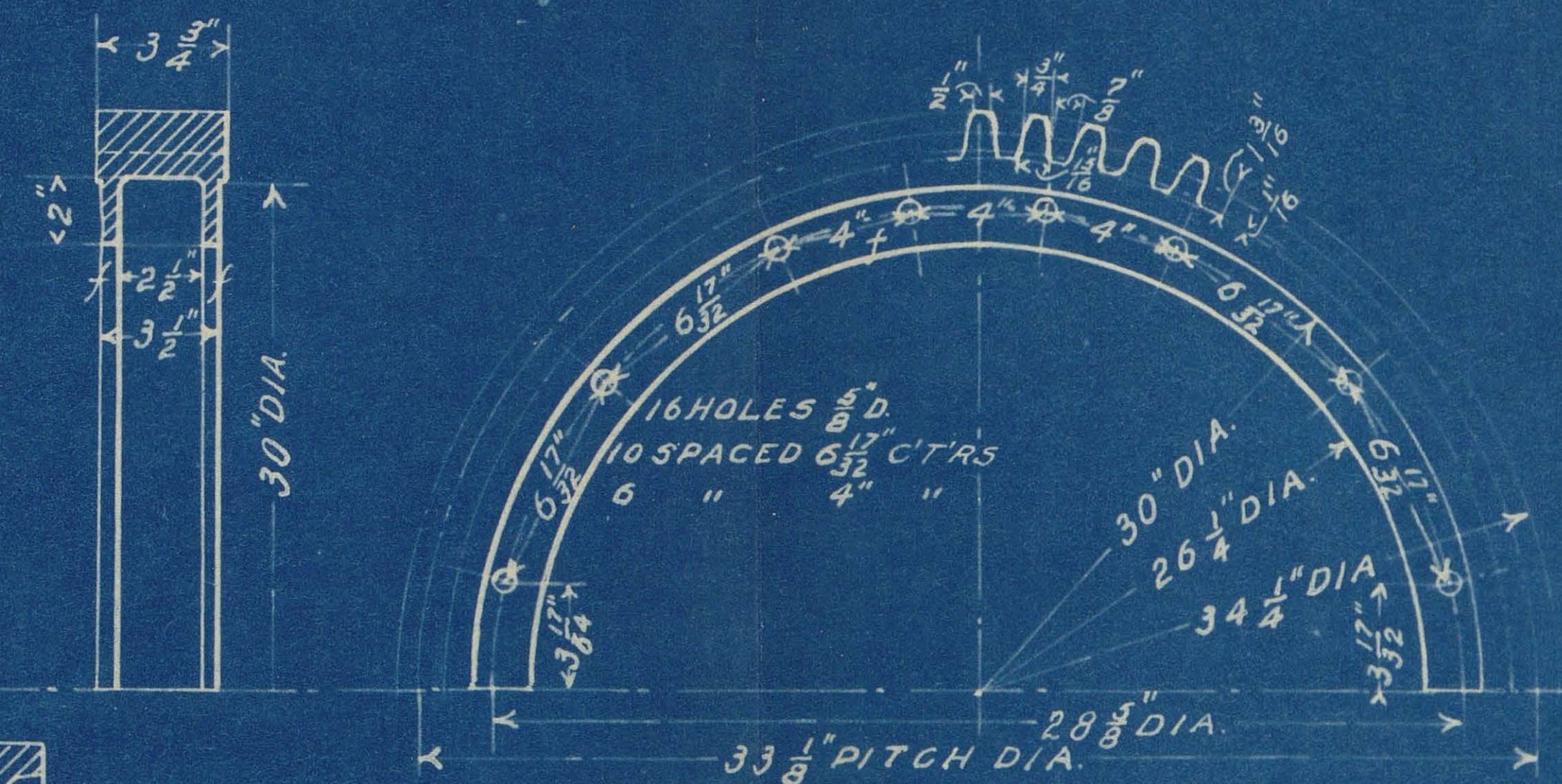
**Z202**  
**COMPENSATING GEAR**  
32 TEETH CIRC. PITCH = 1.50"  
1 REQ'D GRAY IRON  
SCALE - 3" = 1 FT.



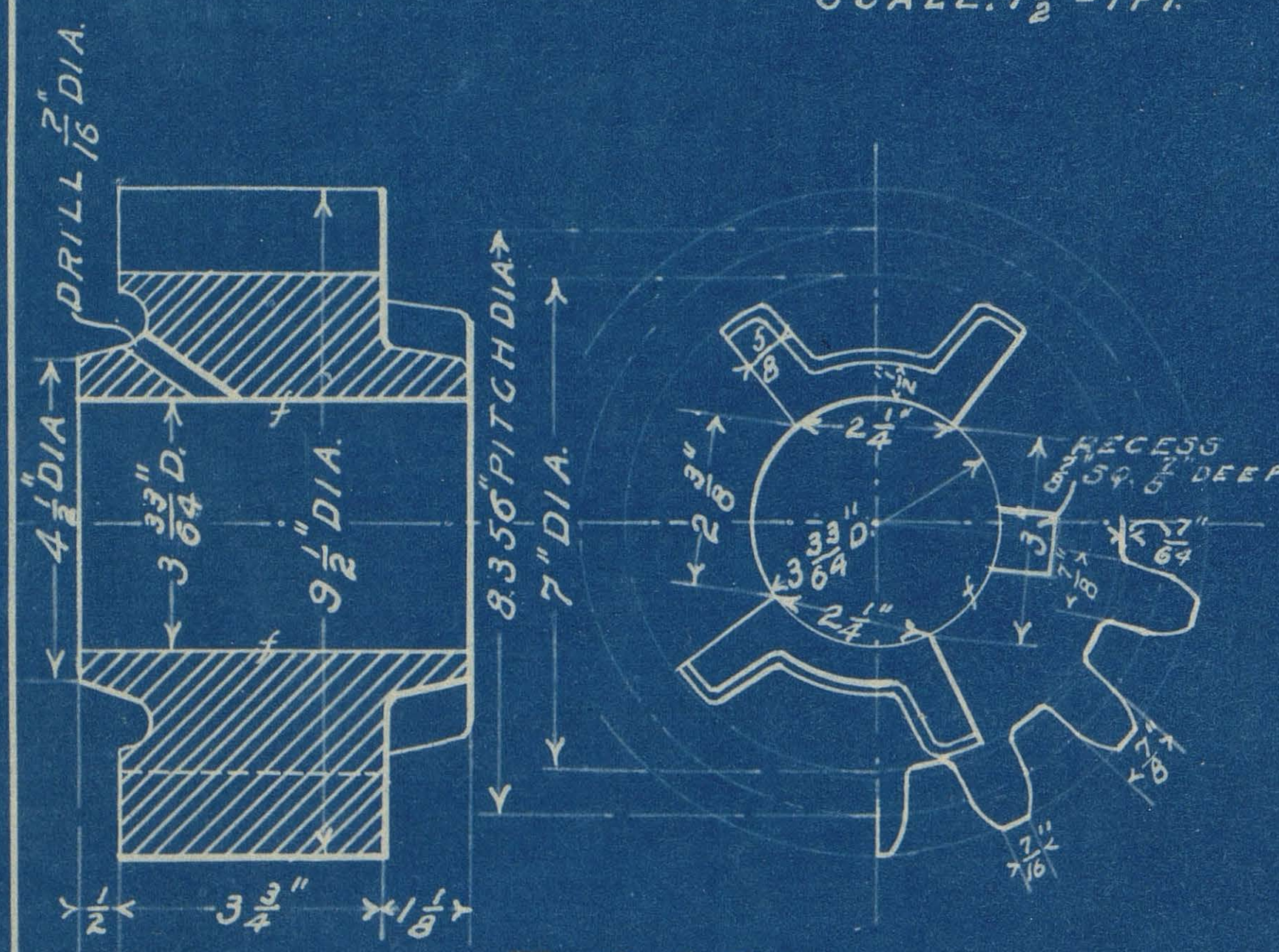
**Z6**  
**COMPENSATING GEAR SHELL WHEEL**  
1 REQ'D ~ 64 TEETH - CIRC. PITCH = 1.625" - CAST STEEL  
SCALE: 1 1/2" = 1 FT.



**Z193**  
**CROSS SHAFT**  
MILD STEEL f ALL OVER  
SCALE 1 1/2" = 1 FT.



**Z193**  
**INTERMEDIATE GEAR**  
1 REQ'D 63 TEETH CAST STEEL  
CIRCULAR PITCH = 1.602"  
SCALE 1 1/2" = 1 FT.  
NOTE: USE 47 ROLLERS 3 3/8" X 5/8" DIA.



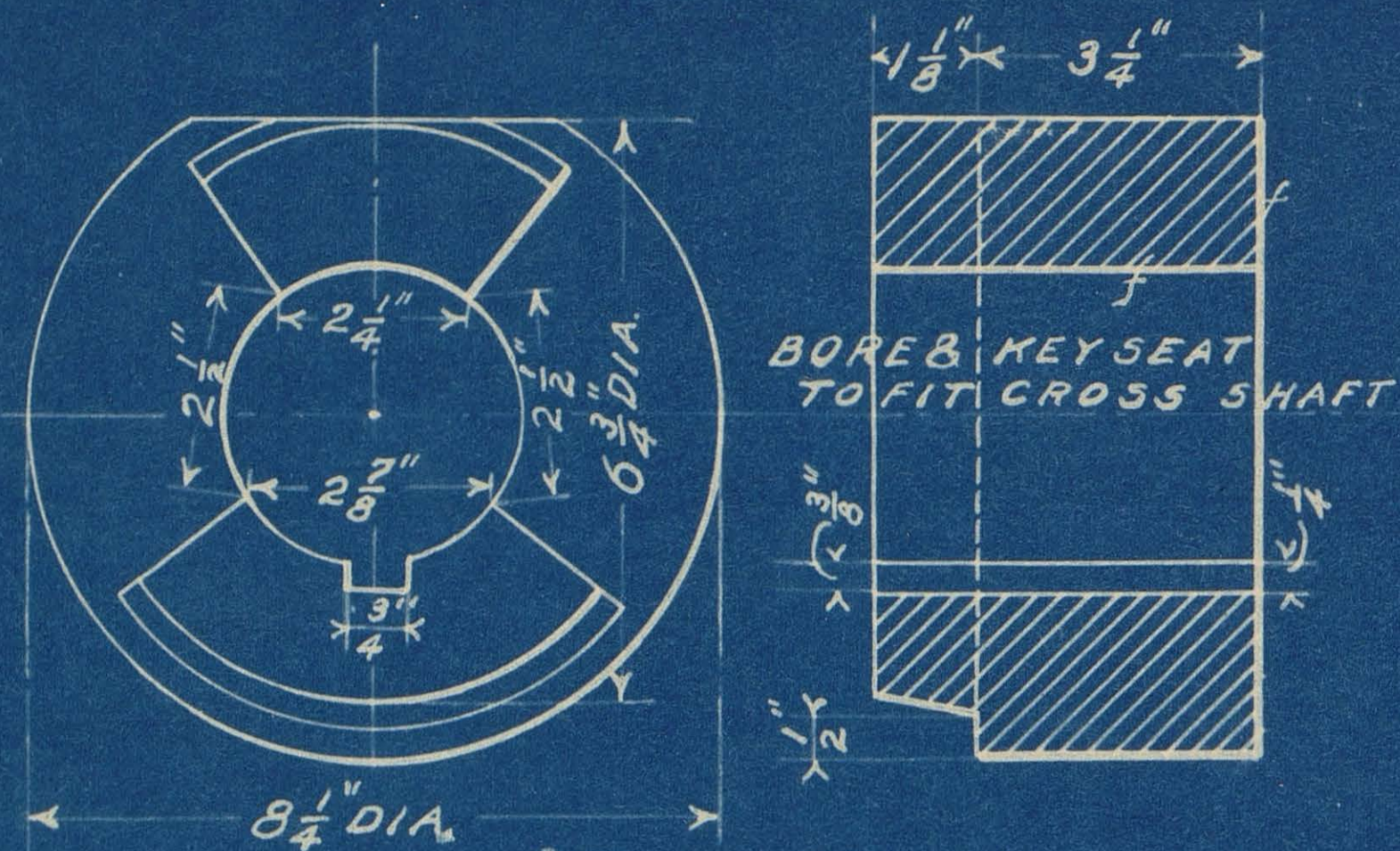
**Z229**  
**BULL WHEEL PINION**  
2 REQ'D ~ 16 TEETH ~ CAST STEEL  
CIRC. PITCH = 1.75" DIAMETRAL PITCH = .557  
SCALE: 3" = 1 FT.

**DETAILS OF 25 HP**  
**GASOLINE TRACTION ENGINE**

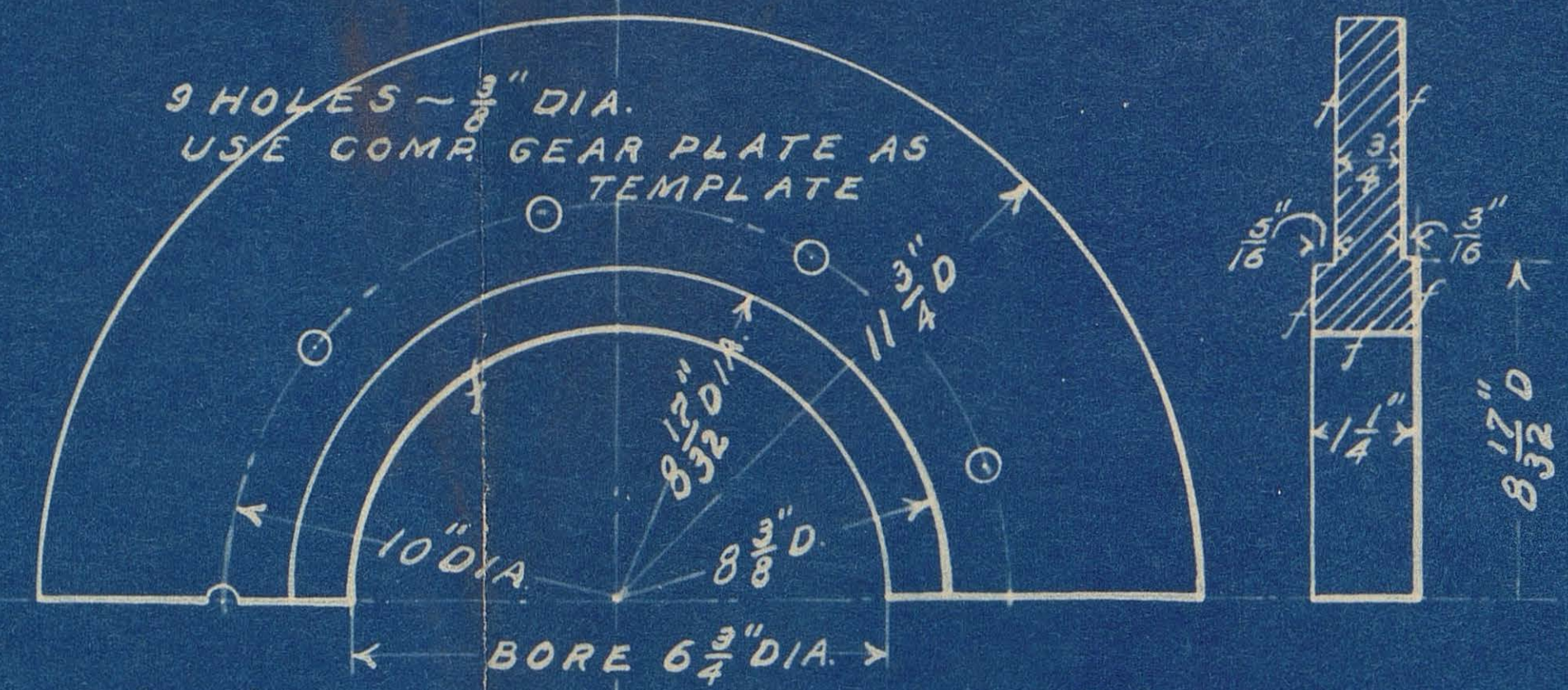
SCALE: - GIVEN f MEANS FINISH  
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DATE - 4-17-05

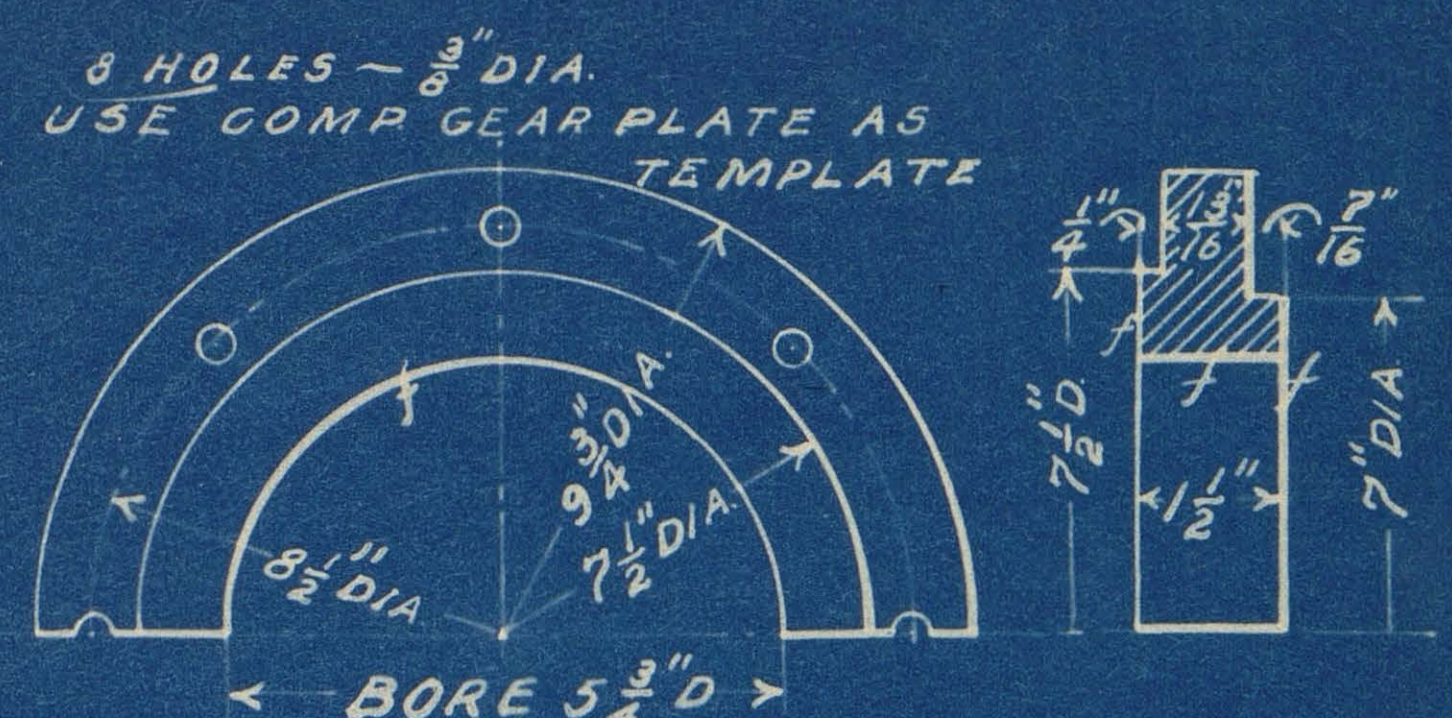




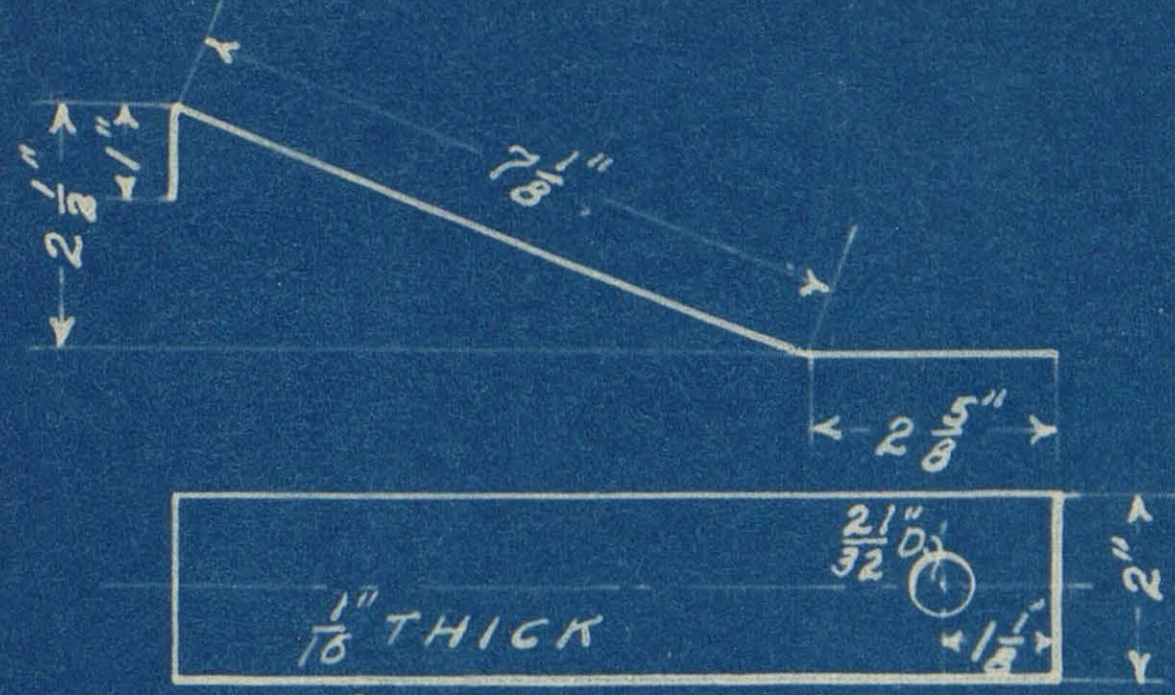
**Z 217**  
**CLUTCH COLLAR**  
 FOR BULL WHEEL PINION  
 1 REQ'D ~ GRAY IRON



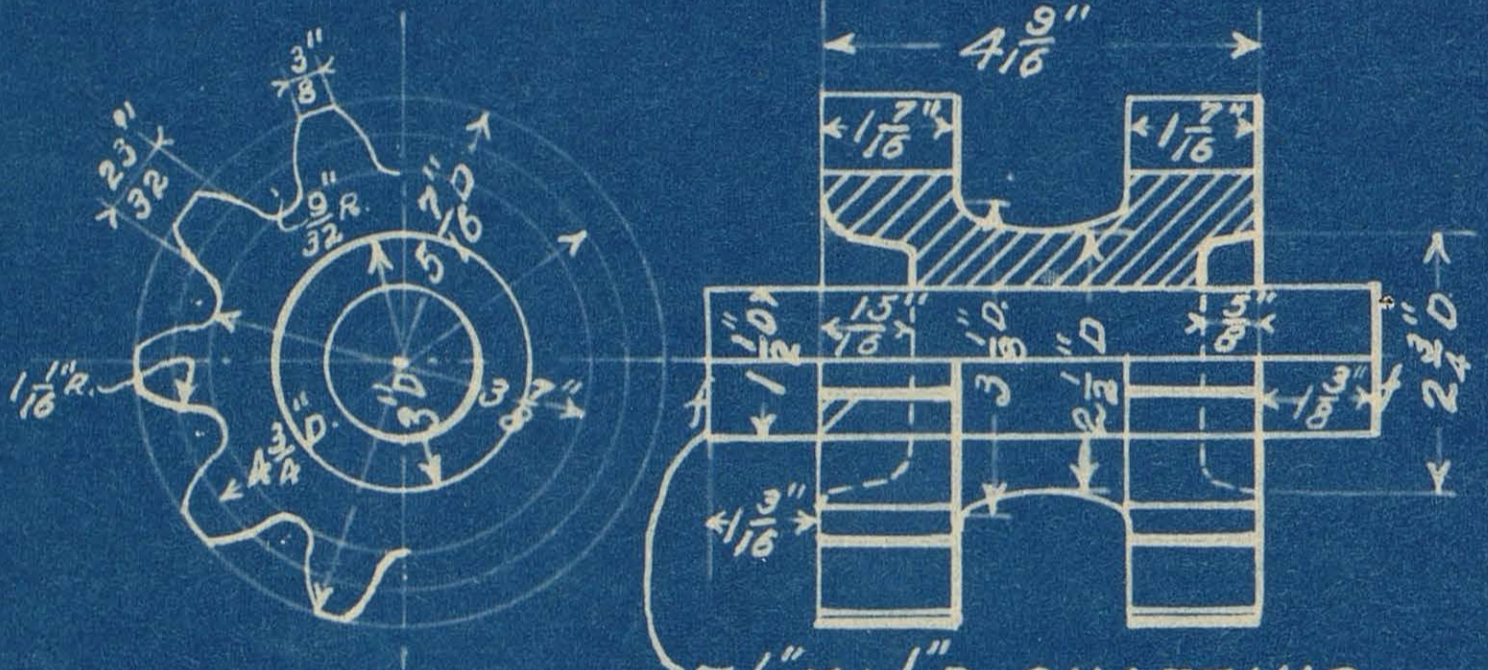
**Z 223**  
**COMP. GEAR HUB FLANGE**  
 1 REQ'D ~ GRAY IRON



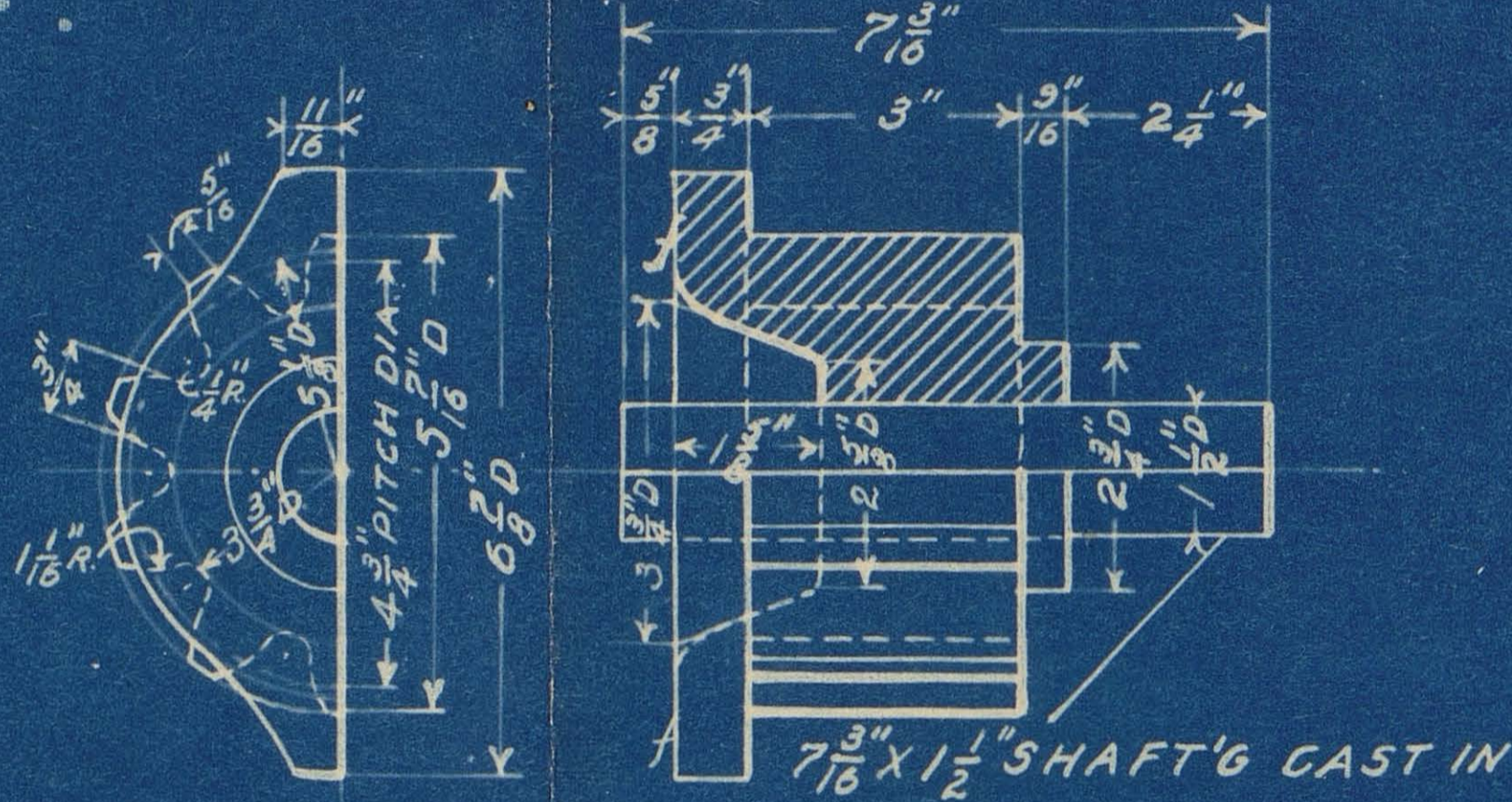
**Z 222**  
**COMP. GEAR HUB FLANGE**  
 1 REQ'D ~ GRAY IRON



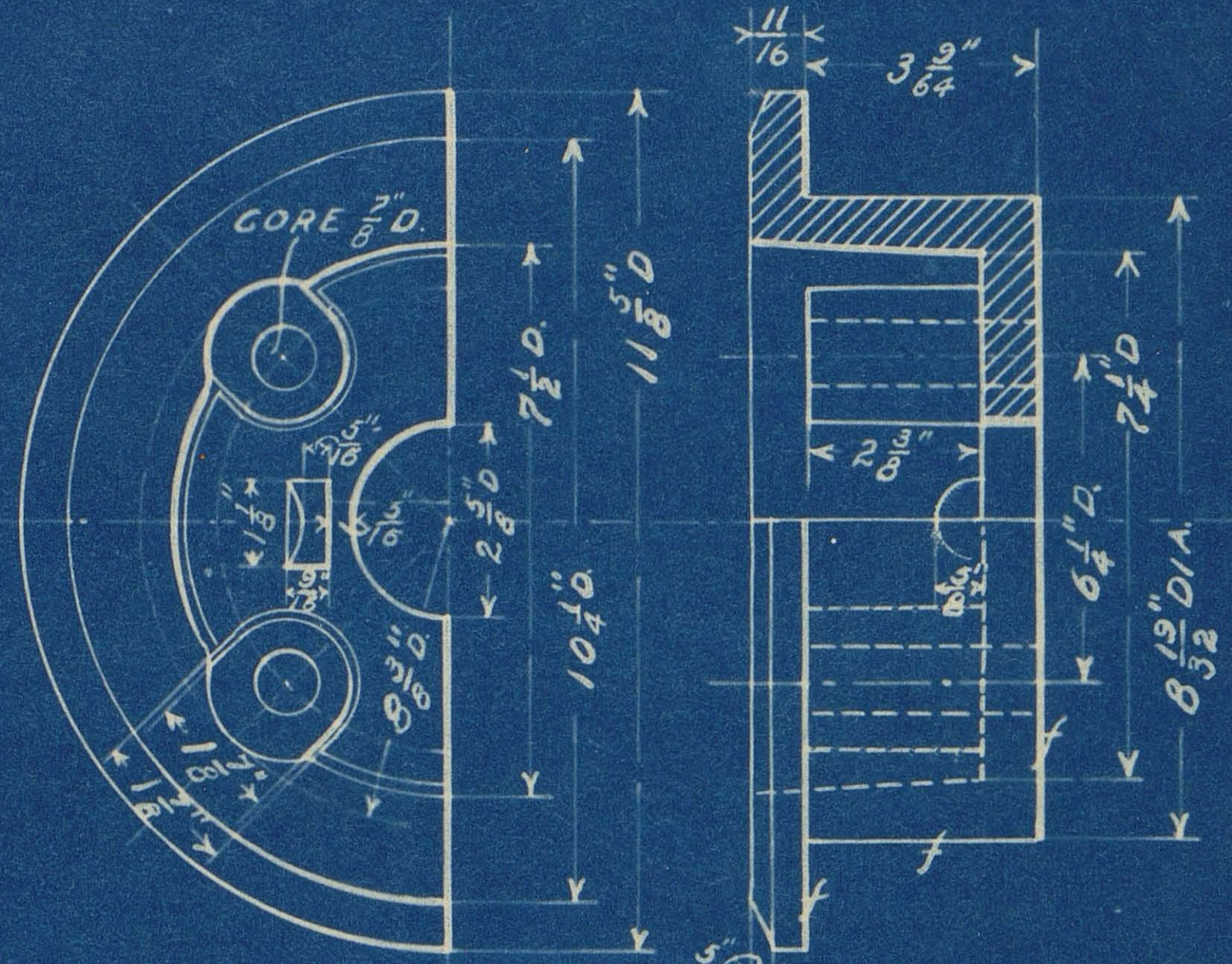
**Z 545**  
**COMP. GEAR SPRING**  
 4 REQ'D ~ S. STEEL



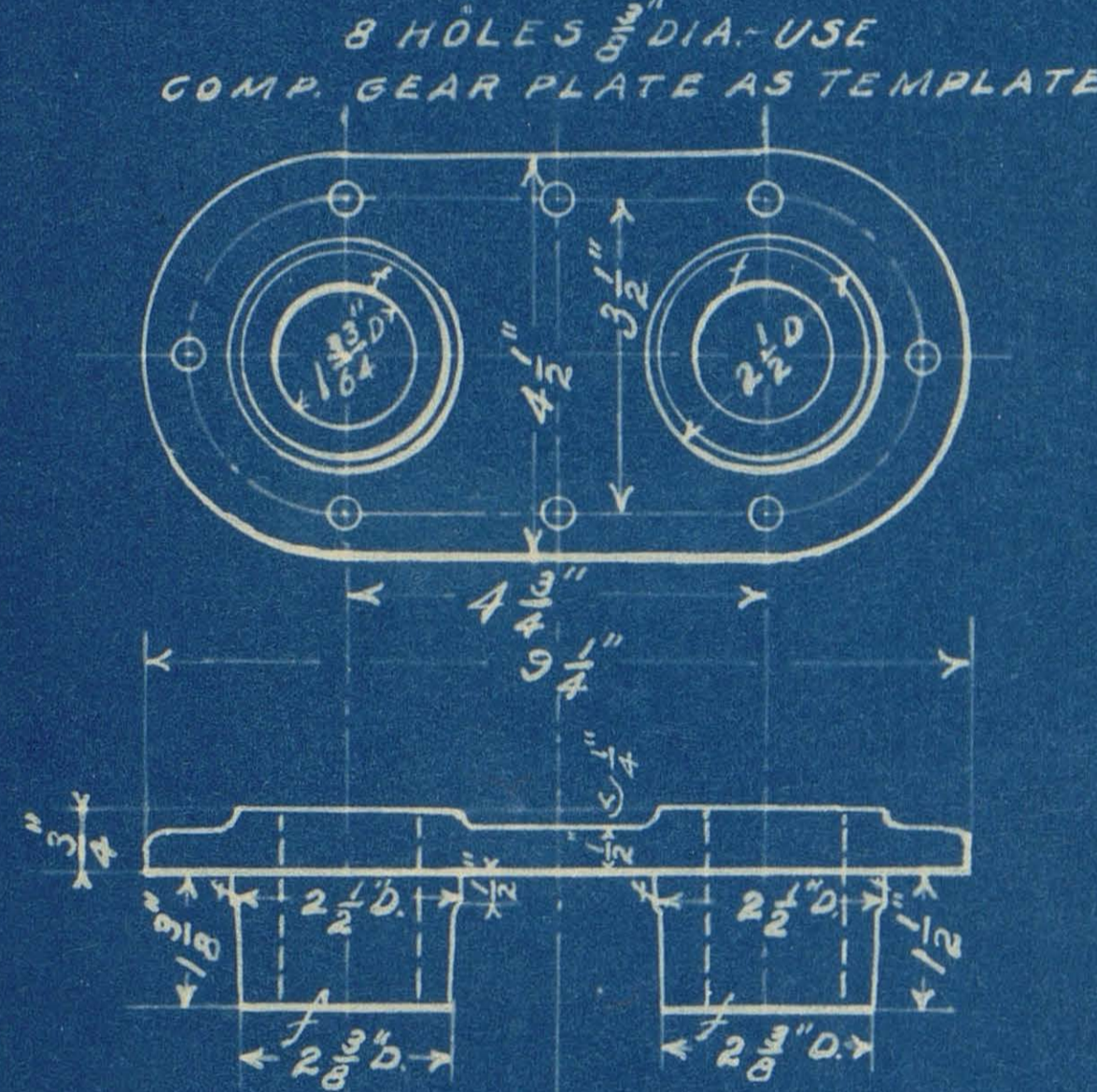
**Z 218**  
**COMP. GEAR DOUBLE PINION**  
 2 REQ'D ~ 10 TEETH ~ GRAY IRON  
 CIRC. PITCH = 1 1/2" ~ DIA. PITCH = .475



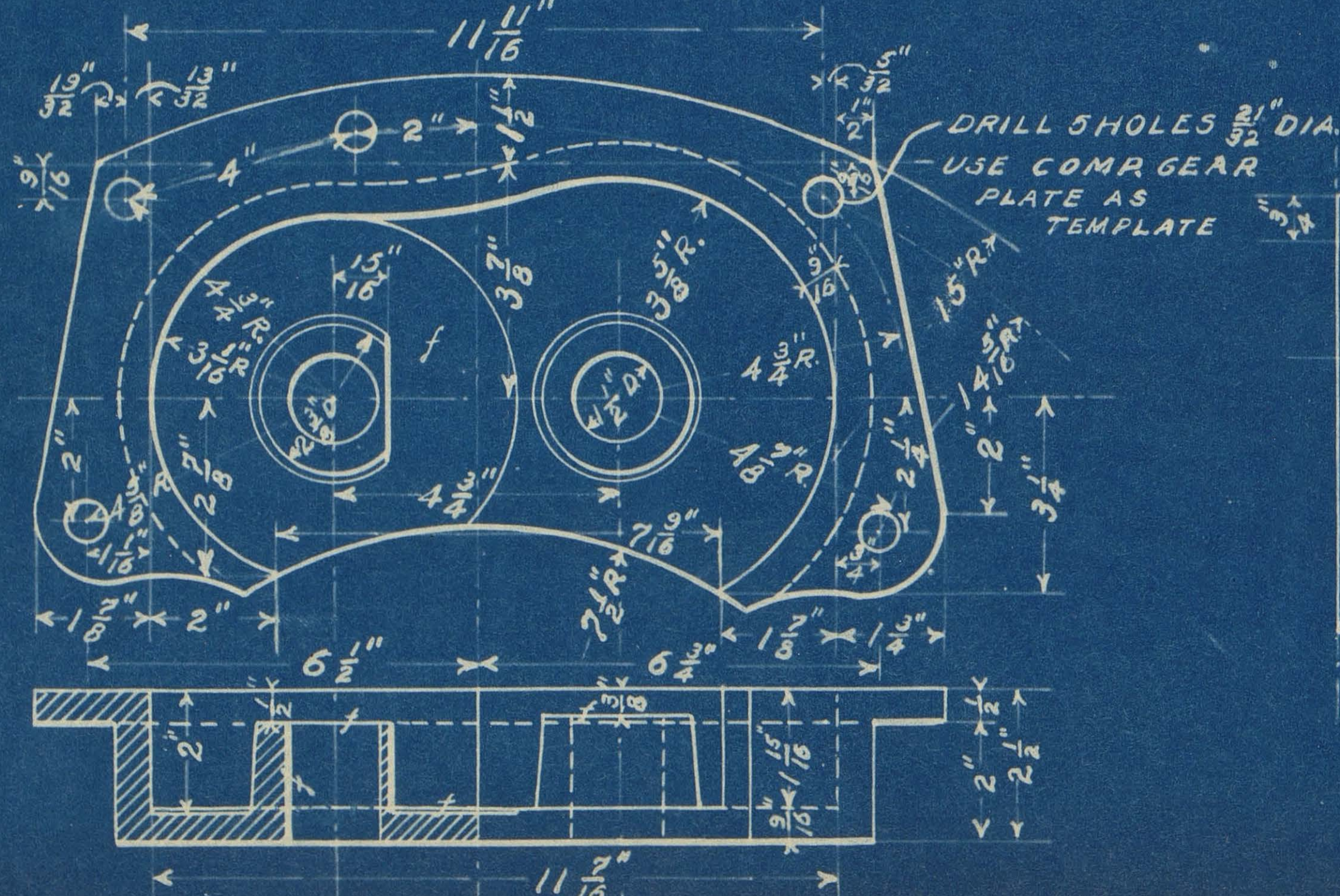
**Z 219**  
**COMP. GEAR LOCK PINION**  
 2 REQ'D ~ 10 TEETH ~ GRAY IRON  
 CIRC. PITCH = 1.5" ~ DIA. PITCH = .475



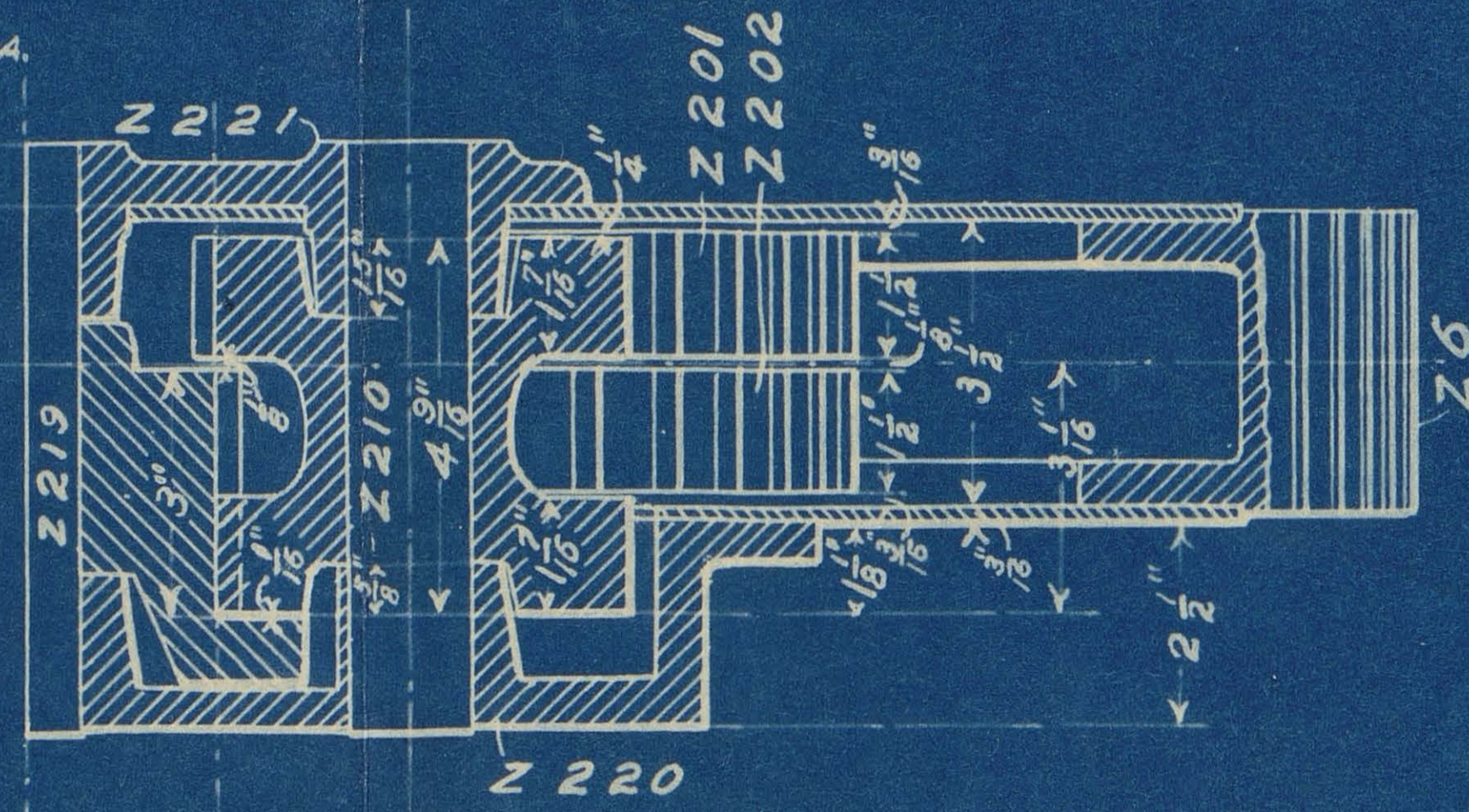
**Z 228**  
**INTER. GEAR TRUNION**  
 1 REQ'D ~ GRAY IRON



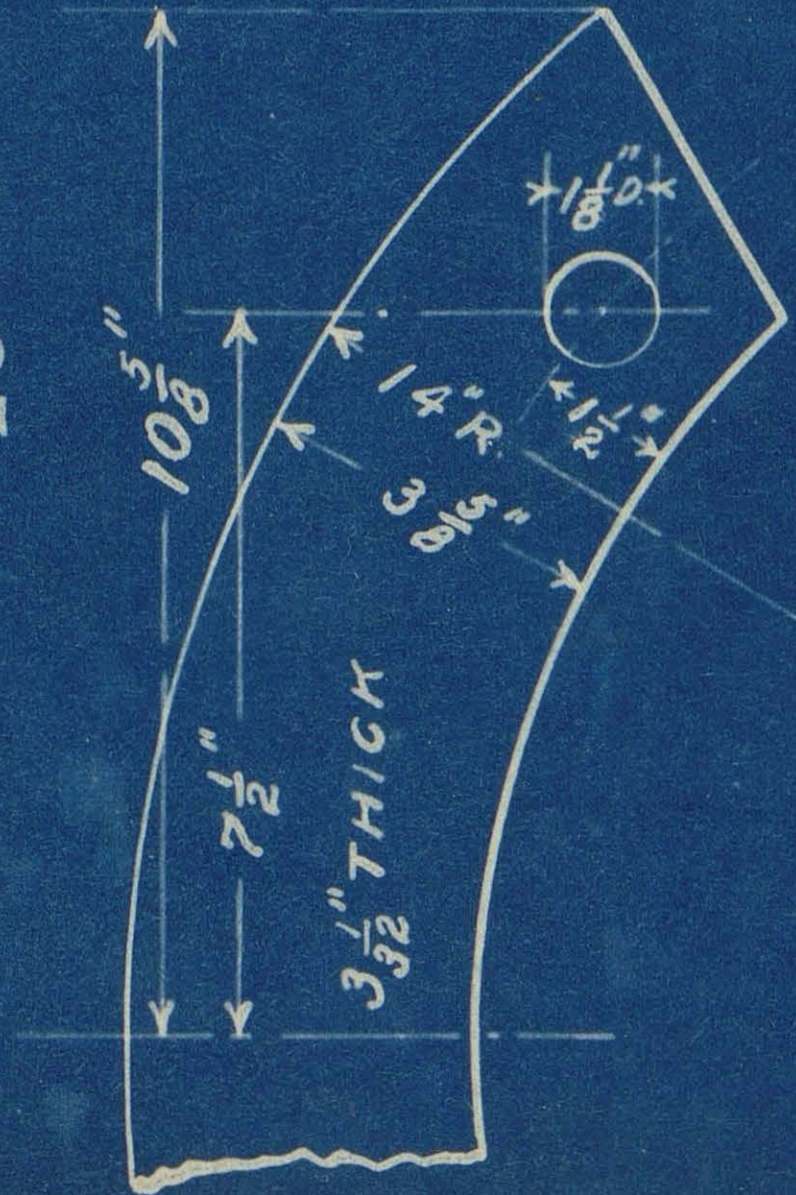
**Z 221**  
**COMP. GEAR BOX**  
 2 REQ'D ~ GRAY IRON



**Z 220**  
**COMP. GEAR BRACKET**  
 2 REQ'D ~ GRAY IRON



**Z 220**  
 PARTIAL SECTION THROUGH  
 COMPENSATING GEAR



**Z 220**  
 WOOD BLOCK  
 INSIDE COMP. GEAR  
 2 REQ'D ~ WHITE ELM

**DETAILS OF 25 HP  
 GASOLINE TRACTION ENGINE**

SCALE: - 3" = 1 FT. f MEANS FINISH

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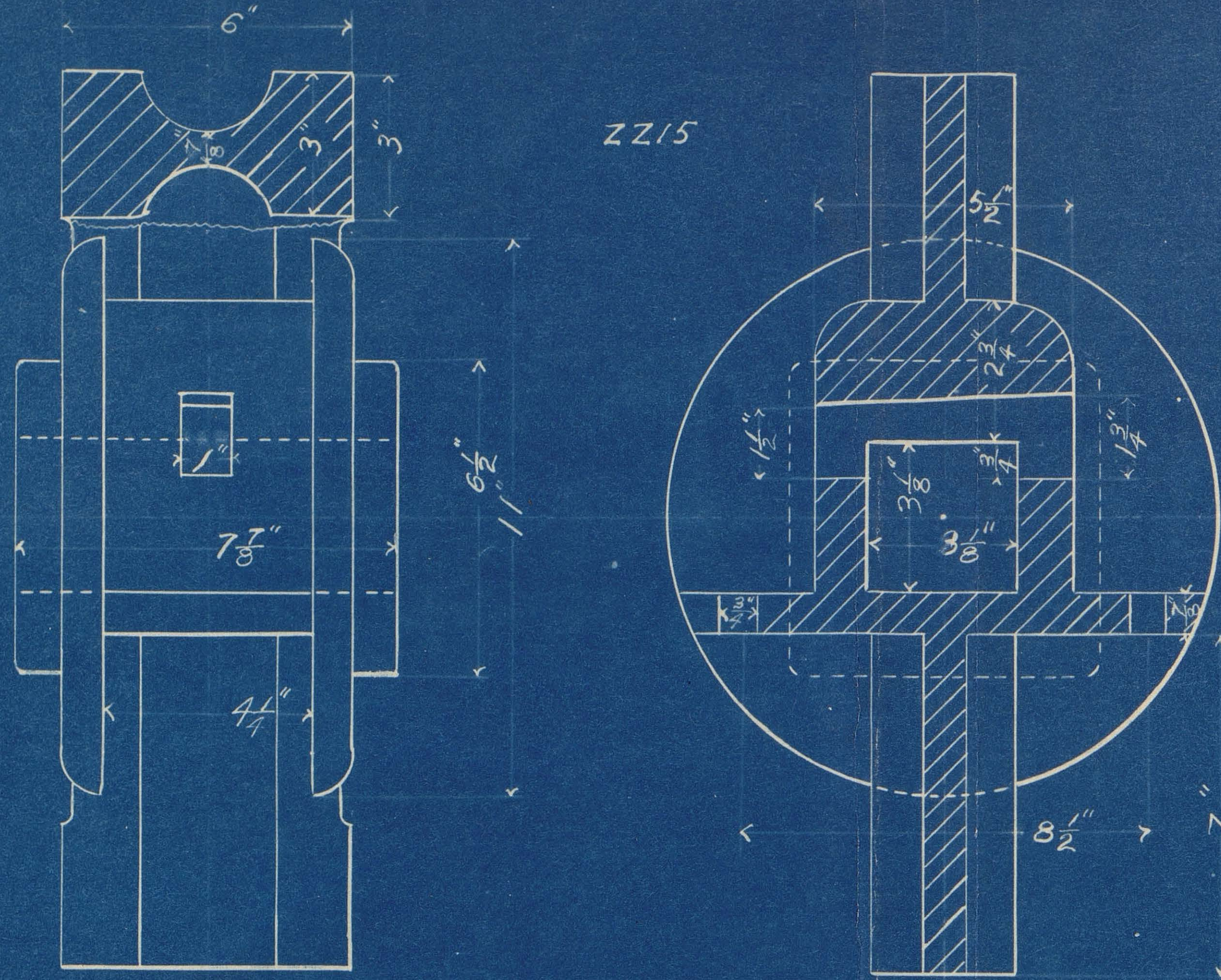
**K.S.A.C.**

MANHATTAN ~ KANSAS  
 DRAWN BY W.J. BRANT TRACED BY W.J.B.  
 DATE ~ 5-1-05

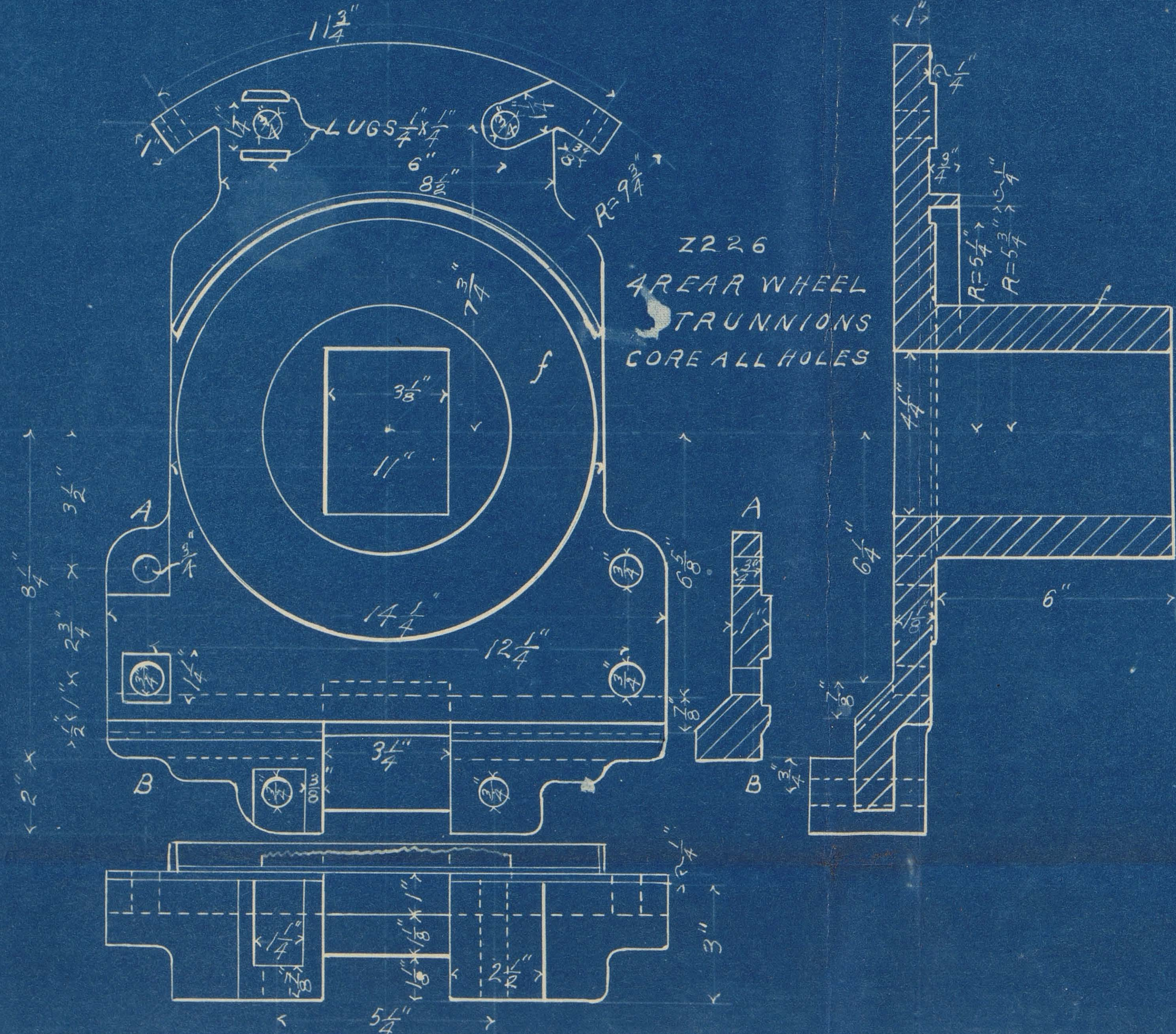




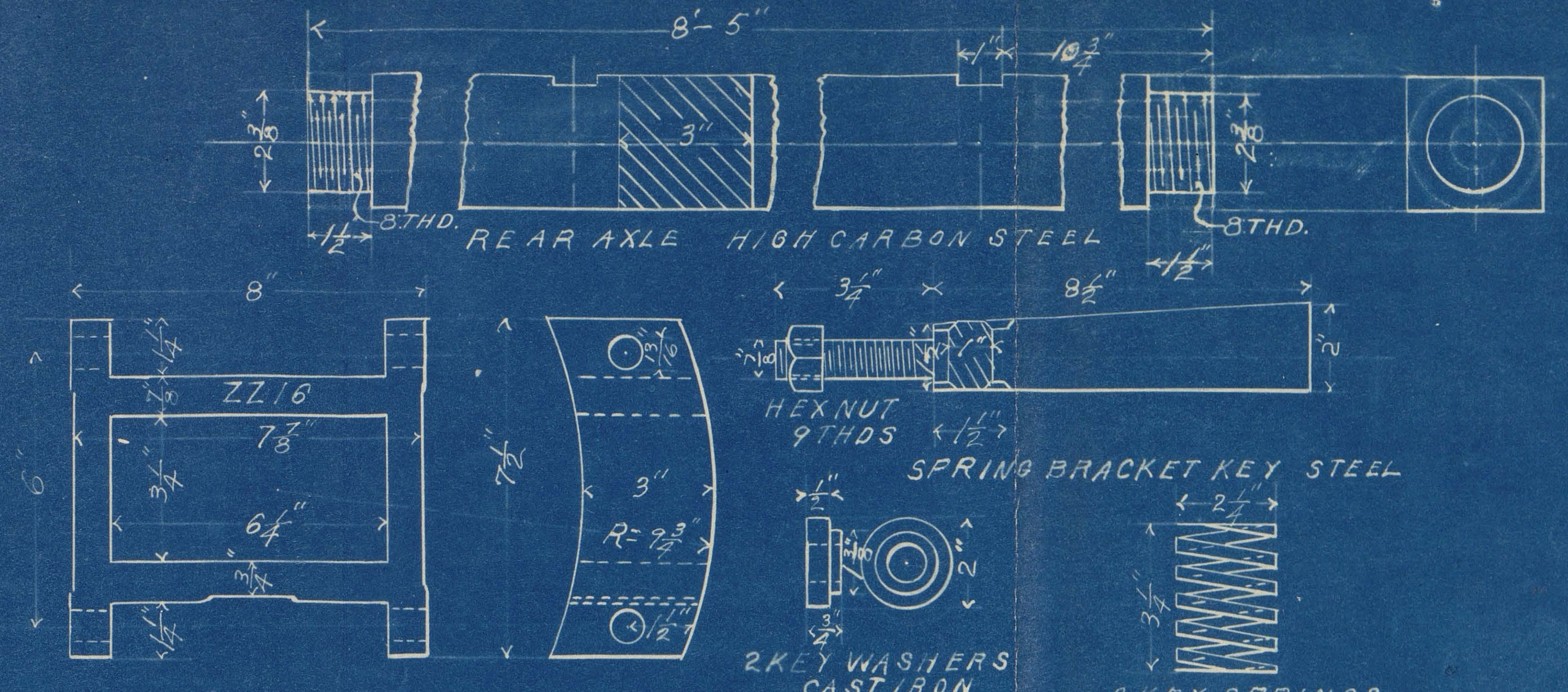




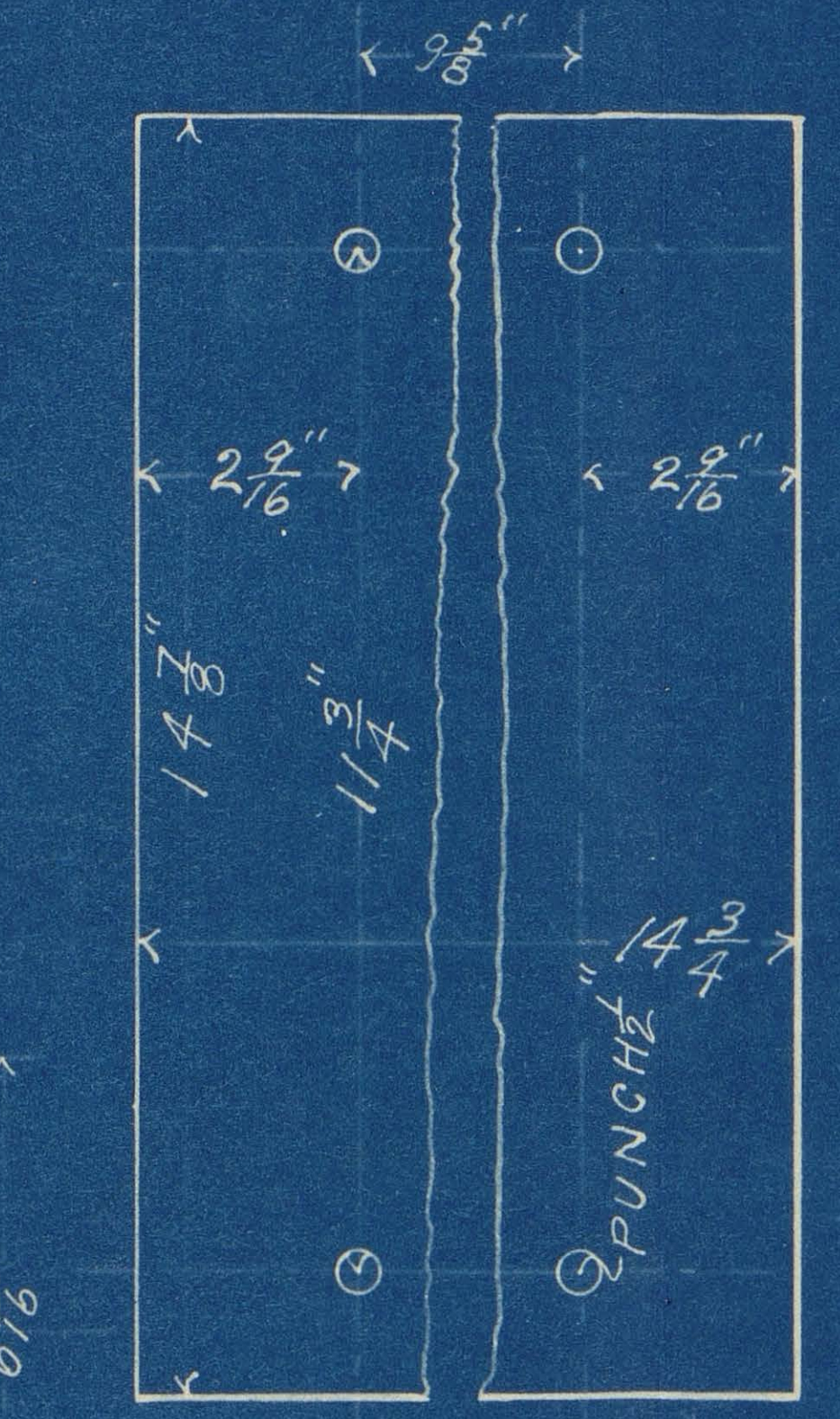
2 REAR WHEEL SPRING BRACKET CASTG



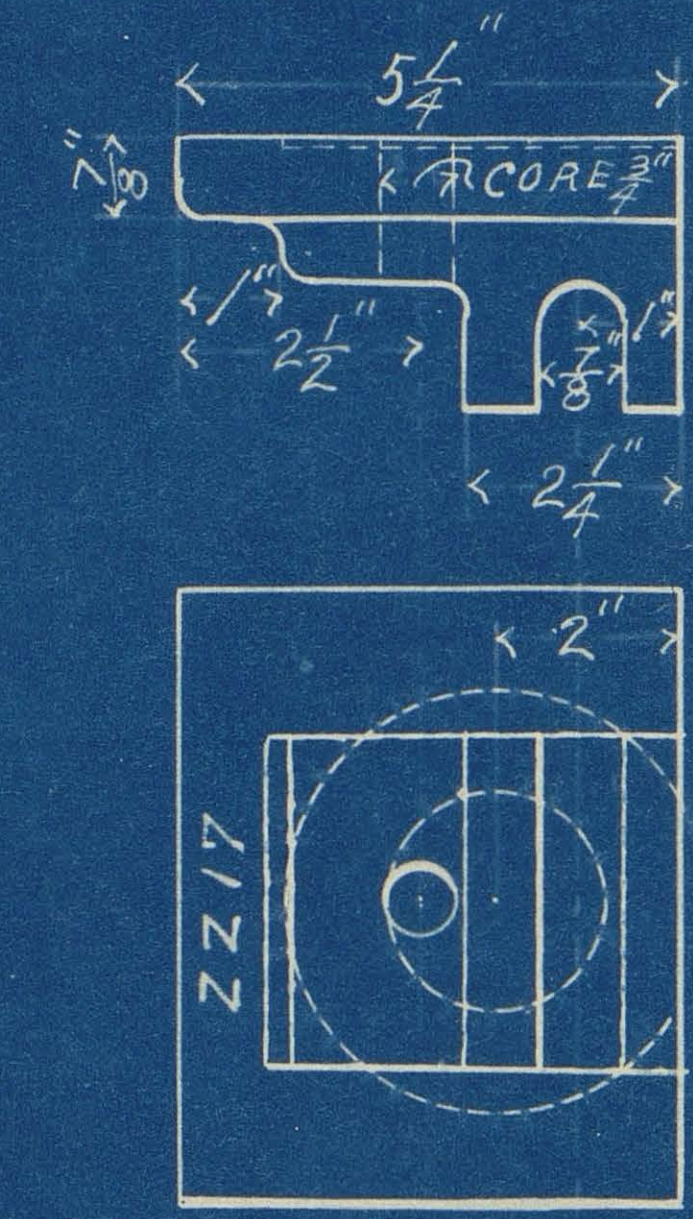
ZZ26  
4 REAR WHEEL  
TRUNNIONS  
CORE ALL HOLES



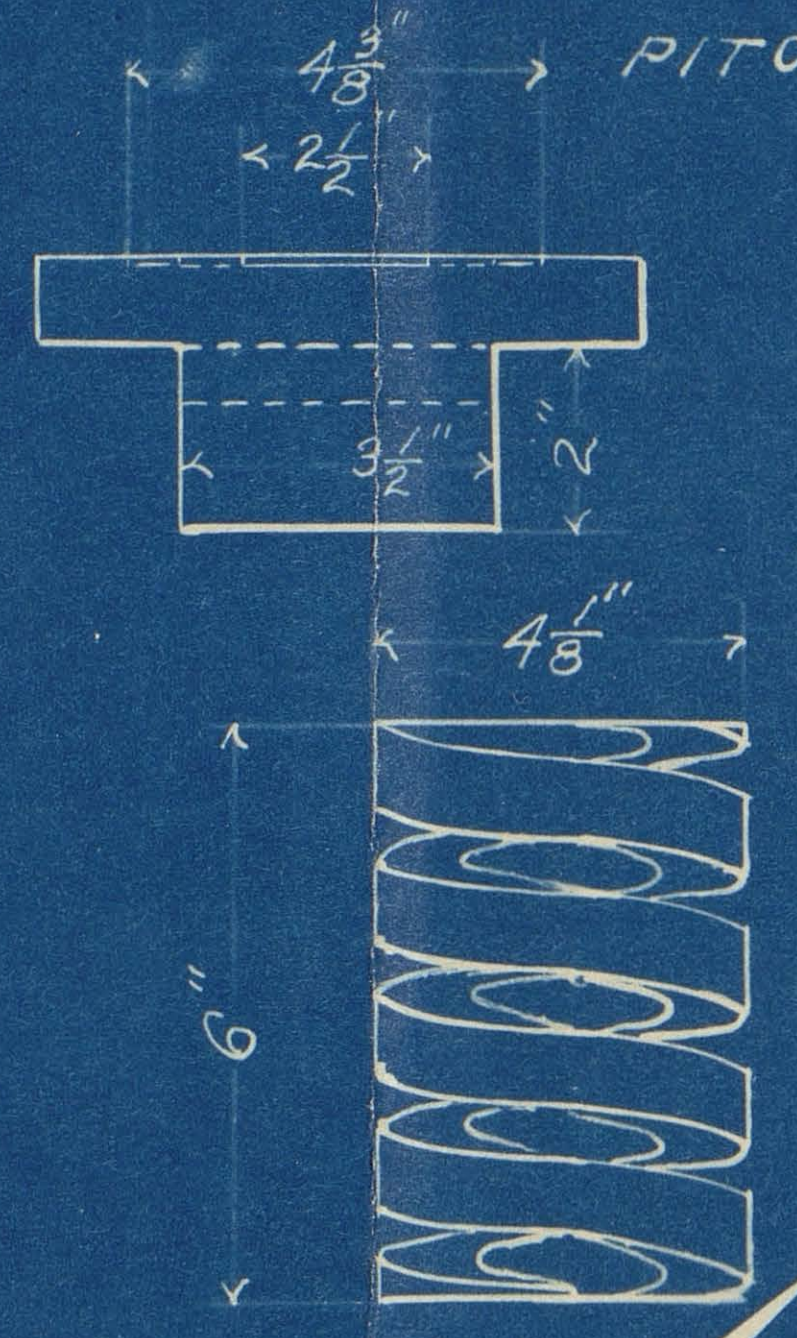
2 R.W. SPRING BRACKET GUIDES  
DRILL HOLES



2 TRUNNION SHIELDS  
1/16 SHEET STEEL



4 REAR WHEEL  
SPRING RESTS



4 REAR WHEEL SPRINGS  
STOCK 7/8 SQ. PITCH=1/8

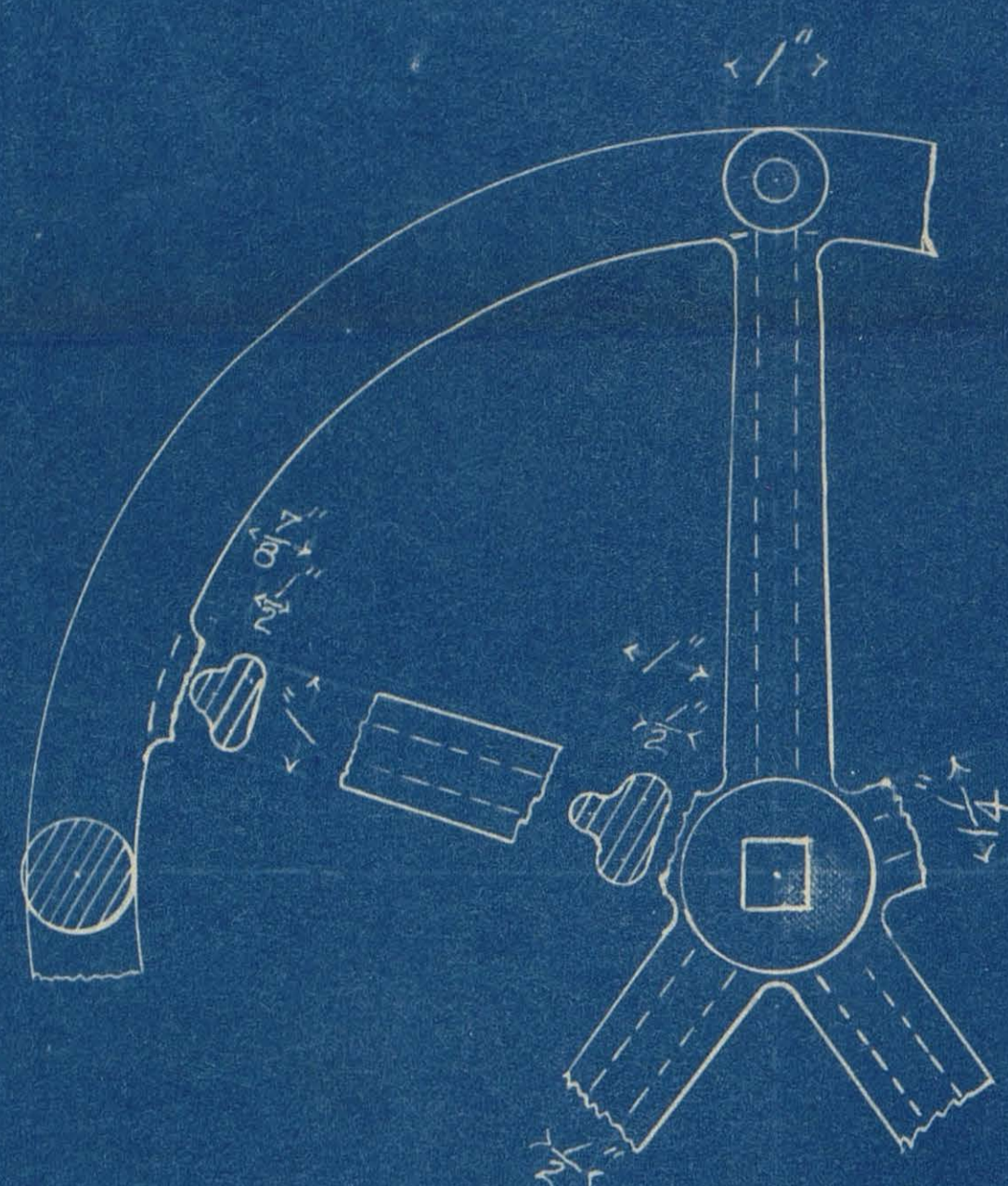
DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: 3"=1 FT. f MEANS FINISH

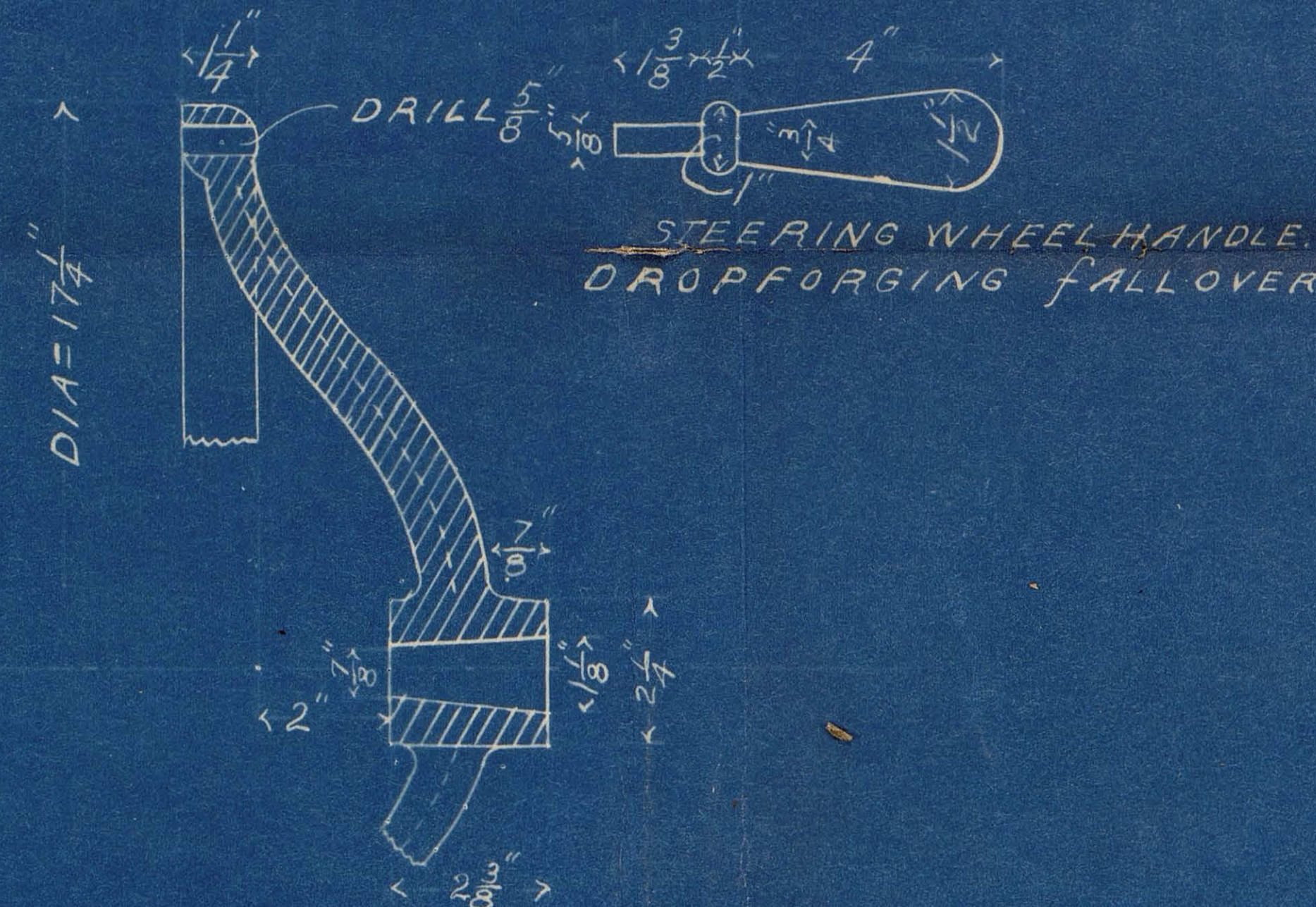
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MECHANICAL ENGINEERING  
K.S.A.C.

MANHATTAN — KANSAS  
DRAWN BY E.D.R. TRACED BY E.D.R.  
DATE - 4-5-05.

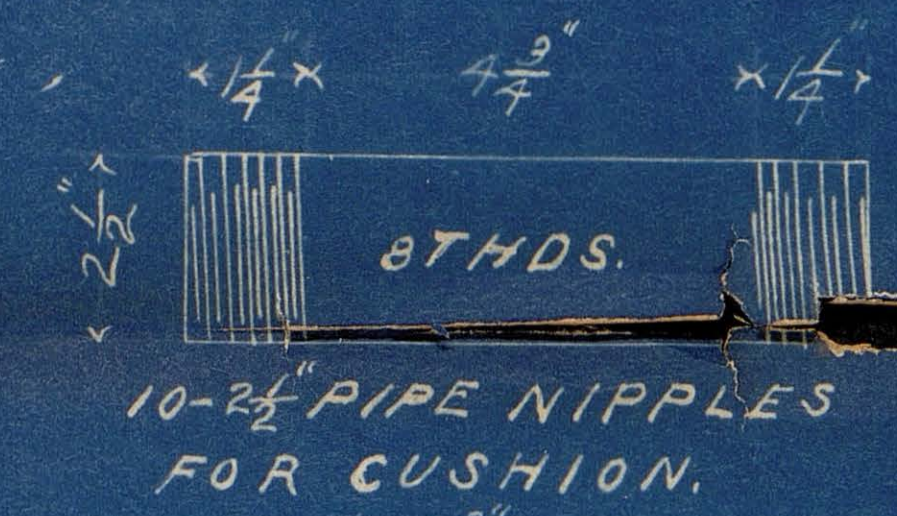




A279  
STEERING HAND WHEEL



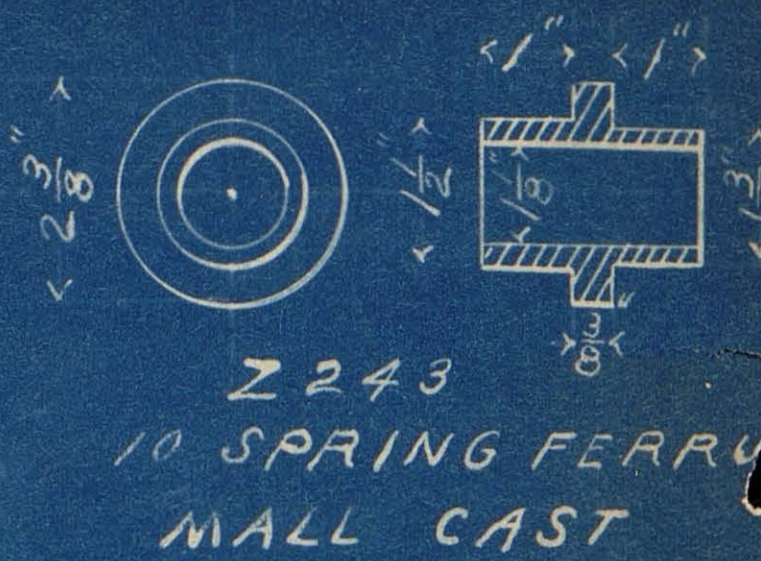
STEERING WHEEL HANDLE.  
DROPPFORGING FALLOVER.



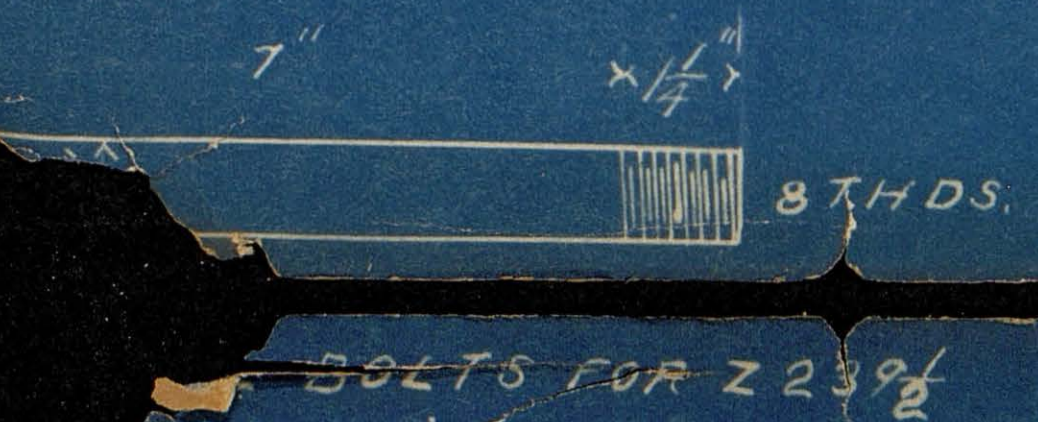
10-2 1/2" PIPE NIPPLES  
FOR CUSHION.



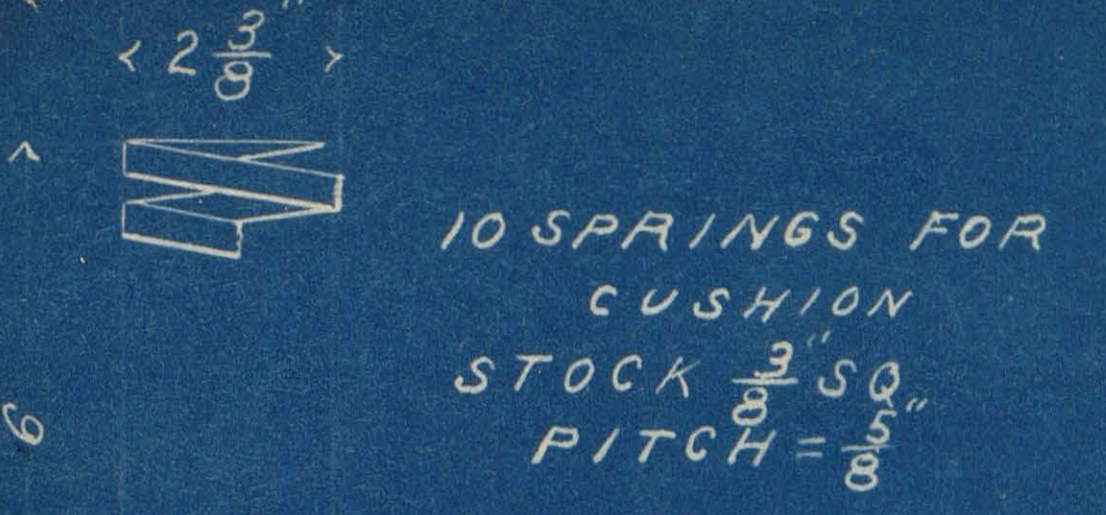
Z242  
10-SPRING FERRULS  
MALL CAST



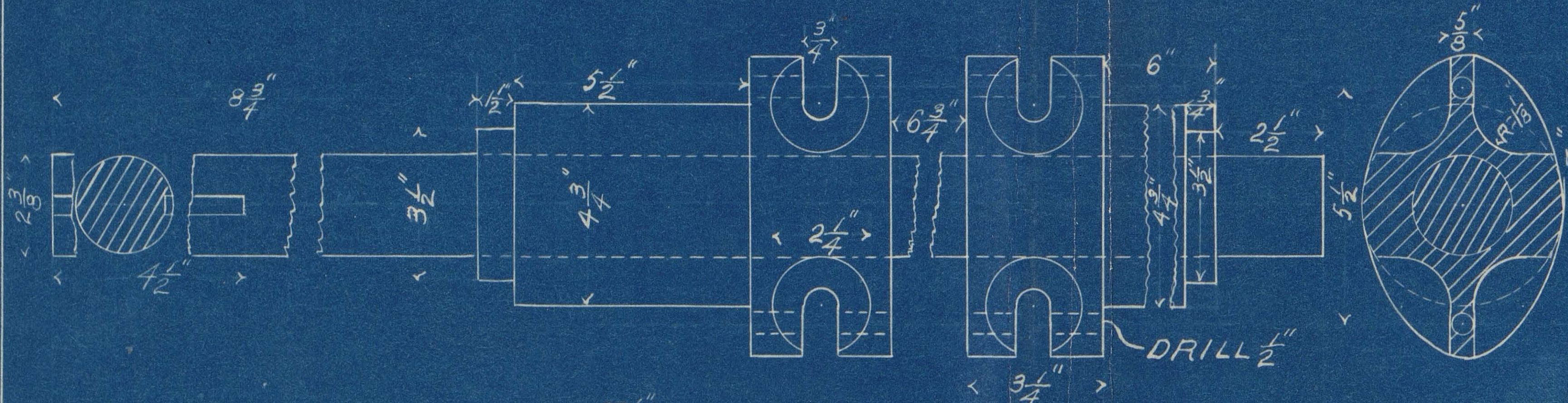
Z243  
10-SPRING FERRULS  
MALL CAST



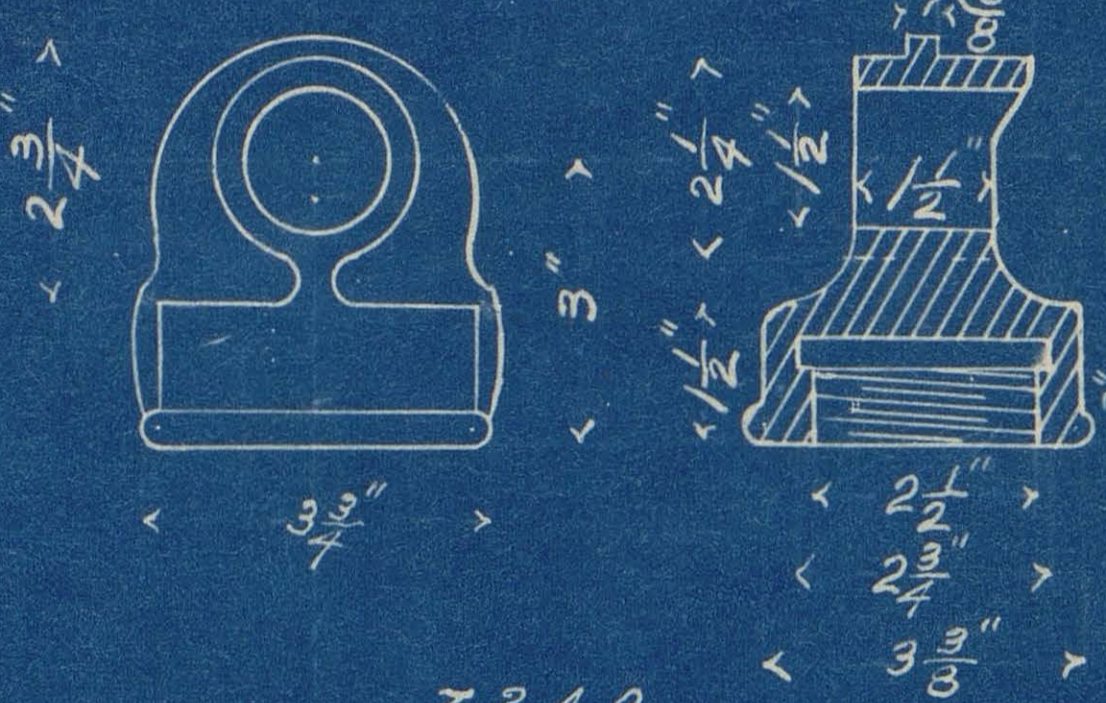
BOLTS FOR Z239 1/2



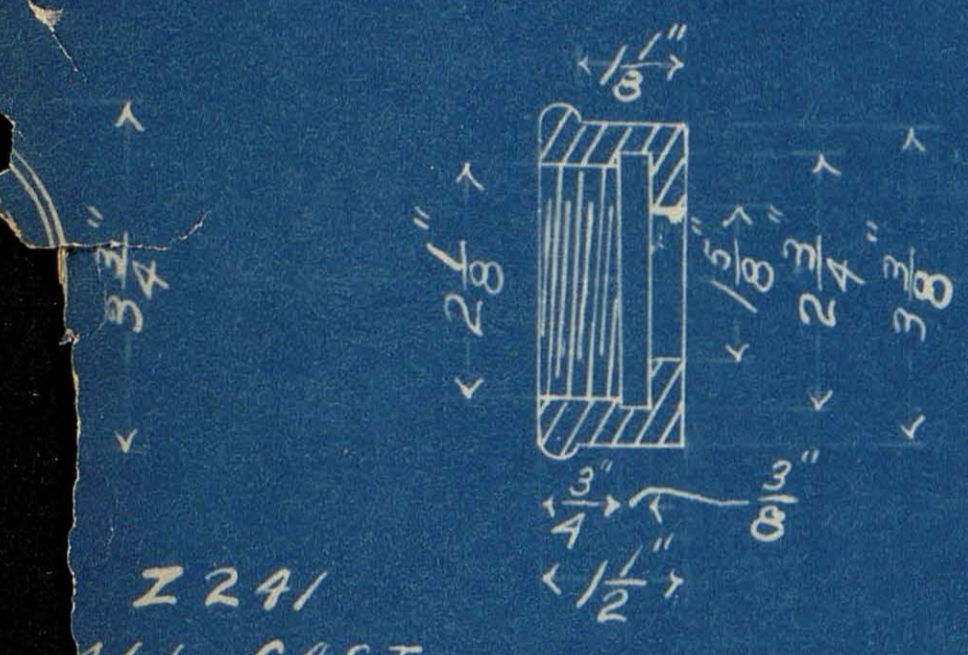
10 SPRINGS FOR  
CUSHION  
STOCK 3/8" SQ  
PITCH = 3/8"



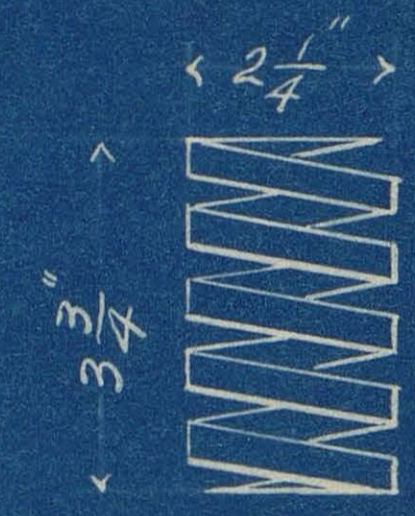
STEERING ROLLER  
Z74  
2 3/8" MACH. STEEL SHAFT-CAST IN



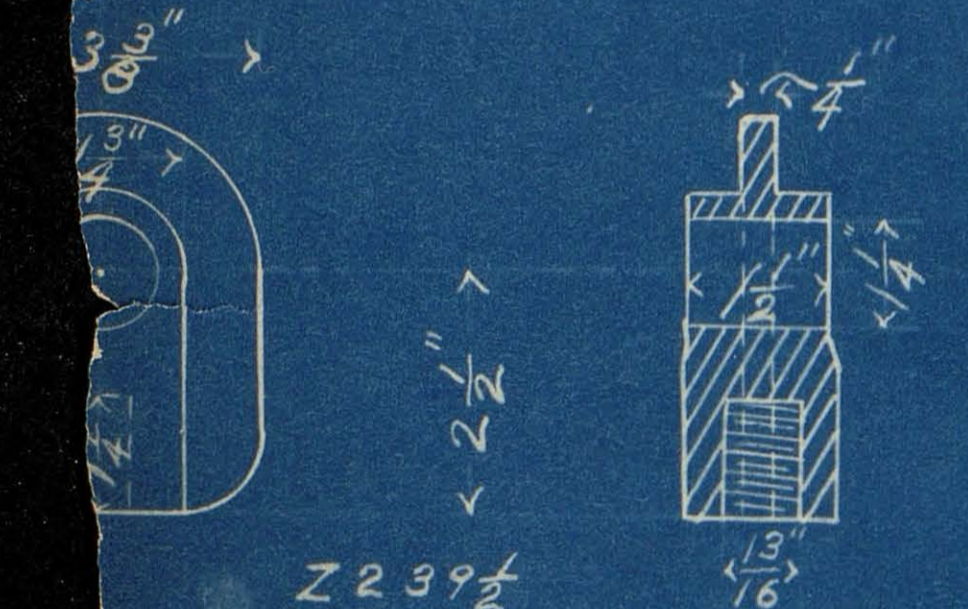
Z240  
10 MALL CAST THREAD FOR 1/2" PIPE.



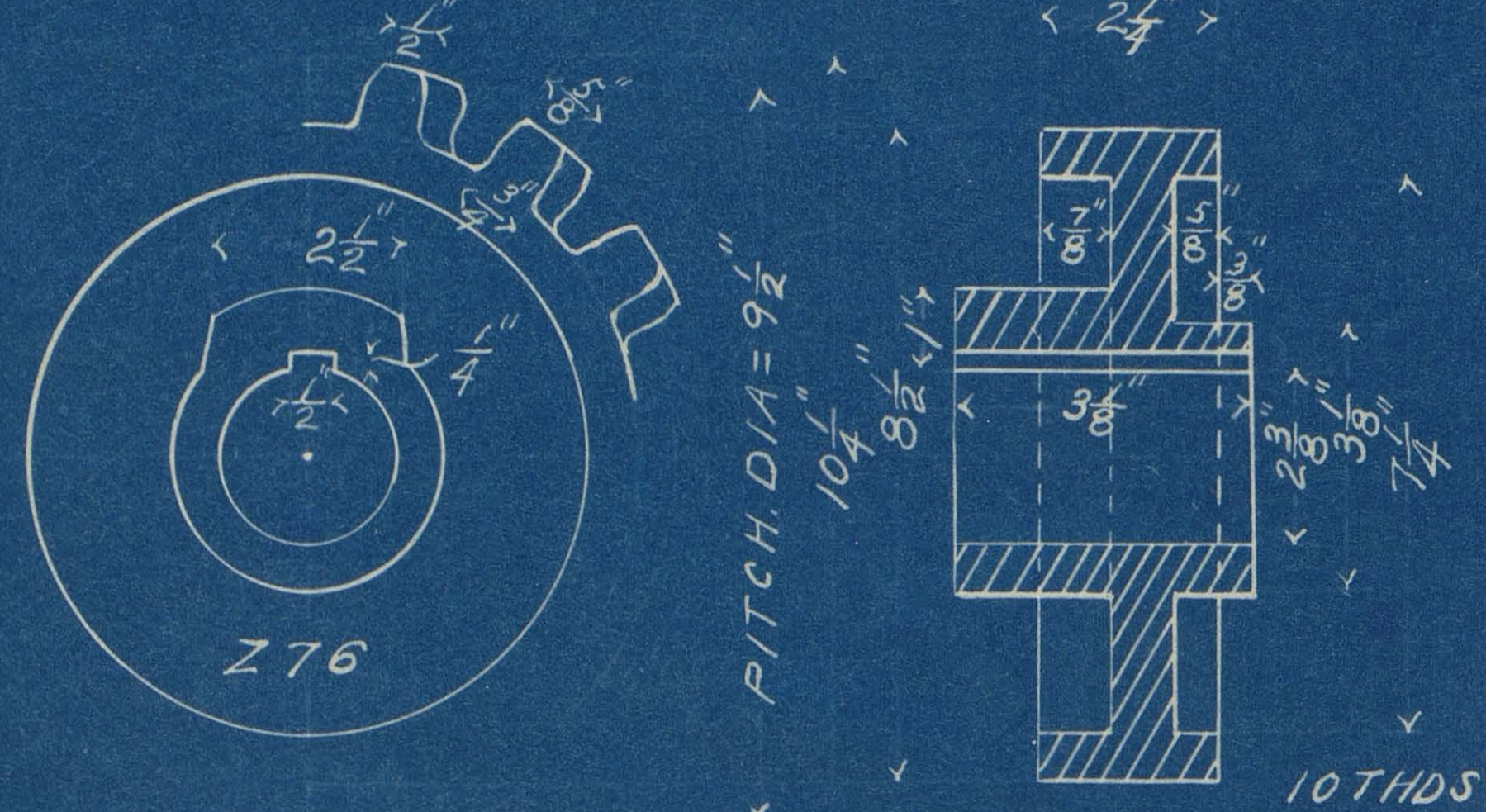
Z241  
10 MALL CAST  
THREAD FOR 2 1/2" PIPE.



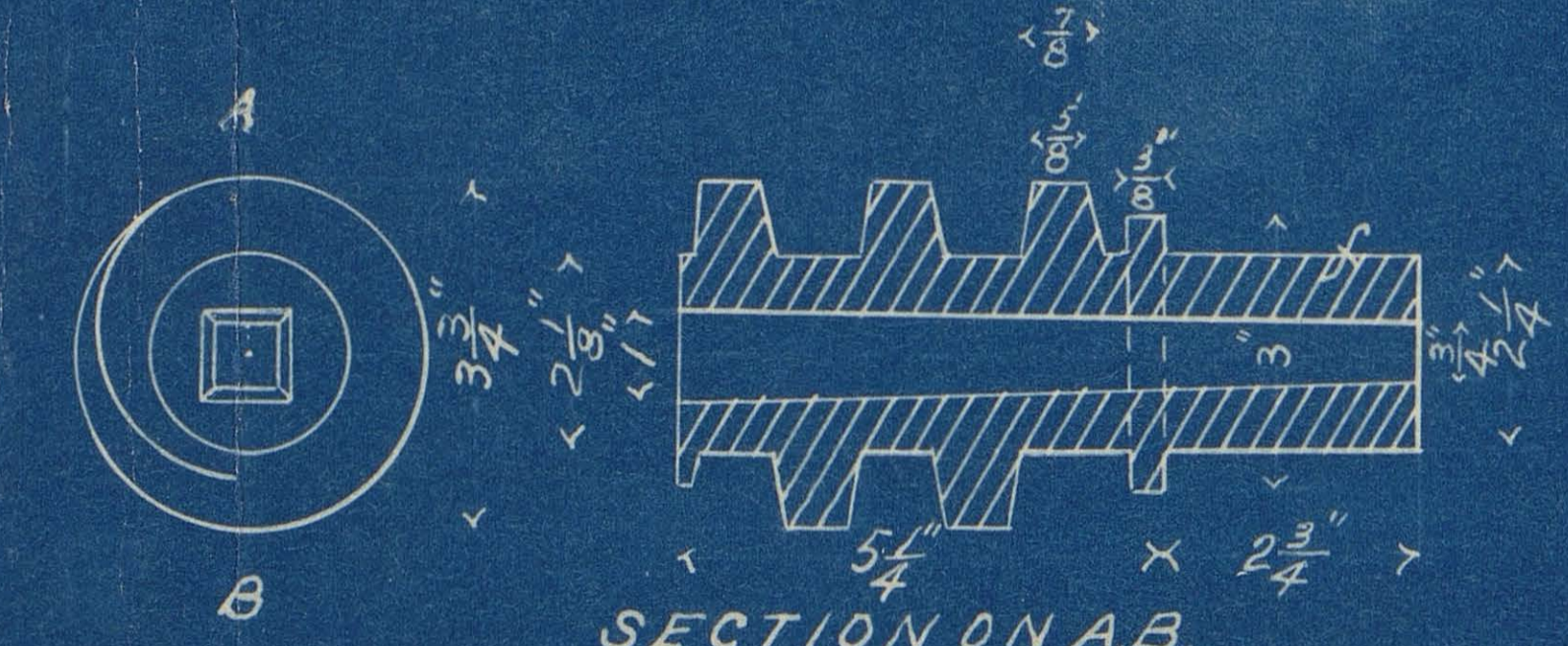
2 SPRINGS FOR STEERING  
SHAFT. PITCH = 3/4" STOCK 3/8"



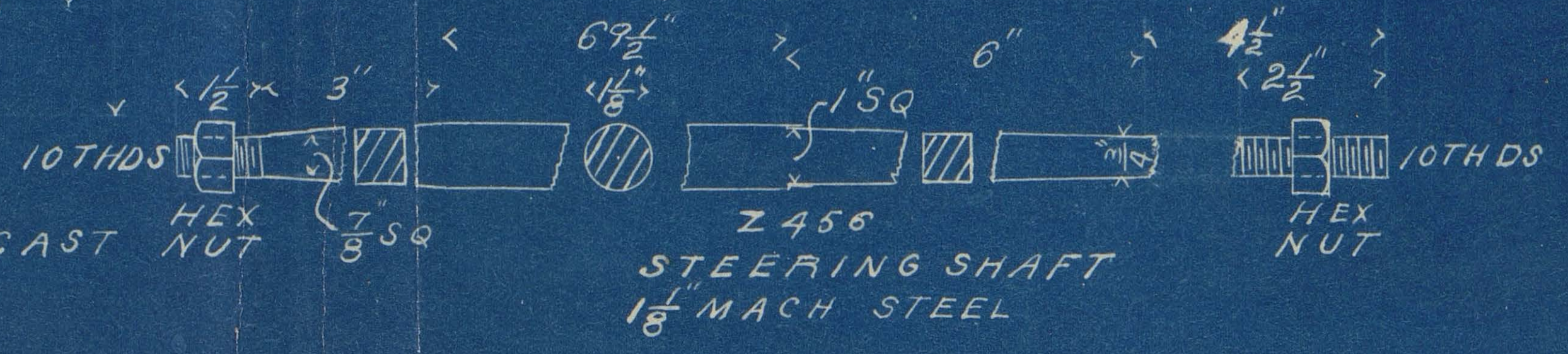
Z239 1/2  
10 MALL CAST  
DRILL AND TAP 1" BOLT 8THDS.



STEERING ROLLER WORM GEAR. CAST  
CIRCULAR P = 1 3/4" 17 TEETH.



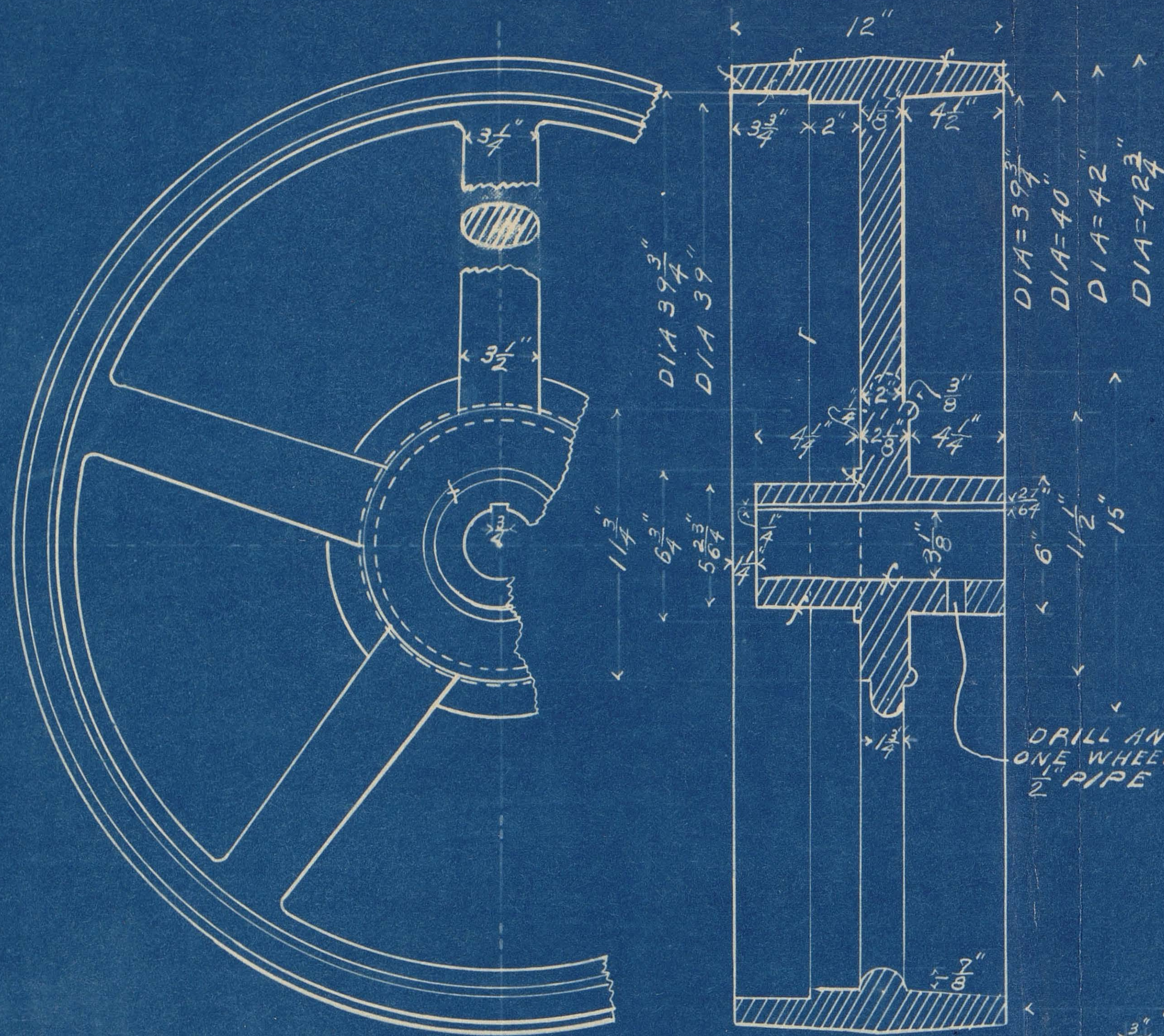
STEERING SHAFT WORM CAST.  
PITCH = 1 3/4"  
7 1/2"



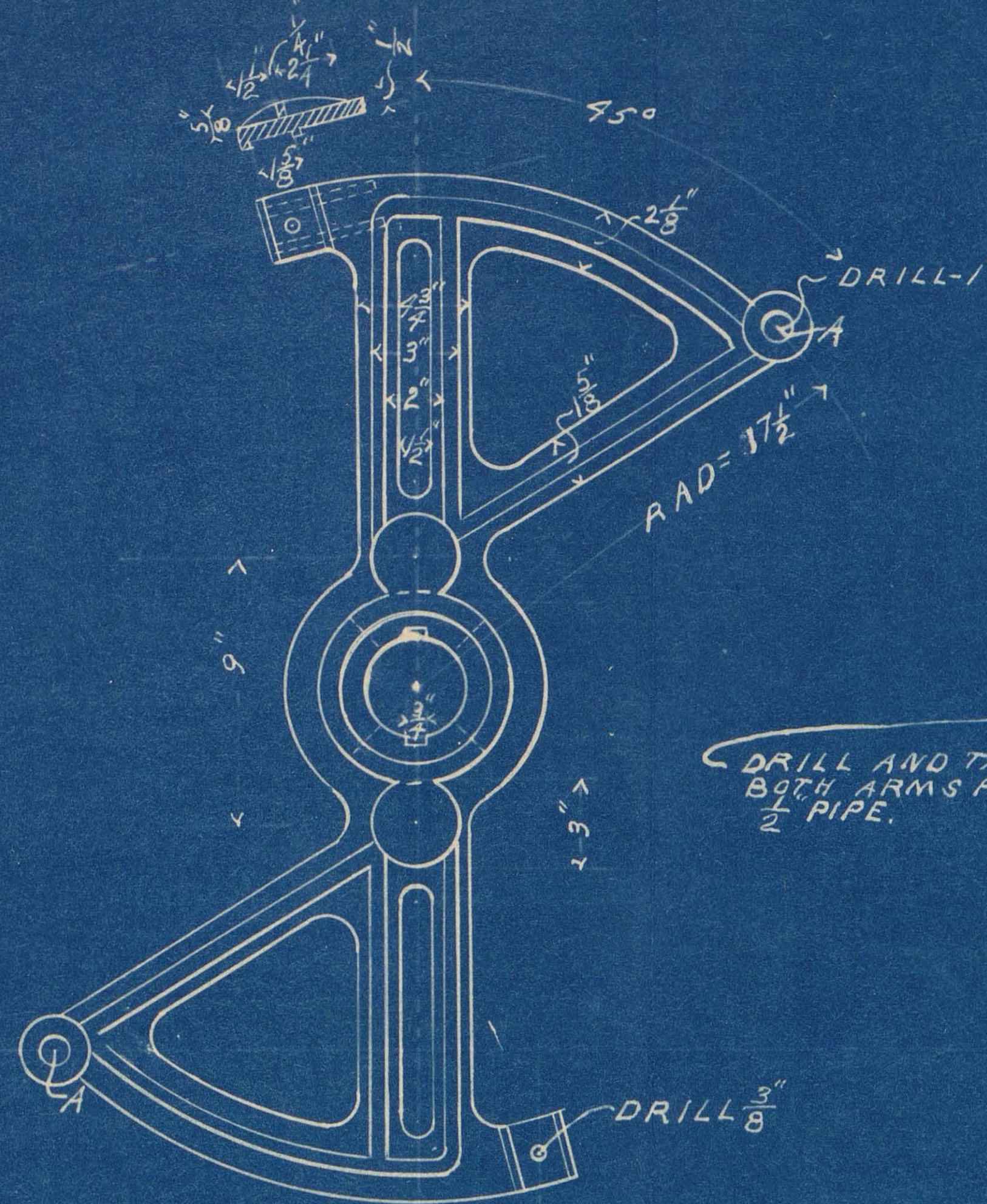
Z456  
STEERING SHAFT  
1 1/8" MACH STEEL

25 HP  
GASOLINE ENGINE  
SCALE: 3"  
MEANS FINISH  
OF  
ENGINEERING  
C.  
ANSAS  
ED BY E.D.R.

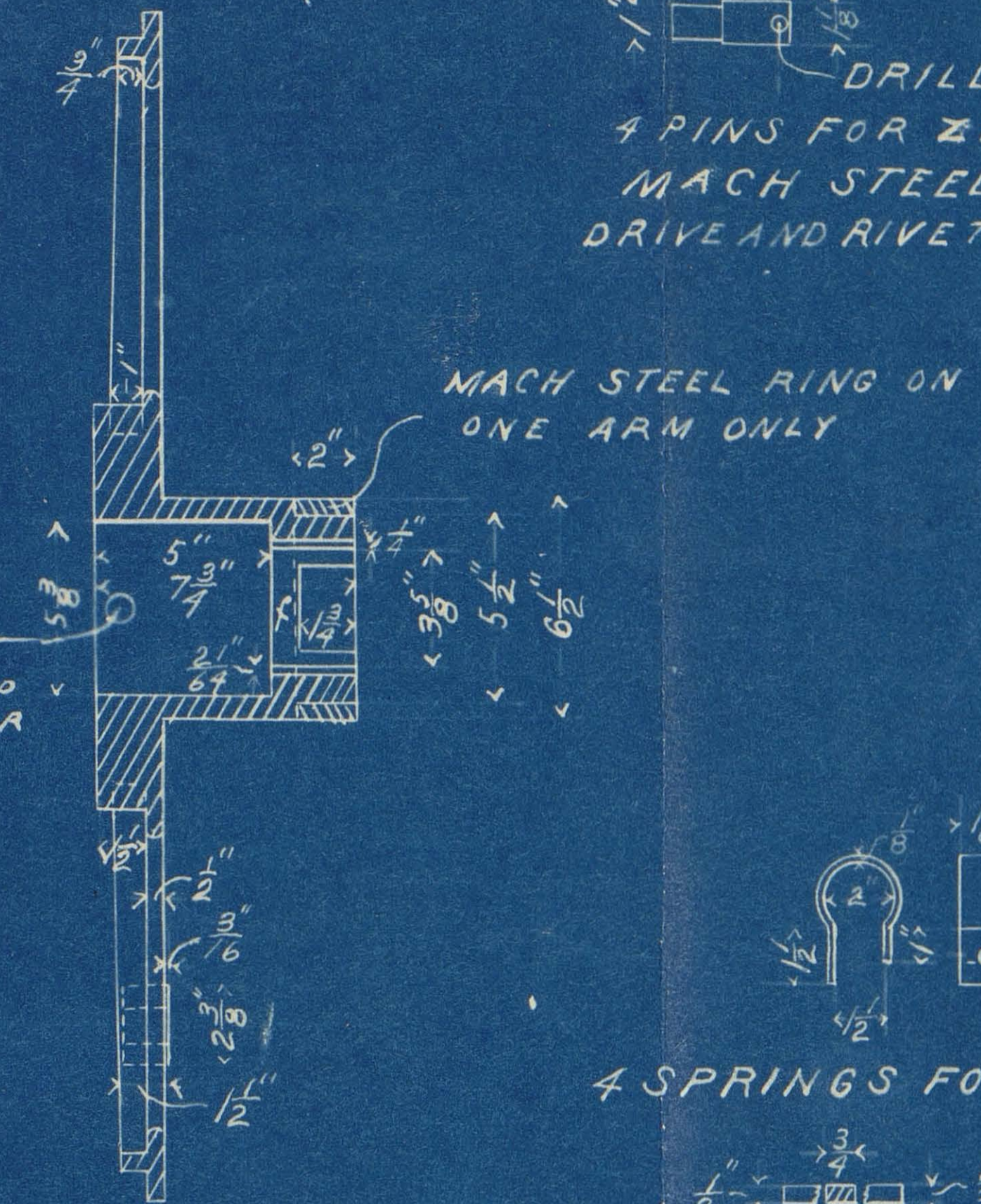




Z1004  
2 BAND WHEELS  
KEY SEAT ONE WHEEL ONLY  
DRILL AND TAP OTHER WHEEL  
FOR 1/2 PIPE

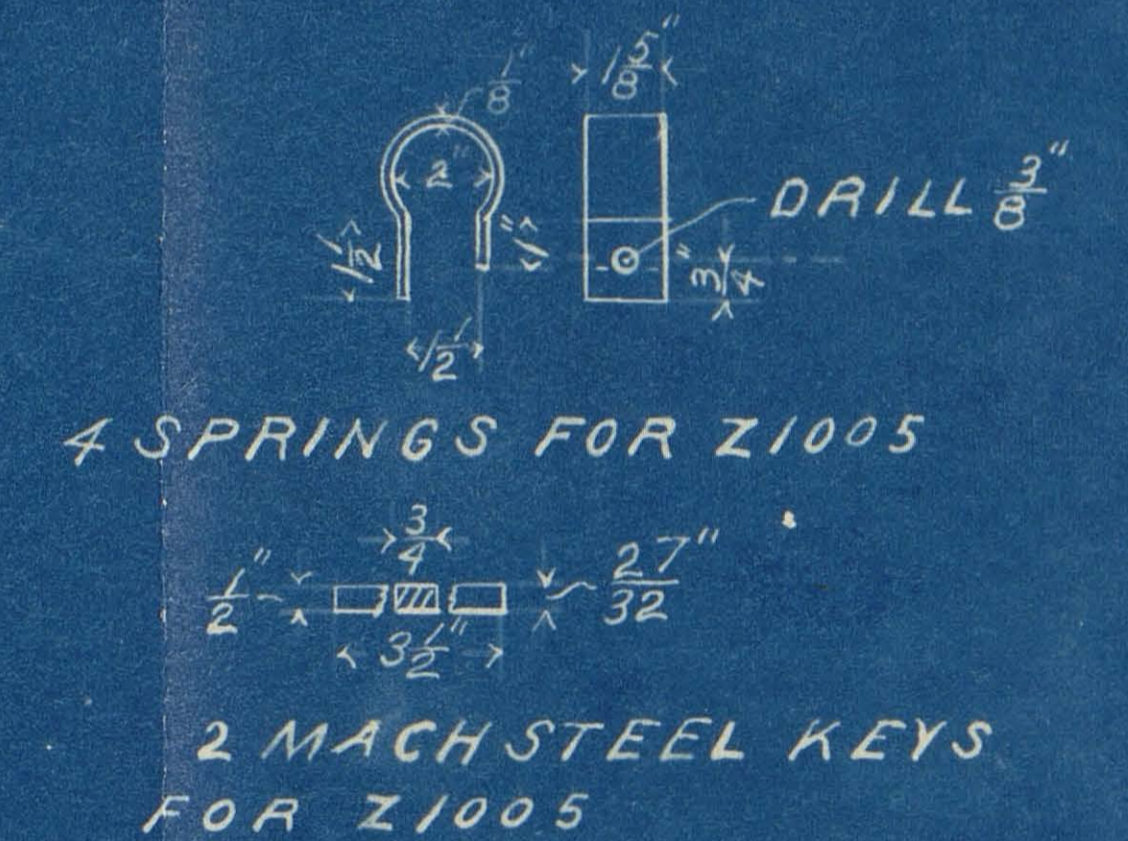


Z1005  
2 CLUTCH ARMS GRAYCAST'G  
KEY SEAT AND SHRINK  
BAND ON ONE ARM ONLY DRILL  
AND TAP OTHER ARM FOR 1/2 PIPE

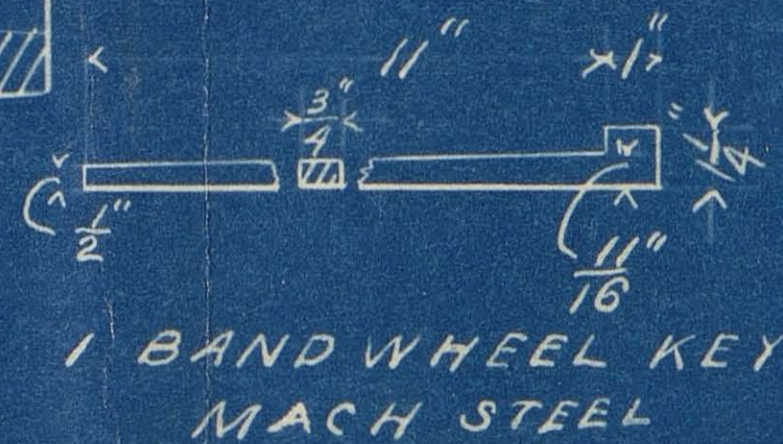


DRILL 1/4"  
4 PINS FOR Z1005  
MACH STEEL  
DRIVE AND RIVET AT A

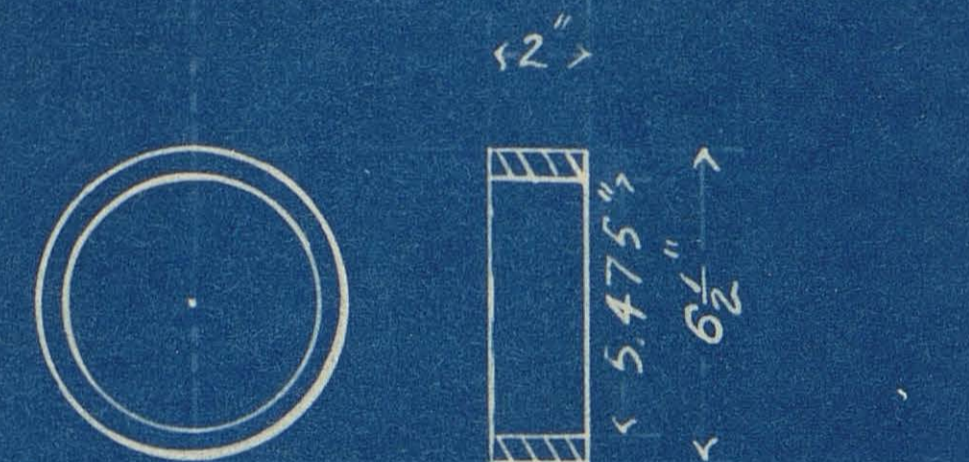
MACH STEEL RING ON  
ONE ARM ONLY



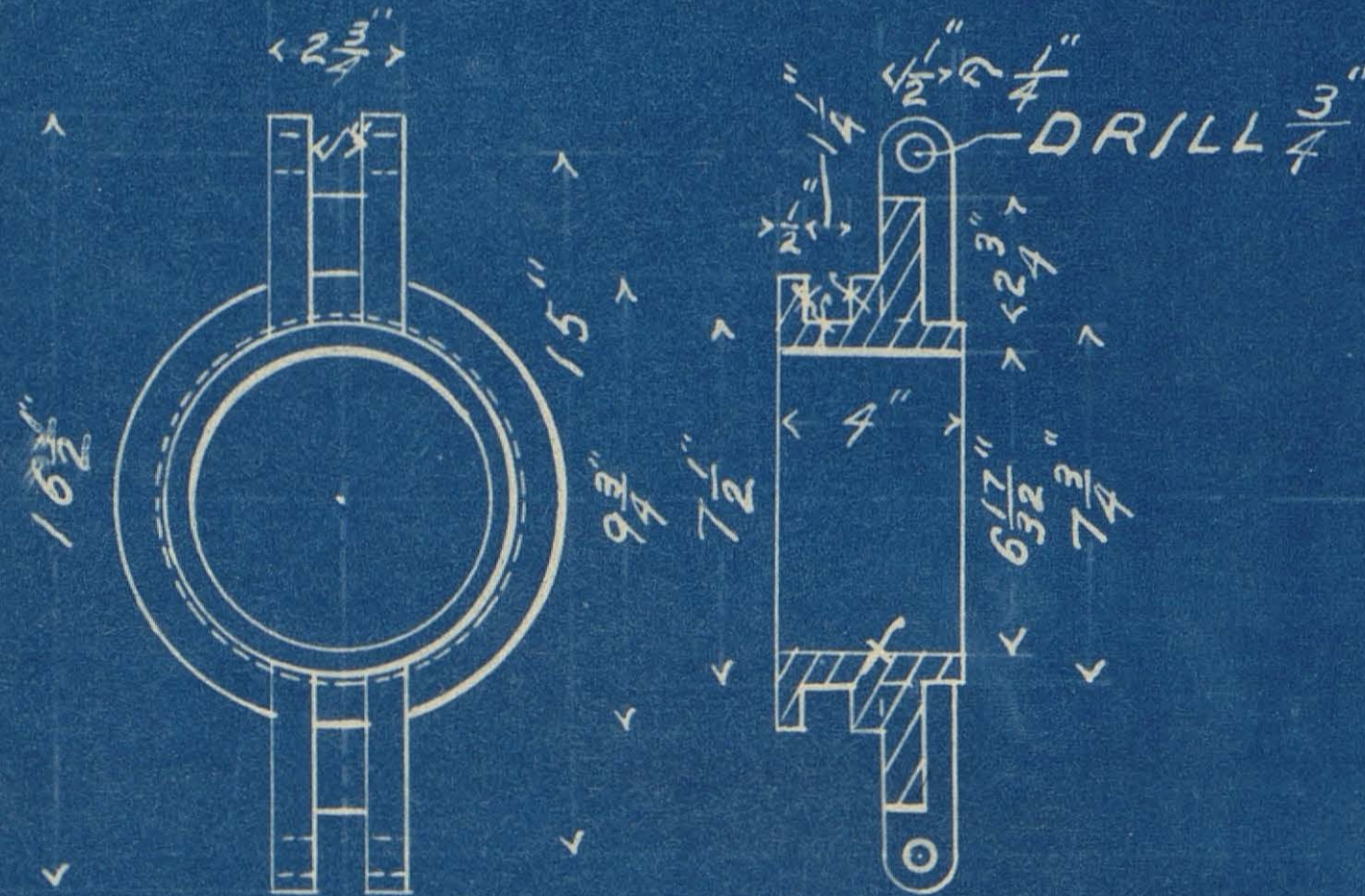
4 SPRINGS FOR Z1005  
2 MACH STEEL KEYS  
FOR Z1005



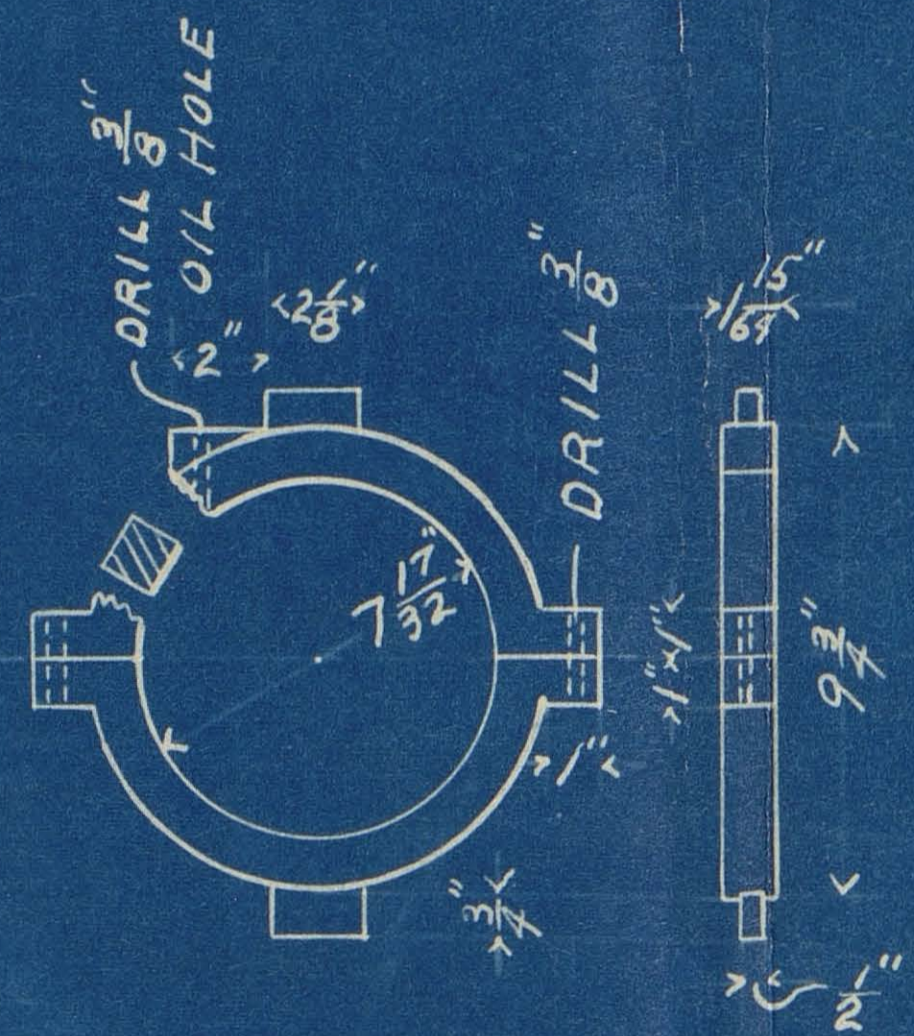
1 BAND WHEEL KEY  
MACH STEEL



1 MACH STEEL RING FOR  
Z1005 FALL OVER  
SHRINK ON Z1005



ZZ87  
2 FRICTION COLLARS



A1100 UPPER HALF  
A1101 LOWER HALF  
2 SET SHIFTING COLLARS



4 MACH STEEL PINS  
FOR ZZ87

## DETAILS OF 25 HP GASOLINE TRACTION ENGINE

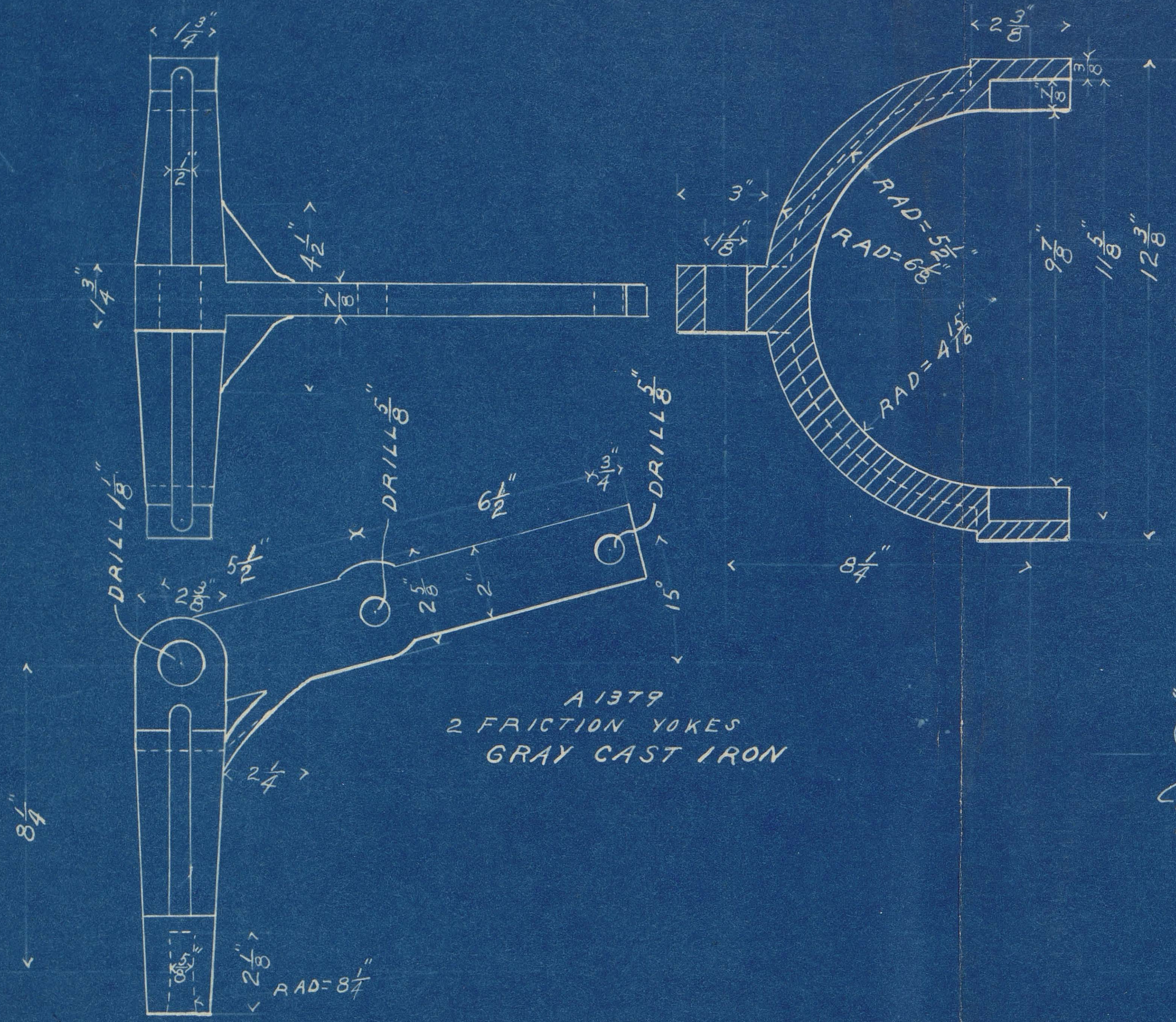
SCALE: 1/2" = 1 FT. f MEANS FINISH

DEPARTMENT OF  
MECHANICAL ENGINEERING

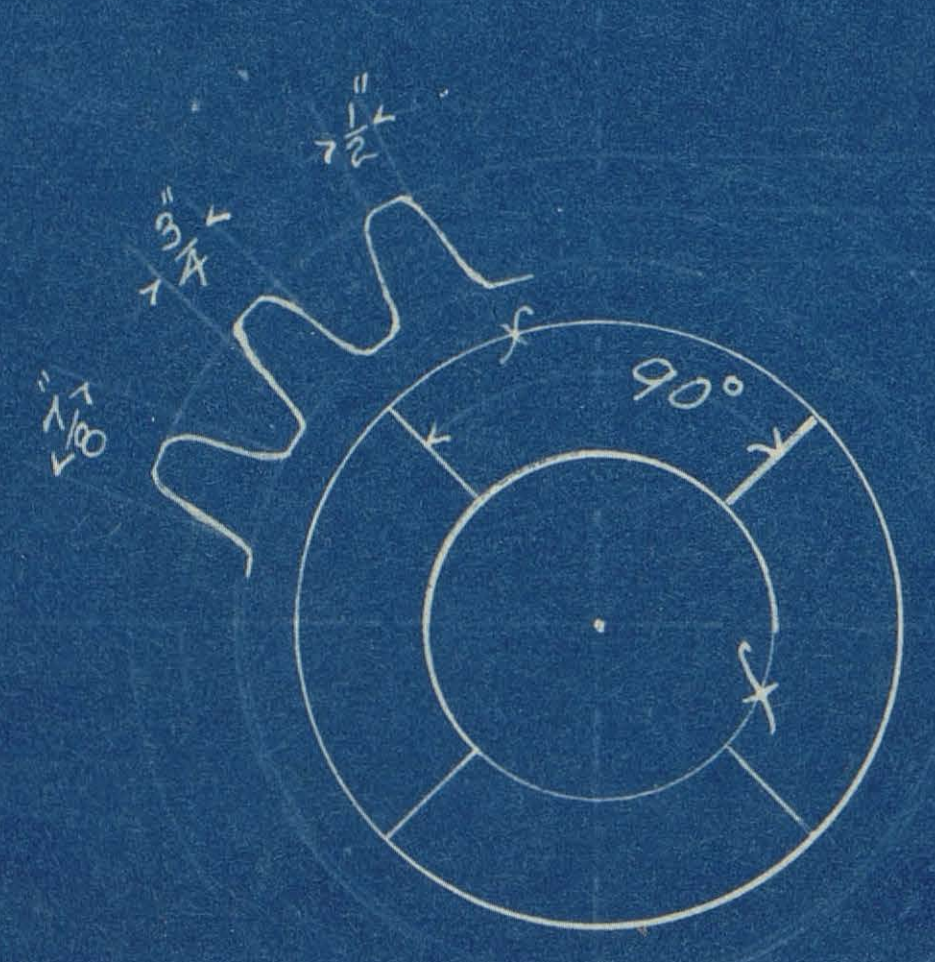
K.S.A.C.

MANHATTAN — KANSAS

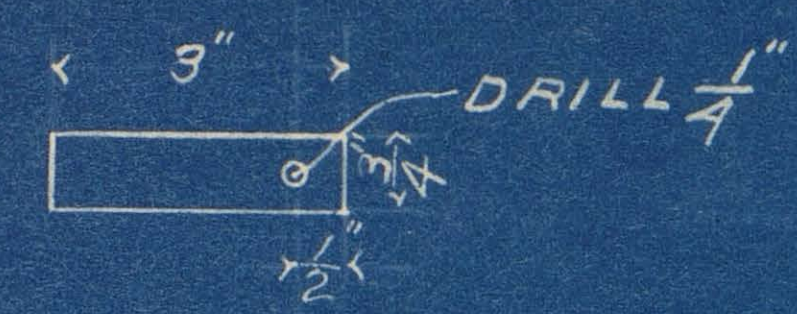
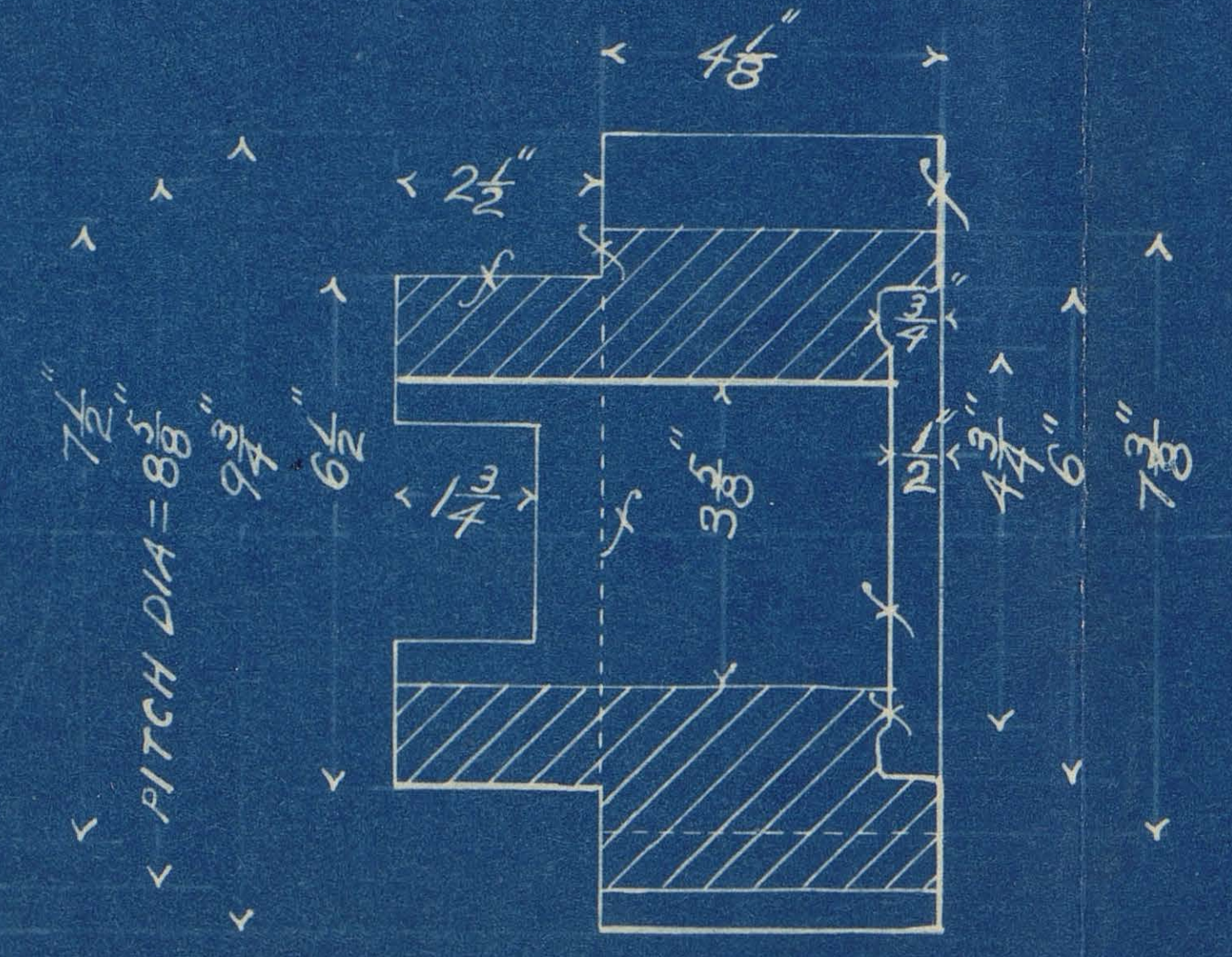




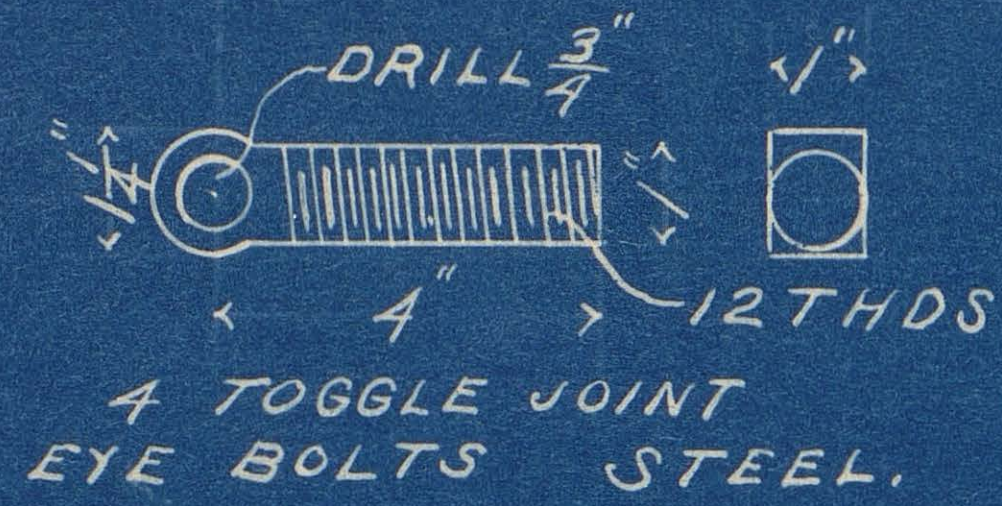
A1379  
2 FRICTION YOKES  
GRAY CAST IRON



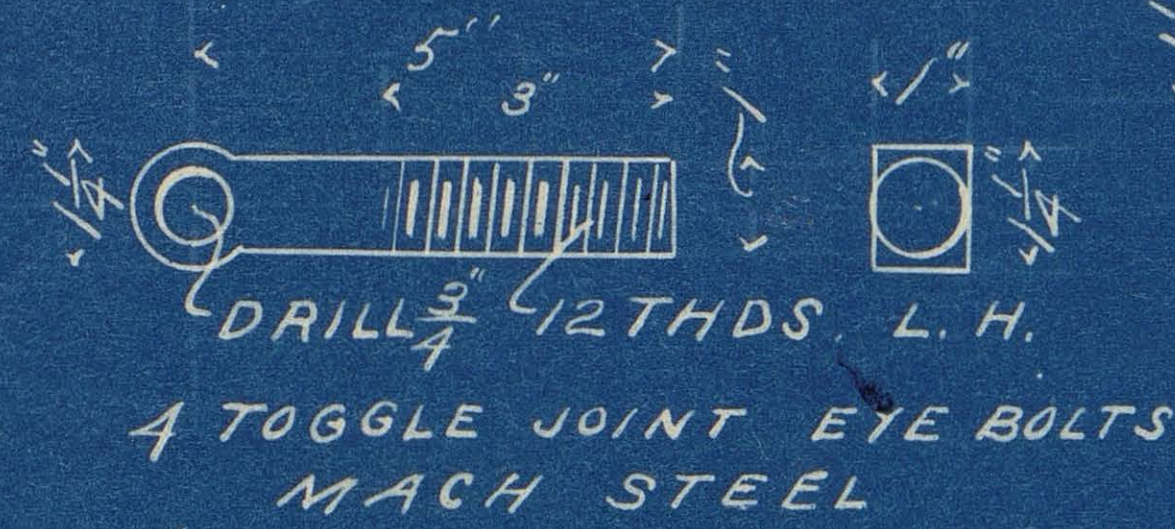
Z153  
1 CLUTCH PINION  
GRAY CAST IRON  
17 TEETH  
CIRC. P. 1.592" DIA. P. 0.5075"



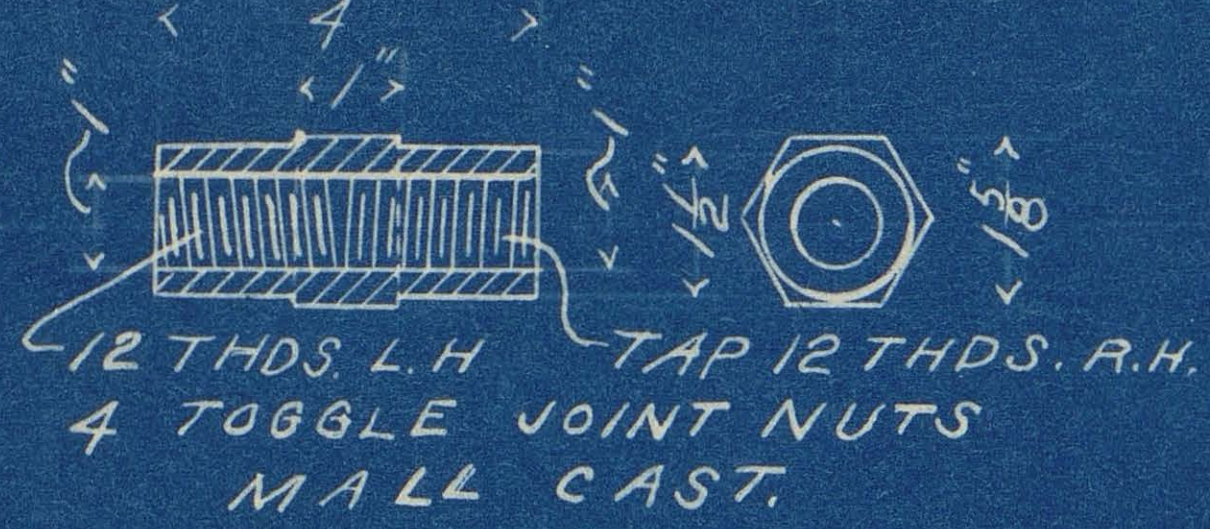
4 MACH STEEL  
PINS FOR Z283



4 TOGGLE JOINT  
EYE BOLTS  
STEEL.



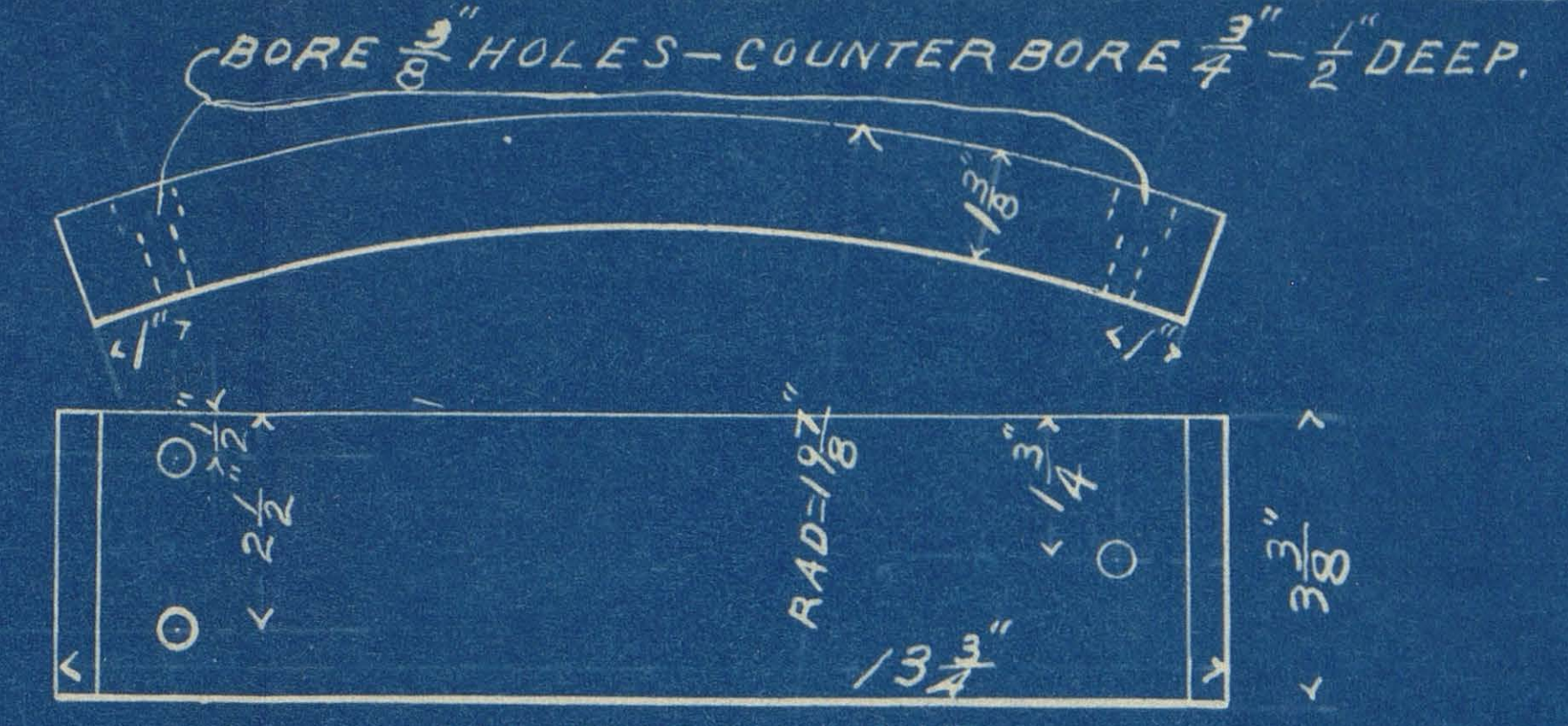
4 TOGGLE JOINT EYE BOLTS  
MACH STEEL



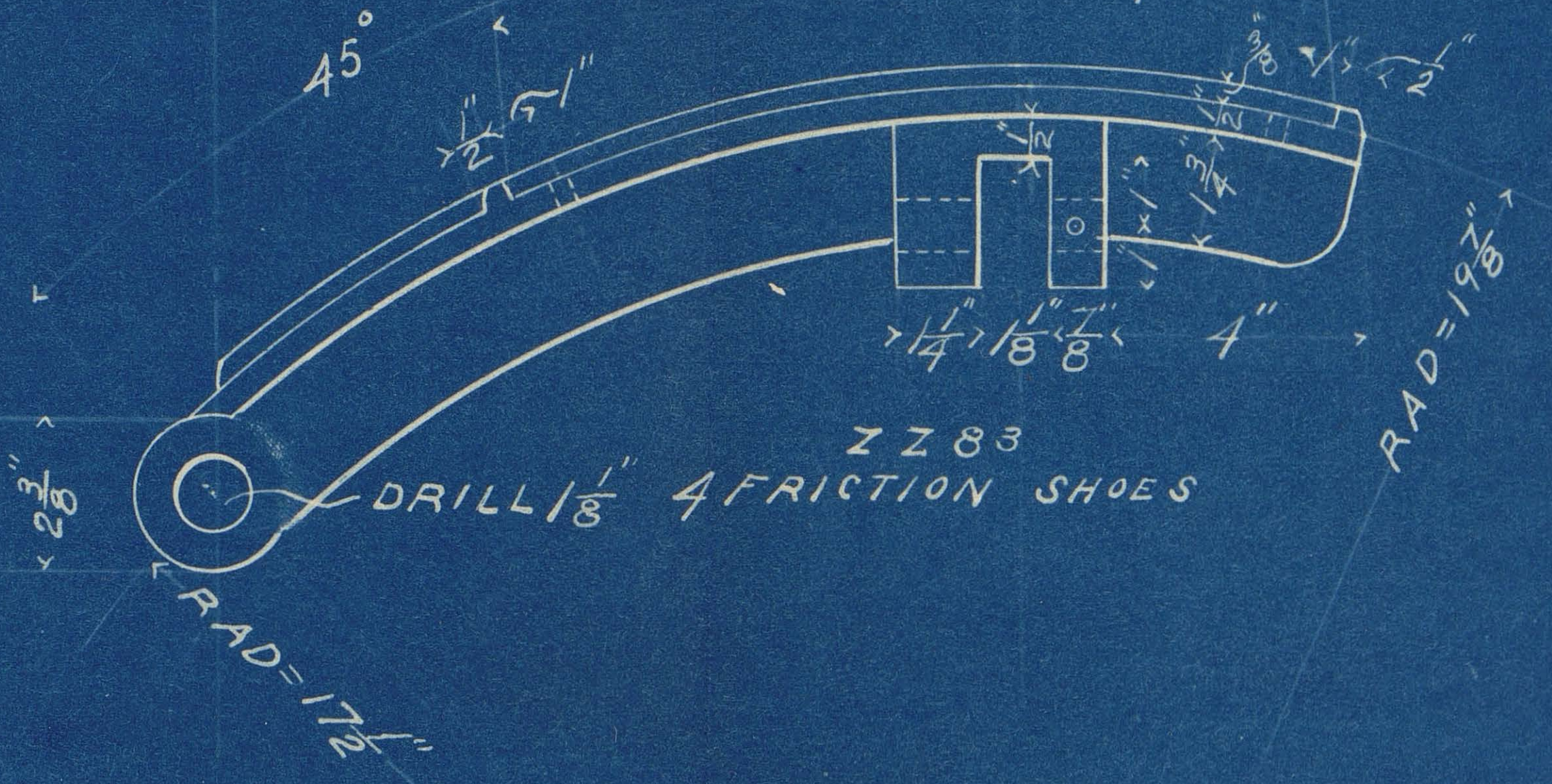
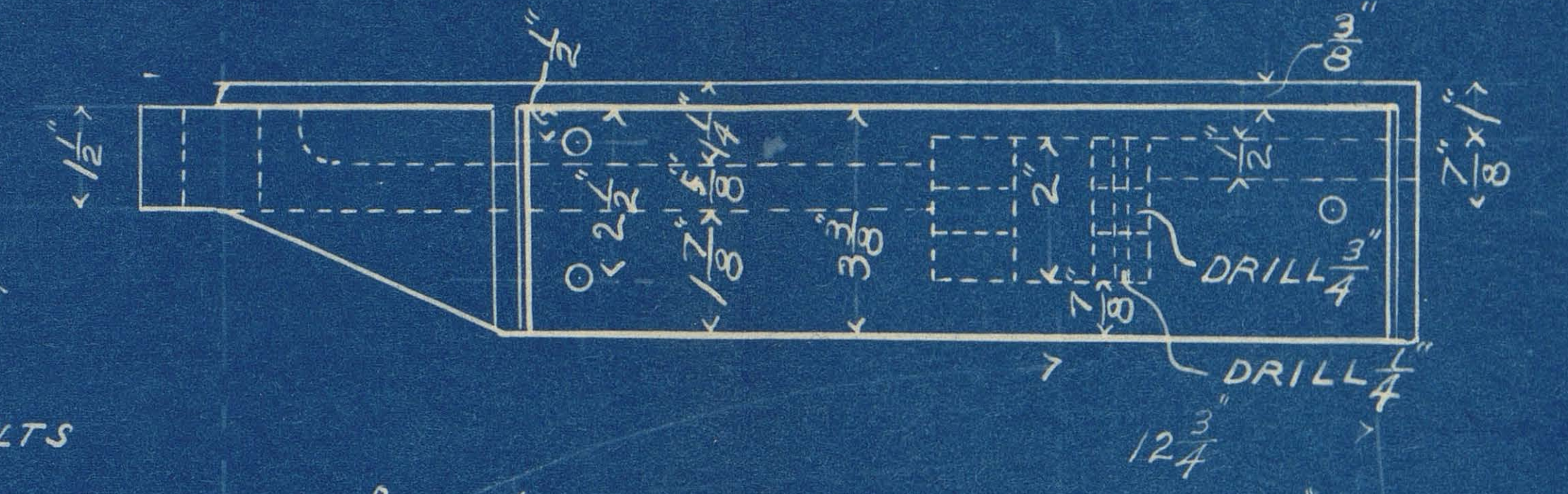
12 THDS. L.H. TAP 12 THDS. R.H.  
4 TOGGLE JOINT NUTS  
MALL CAST.



1" X 12 THDS. R.H.  
4 HEX. LOCK NUTS FOR  
TOGGLE JOINT EYE BOLTS



4 MAPLE BLOCKS  
FOR Z283



DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: - 3" = 1 FT f MEANS FINISH

DEPARTMENT OF  
MECHANICAL ENGINEERING

K.S.A.C.

MANHATTAN - KANSAS

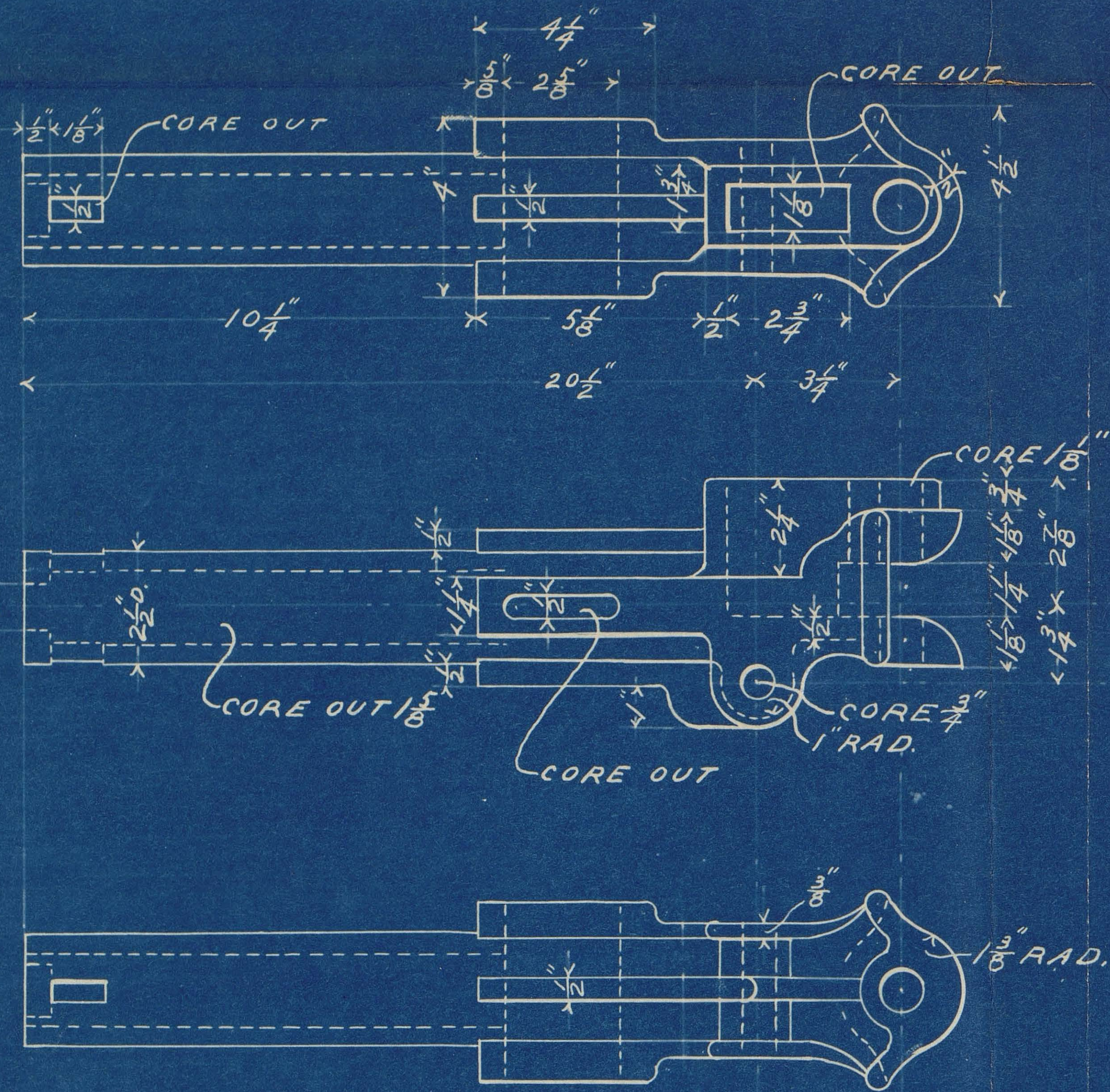




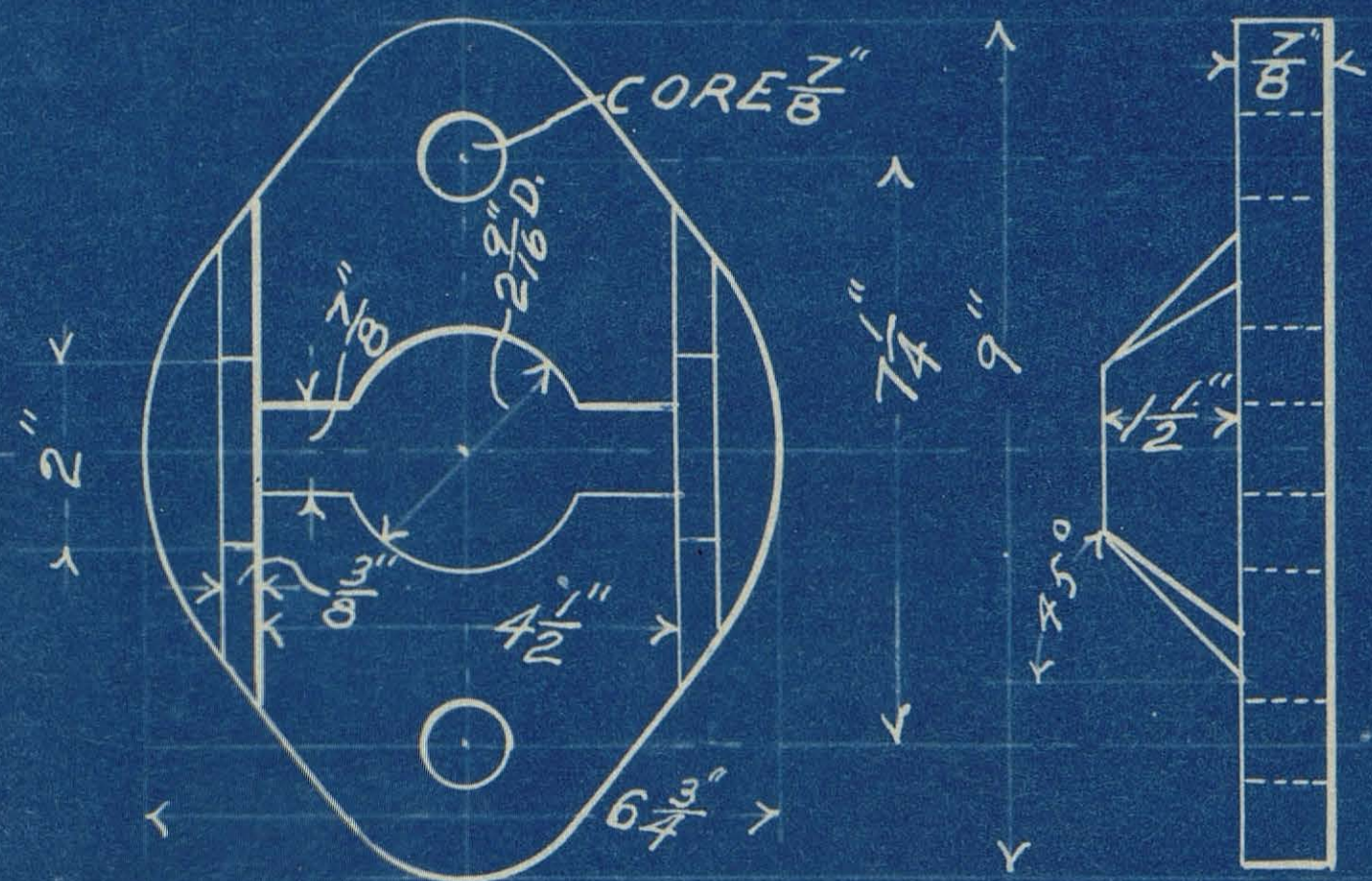




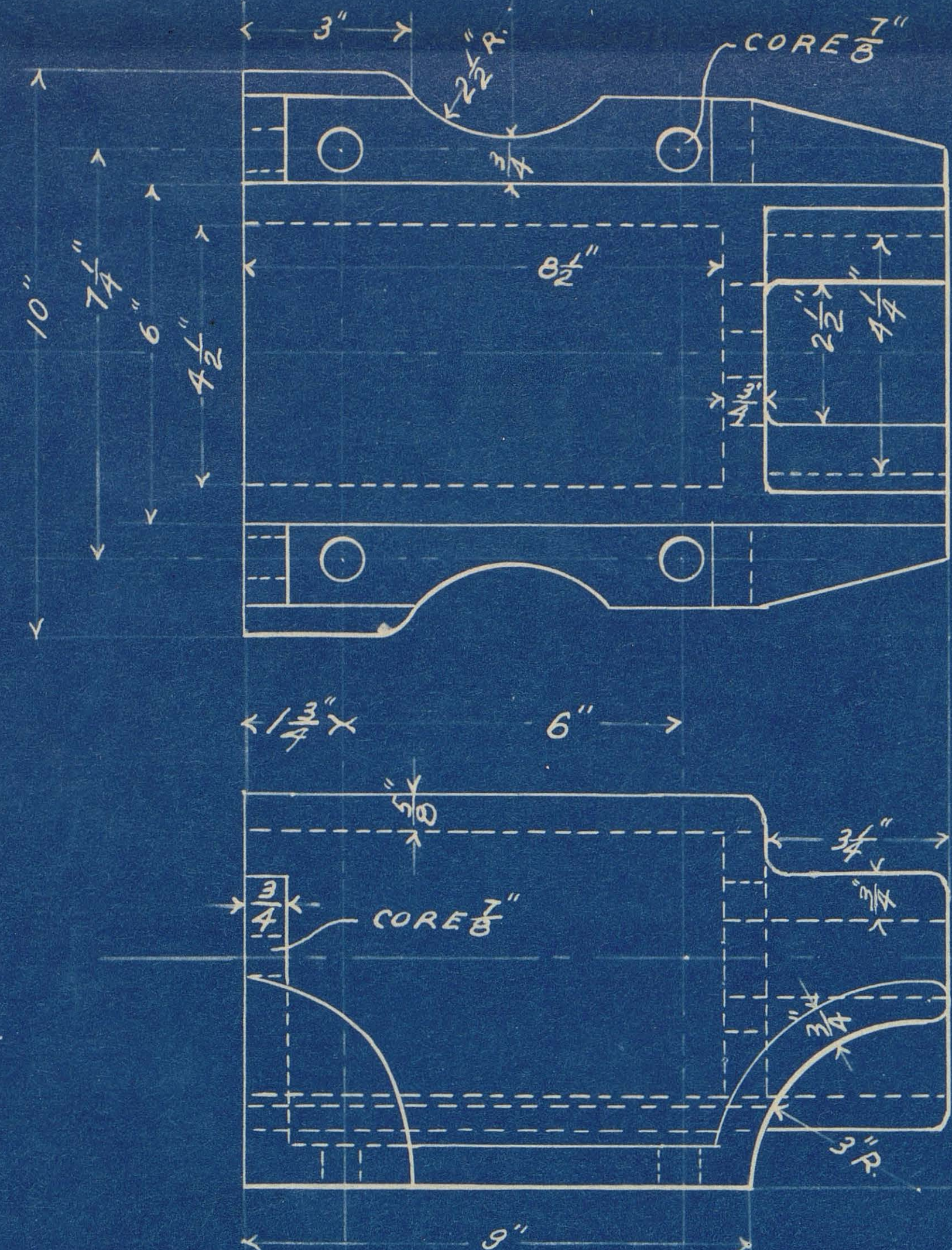




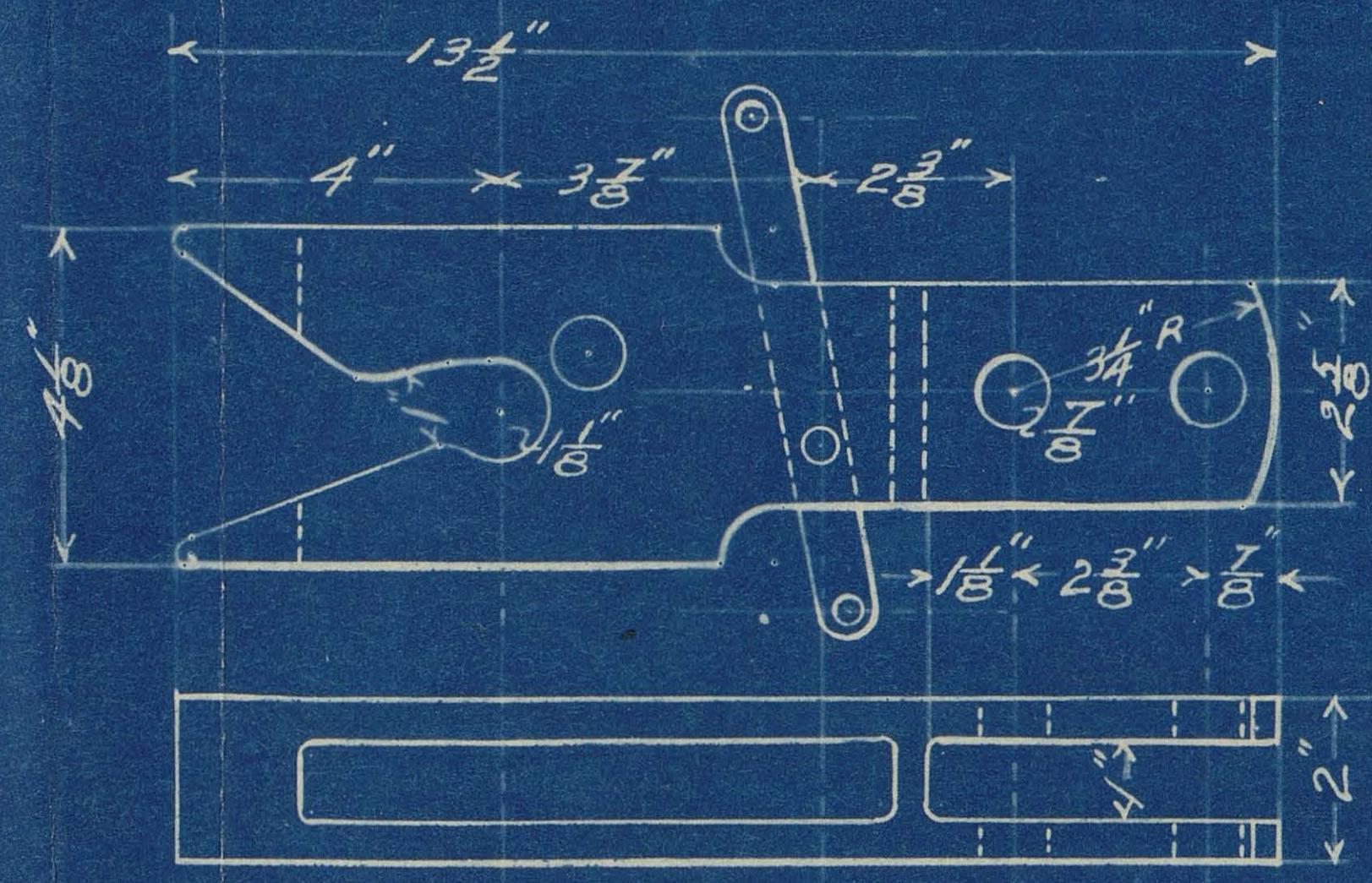
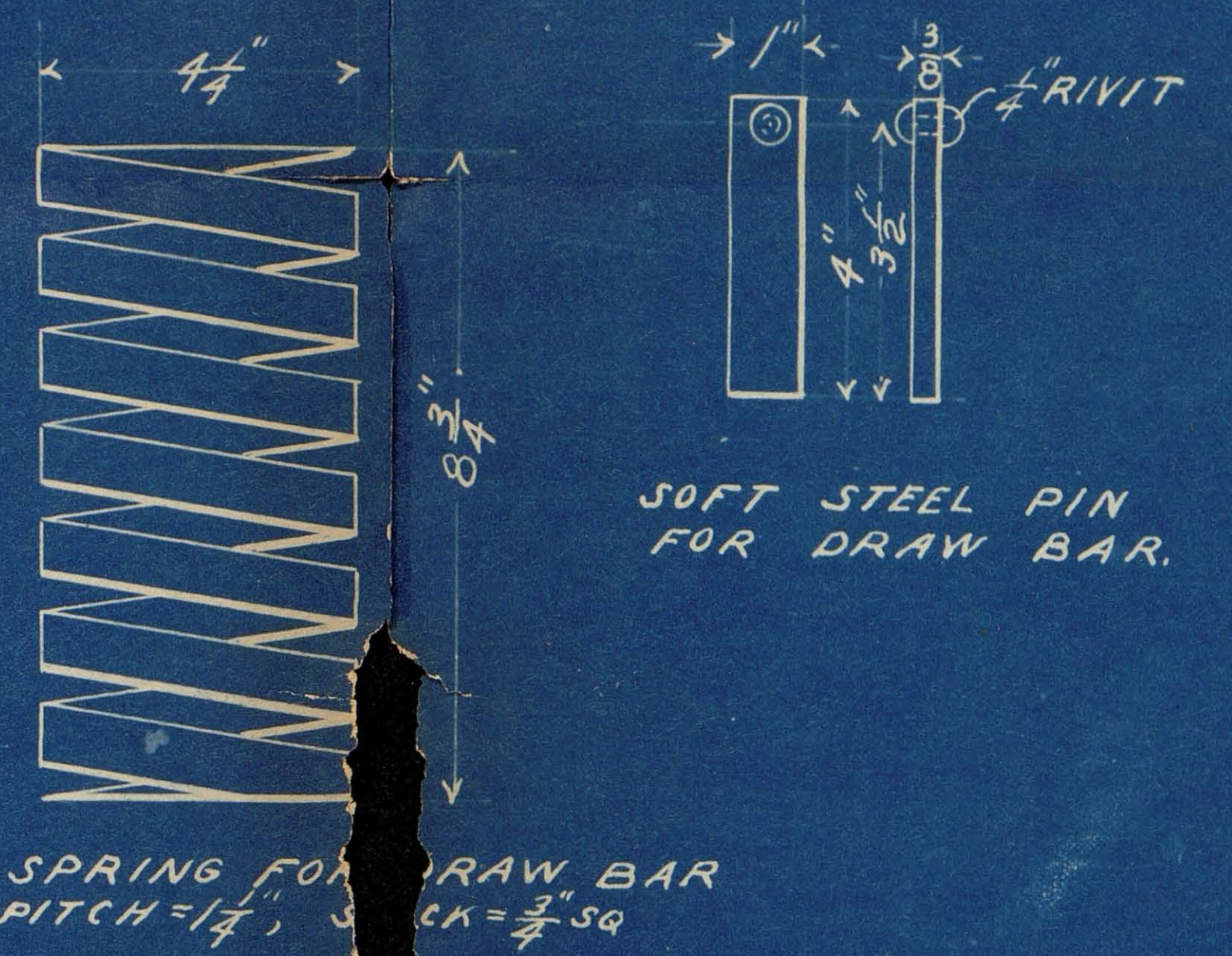
Z833  
DRAW BAR  
GRAY CAST IRON



Z835  
CAP FOR DRAW BAR.



Z834  
DRAW BAR CASING  
GRAY CAST IRON



RANDOLPH COUPLER  
MALL-CASTG.

DETAILS OF 25 HP  
GASOLINE ACTION ENGINE

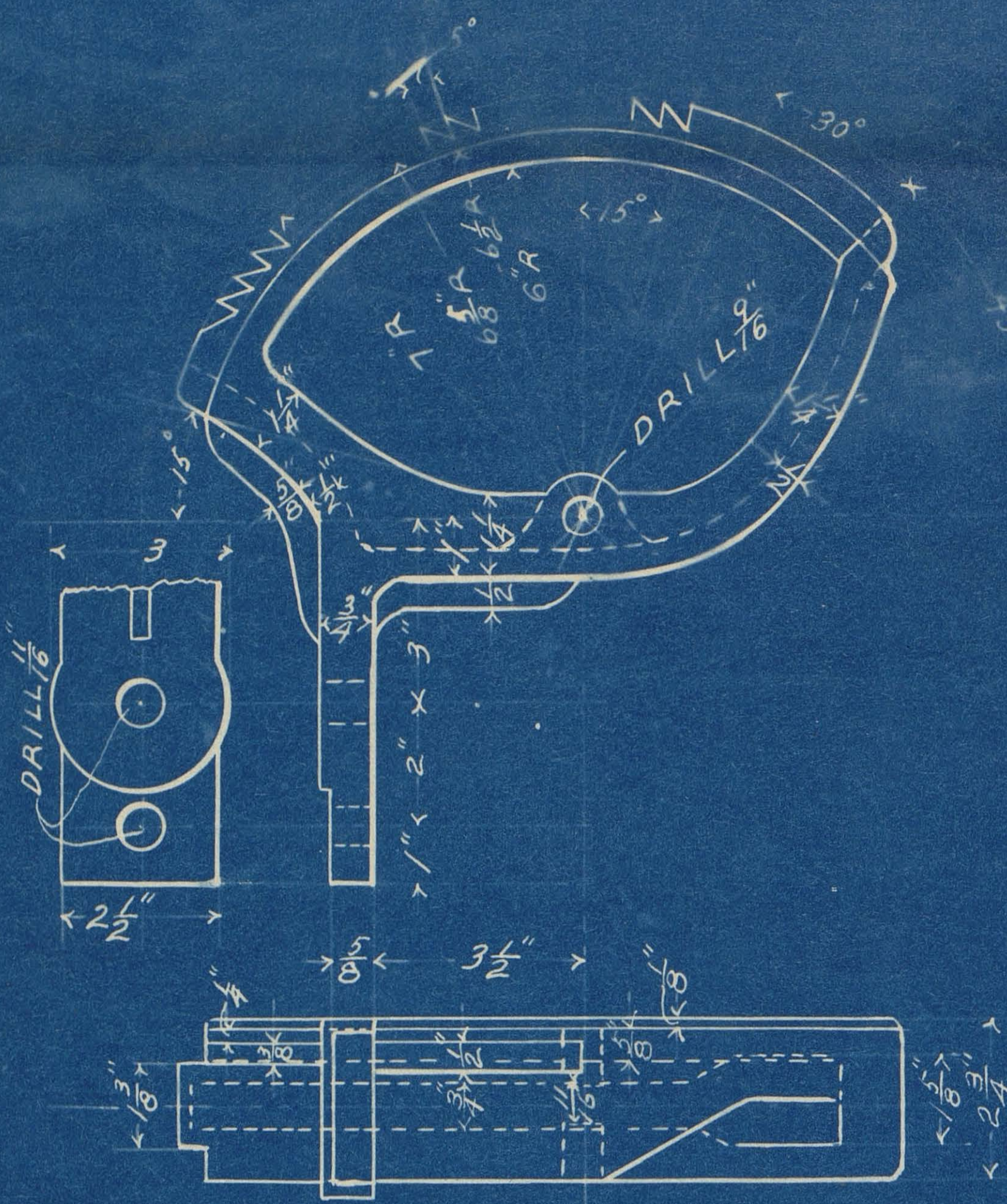
SCALE: - 3" = 1 FT. f MEANS FINISH.

DEPARTMENT OF  
MECHANICAL ENGINEERING  
M.S.A.C.

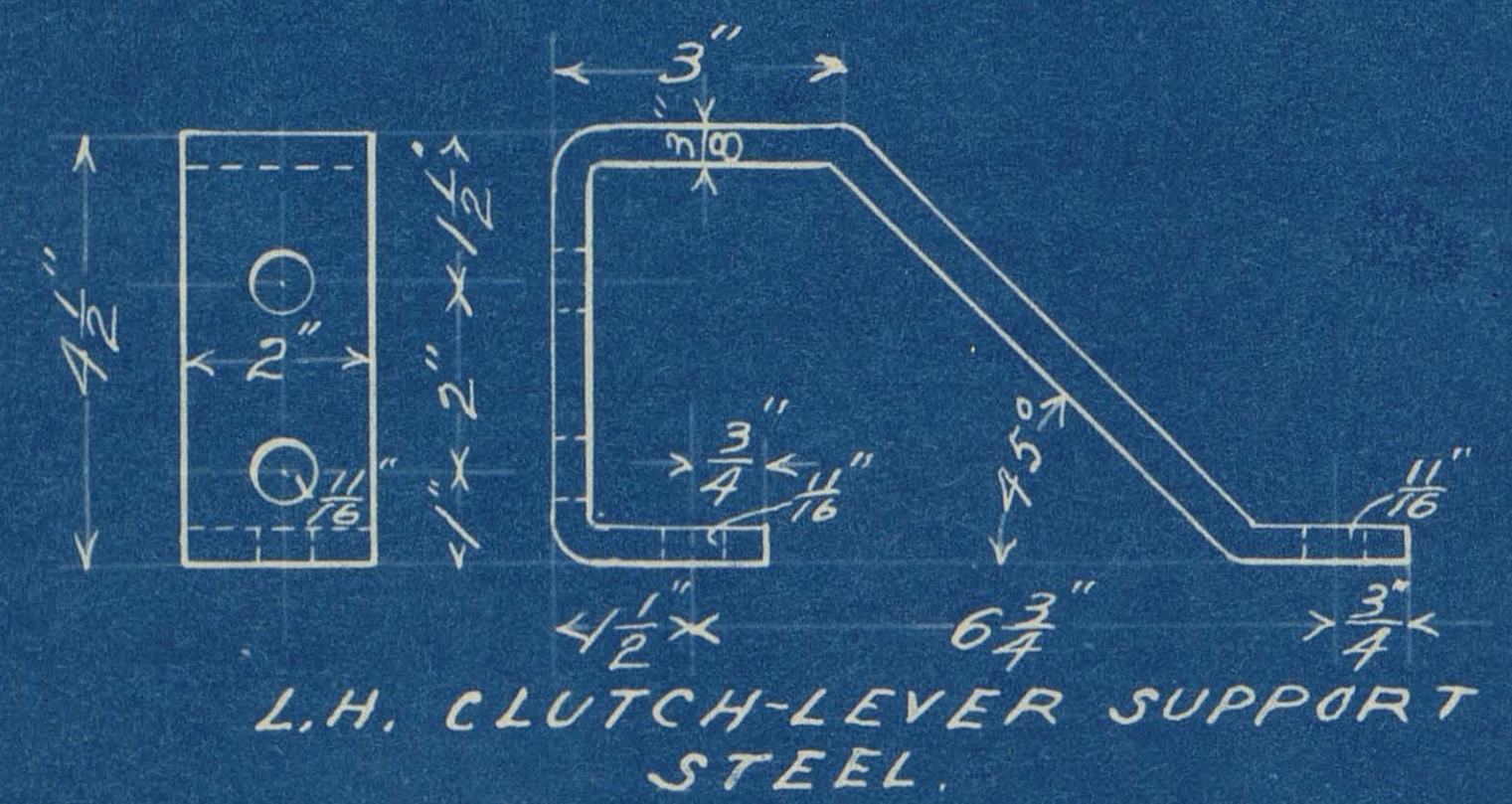




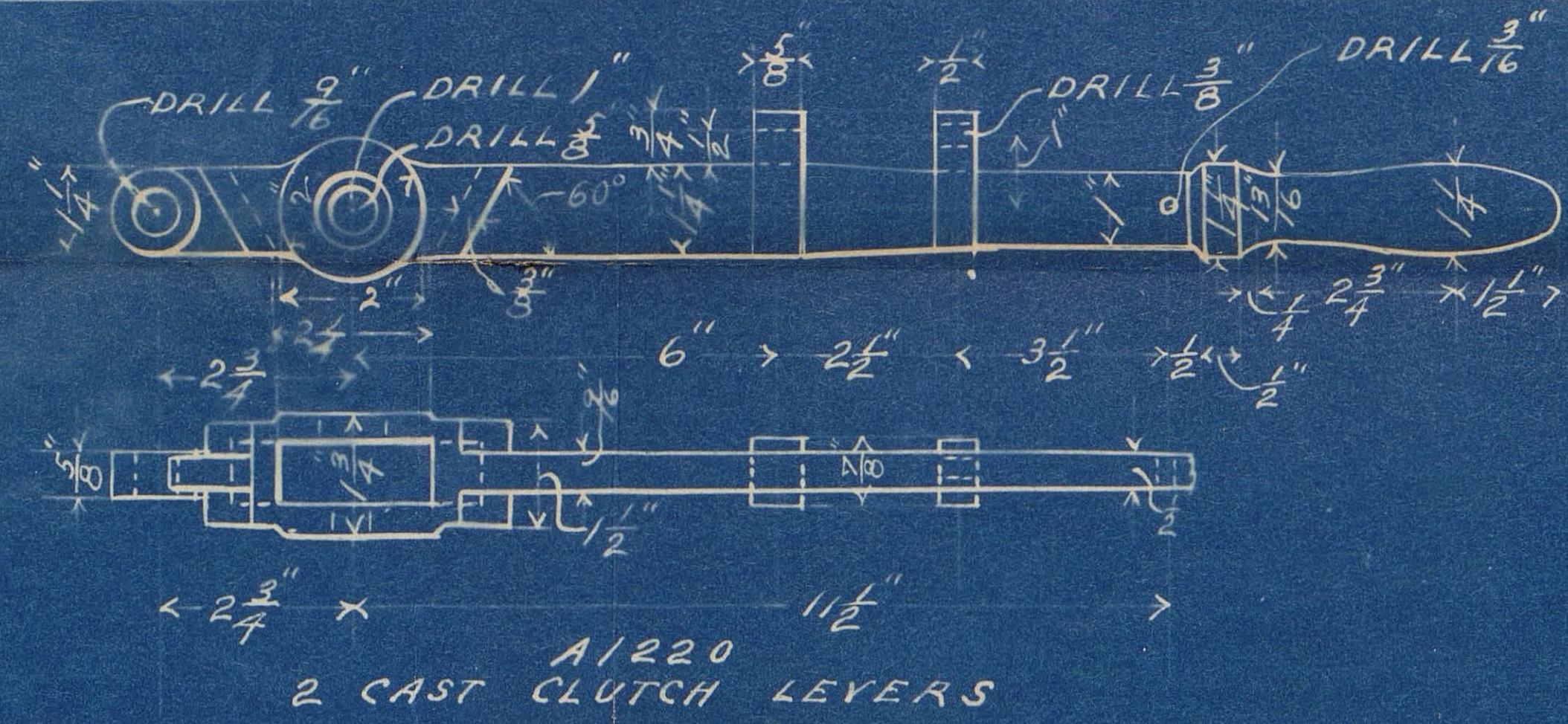




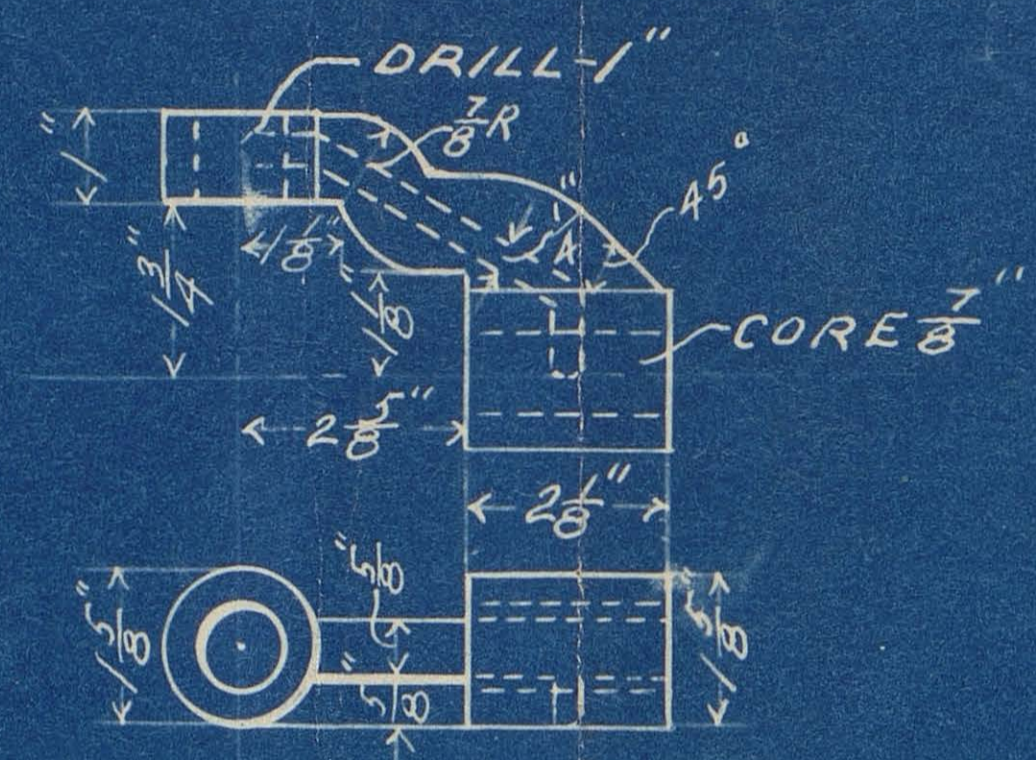
A1218  
2 CLUTCH-LEVER QUADRANTS  
GRAY CAST'GS.



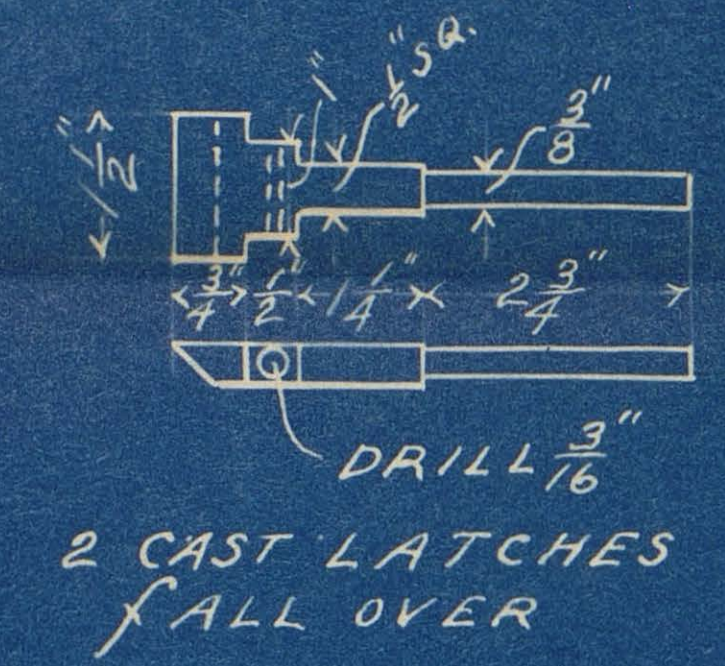
L.H. CLUTCH-LEVER SUPPORT  
STEEL.



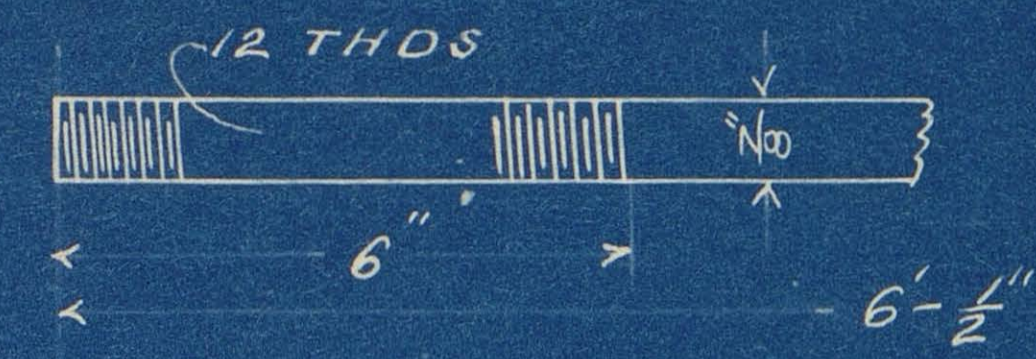
A1220  
2 CAST CLUTCH LEVERS



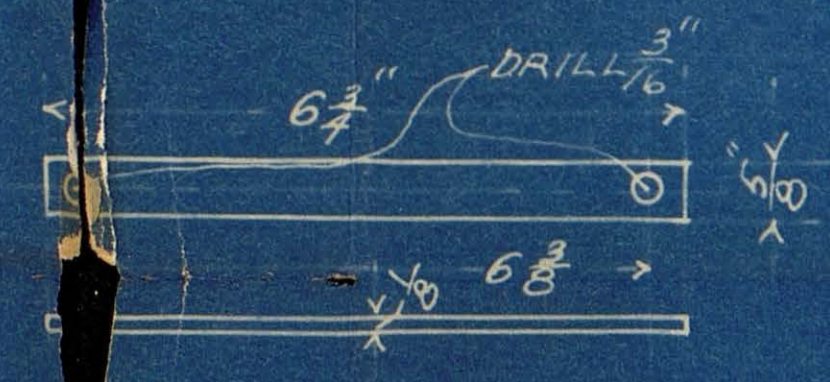
Z837  
2 CLUTCH-ROD ENDS  
CAST IRON



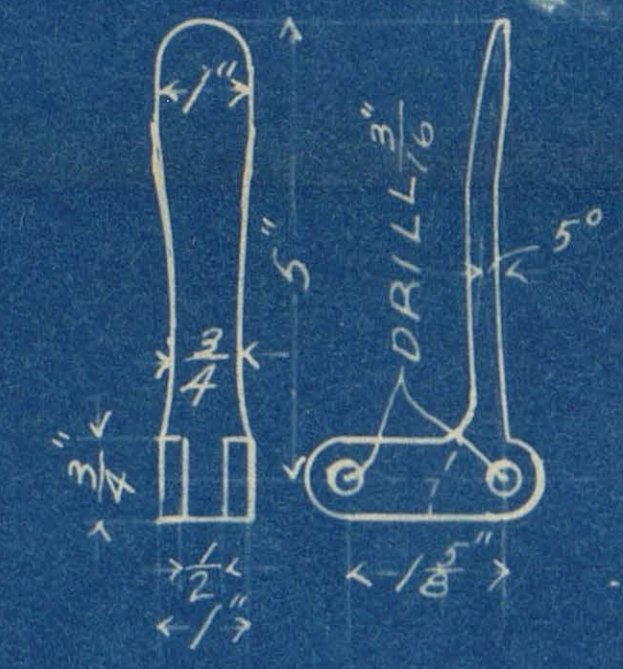
2 CAST LATCHES  
FALL OVER



2 CLUTCH LEVER ROD'S  
MACH STEEL  
4-7/8 HEX NUTS REQ'D.



4 STEEL SIDE  
LINKS FOR LATCHES



2 CAST IRON  
LATCH LIFTERS.

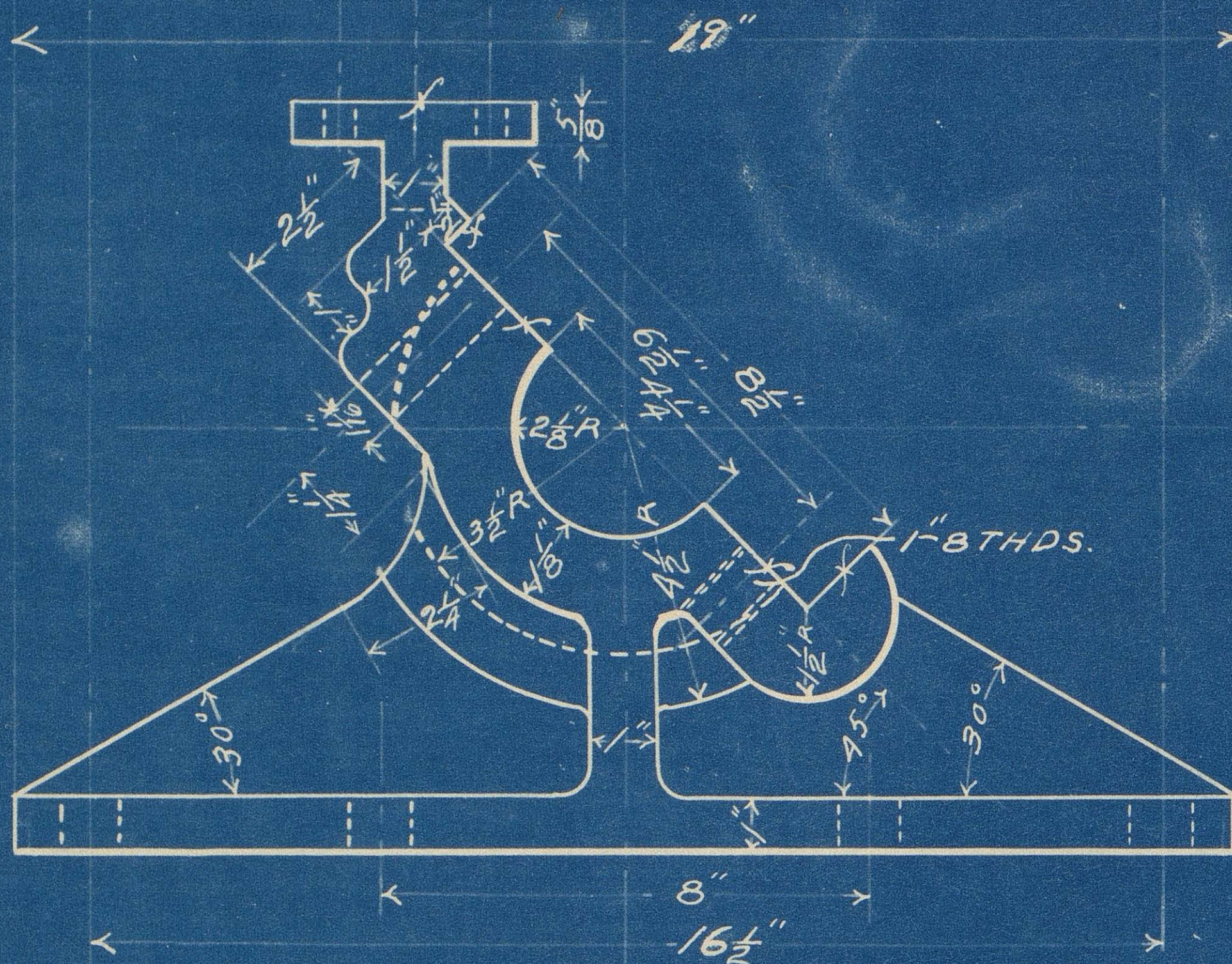
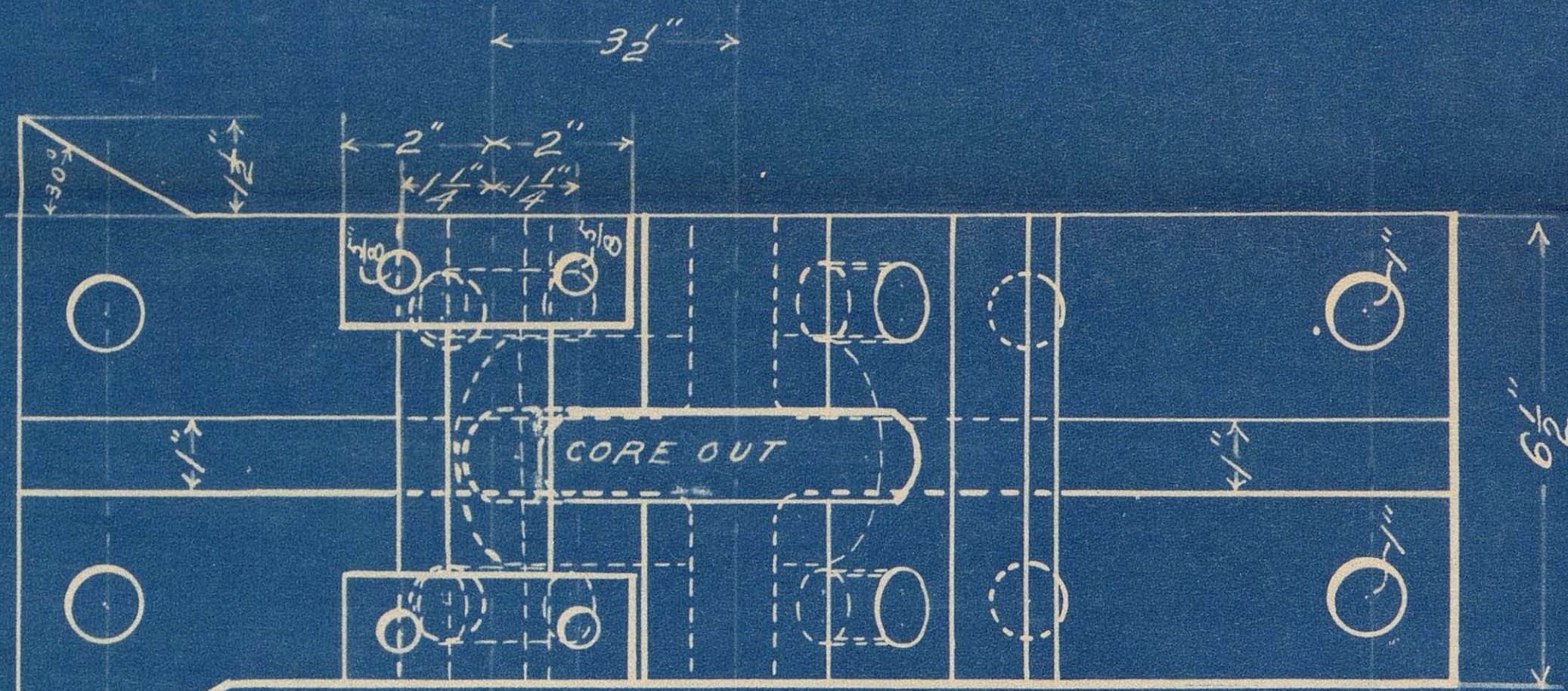
DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE 1-3"=1 FT. f MEANS FINISH  
DEPARTMENT OF  
MECHANICAL ENGINEERING  
K.S.A.C.

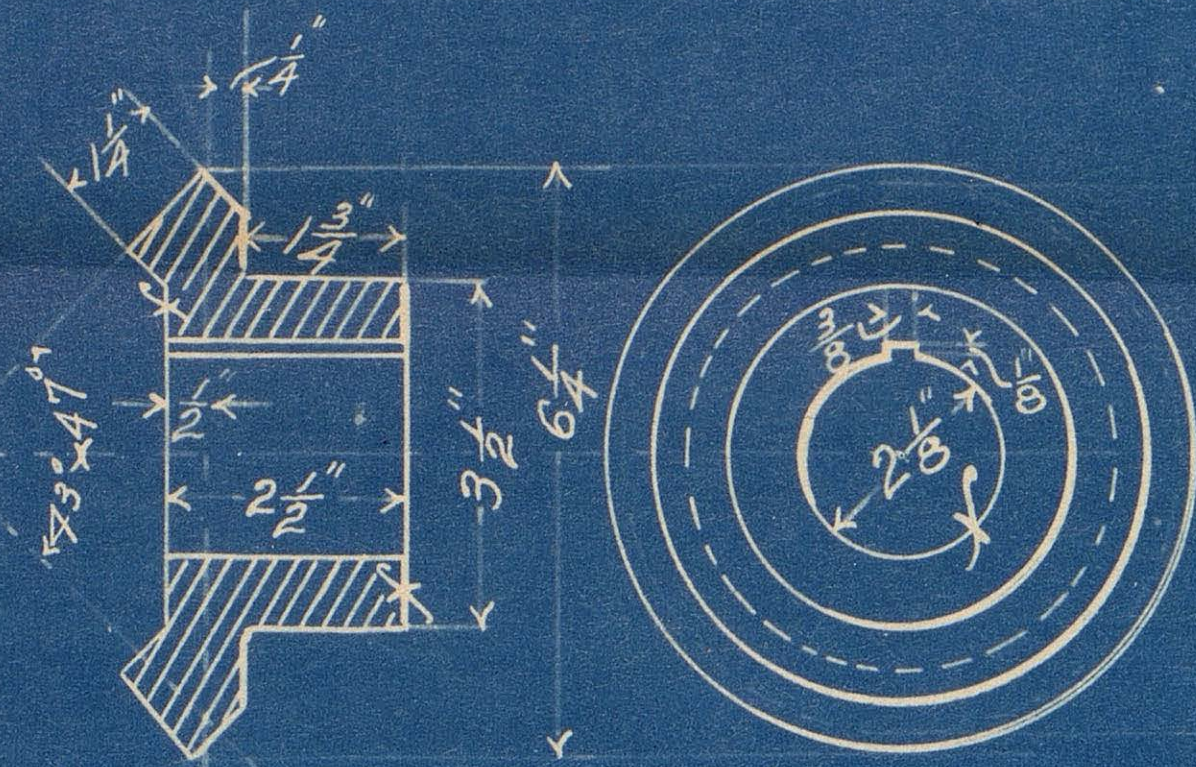




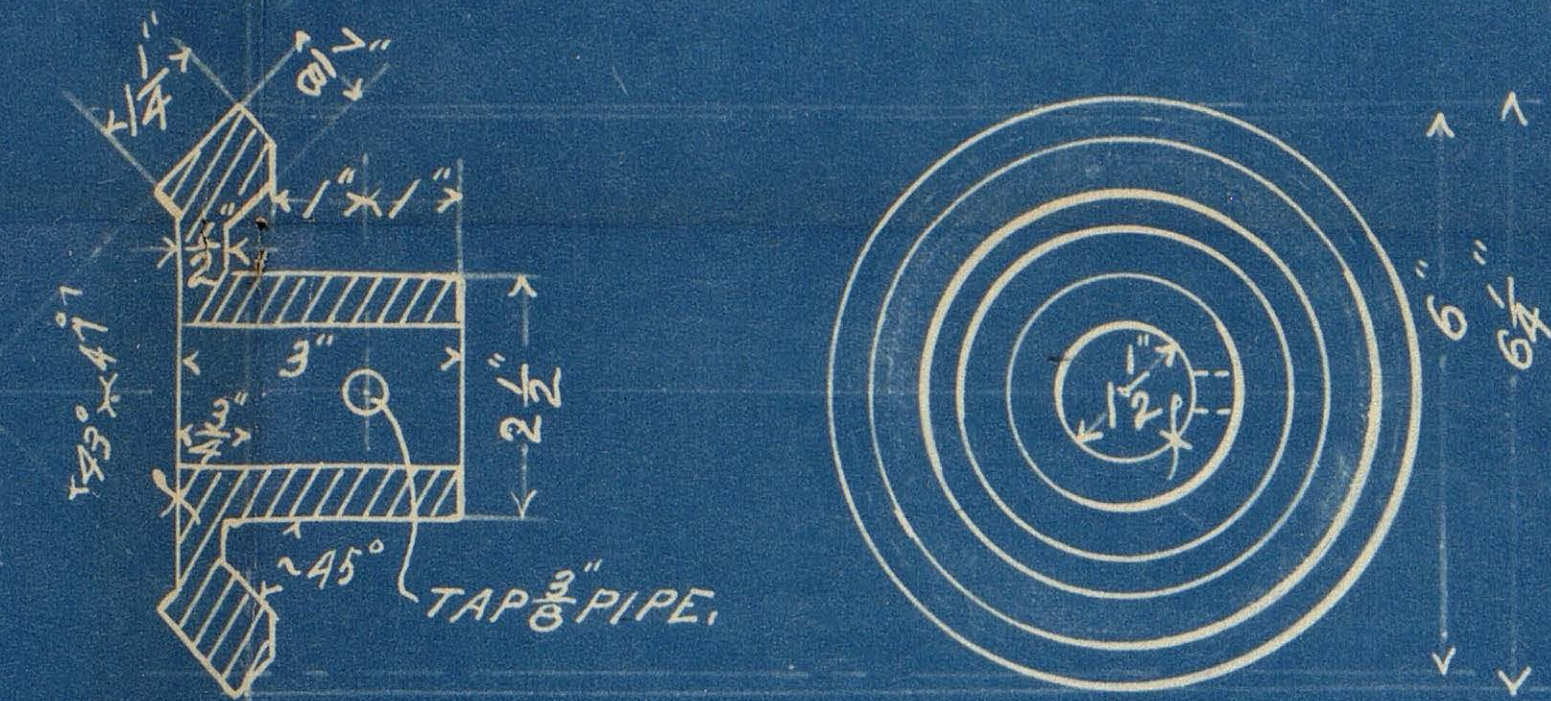




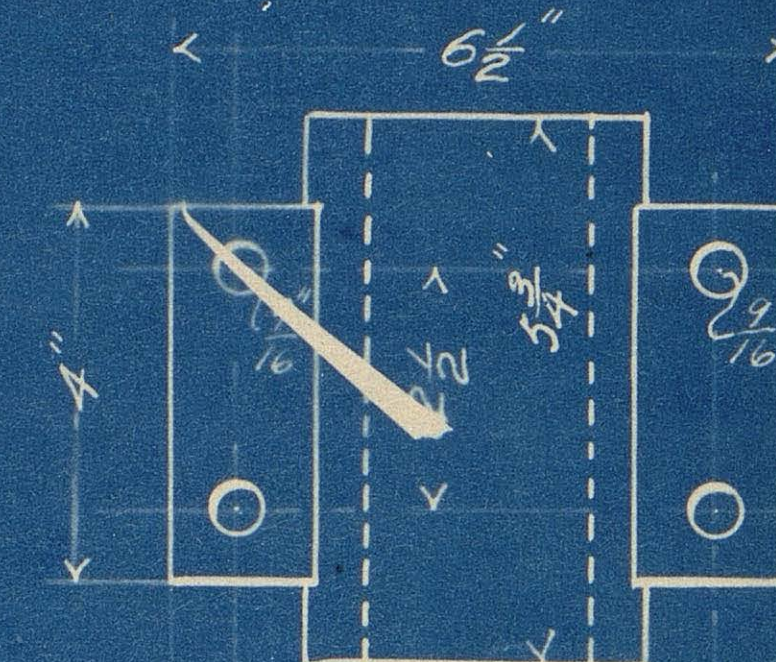
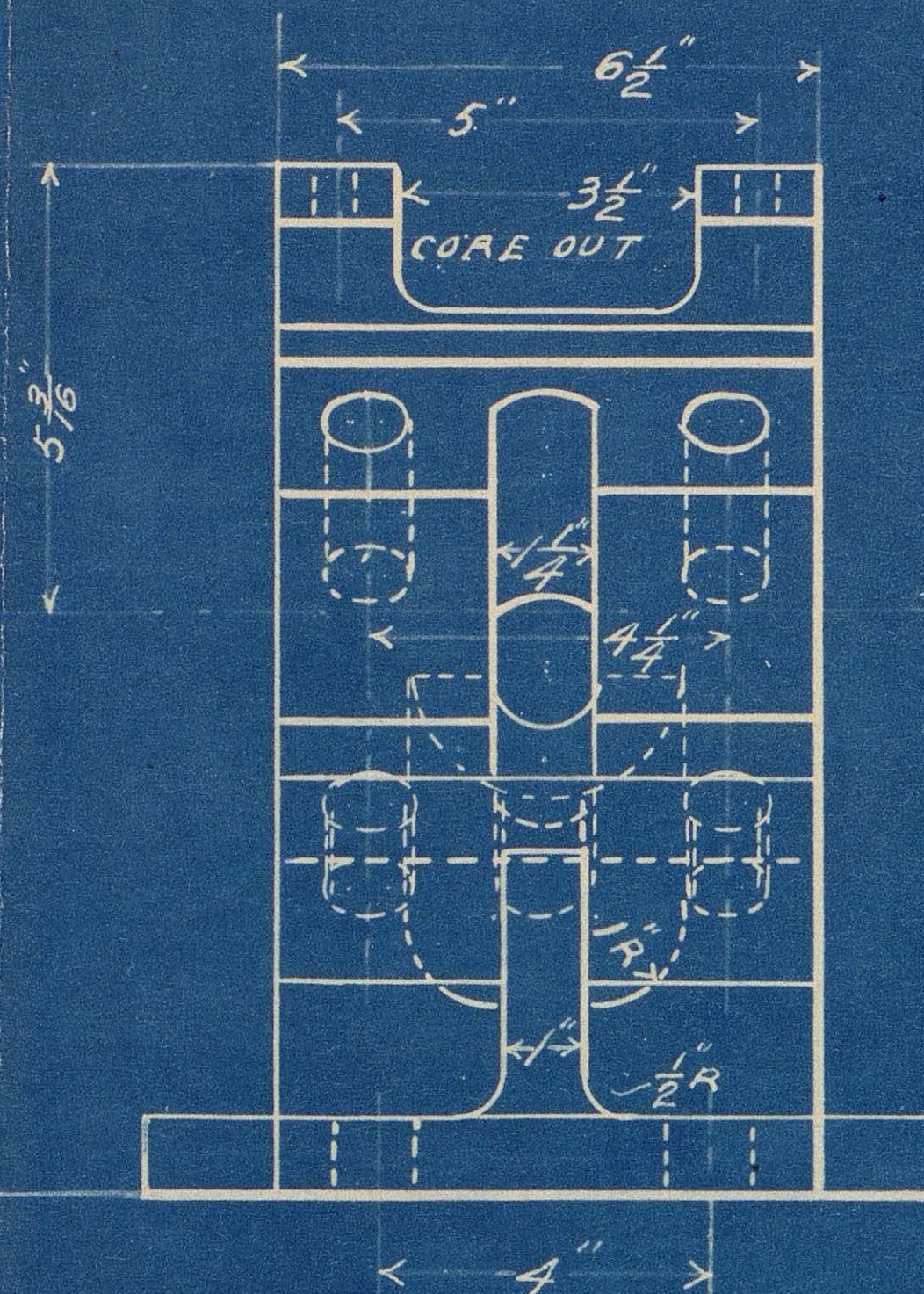
CENTER CRANK-BOX.  
CORE FOR SPIRAL GEAR



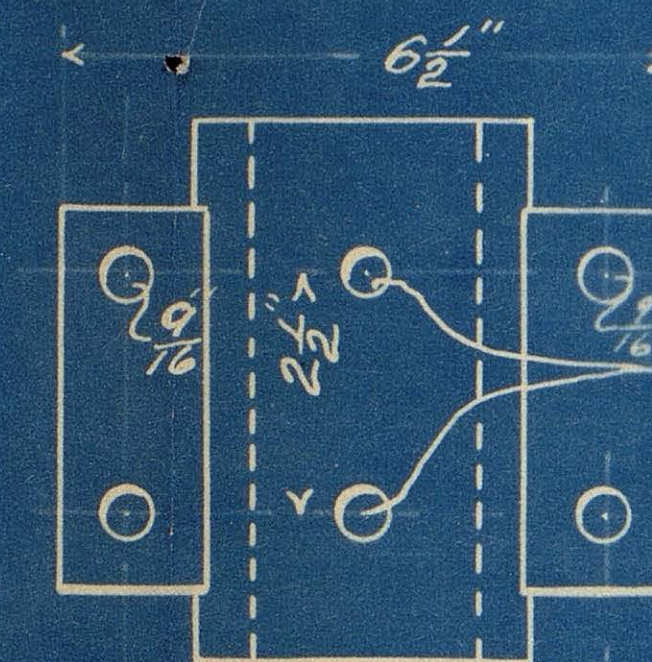
MITER GEAR.  
6-P. 36 TEETH.



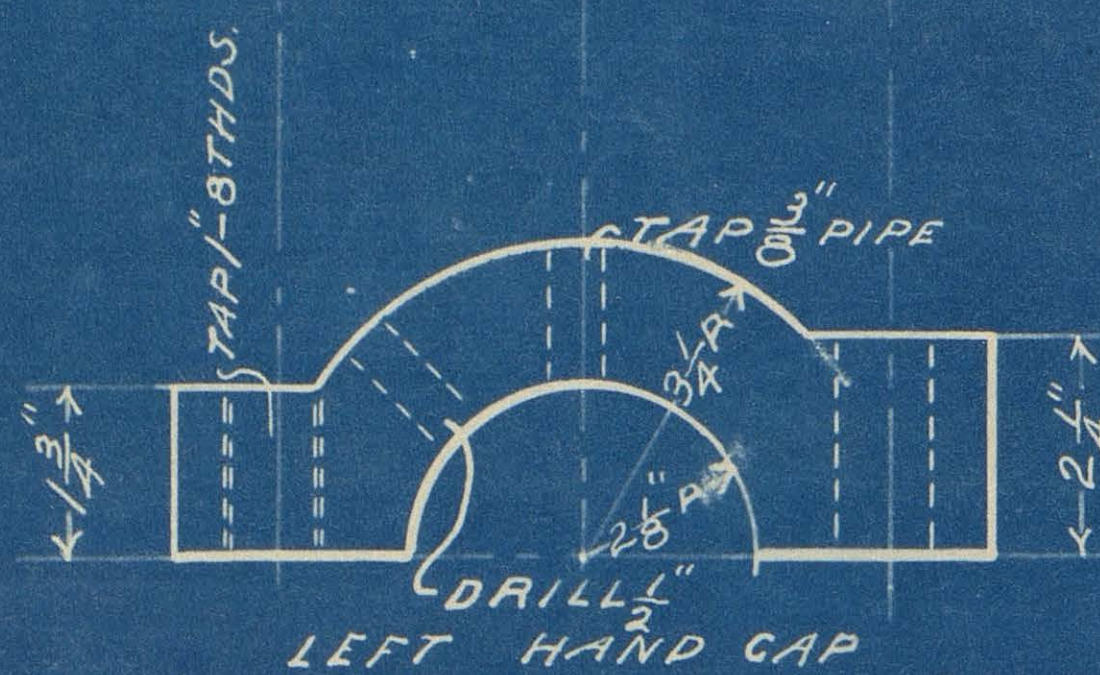
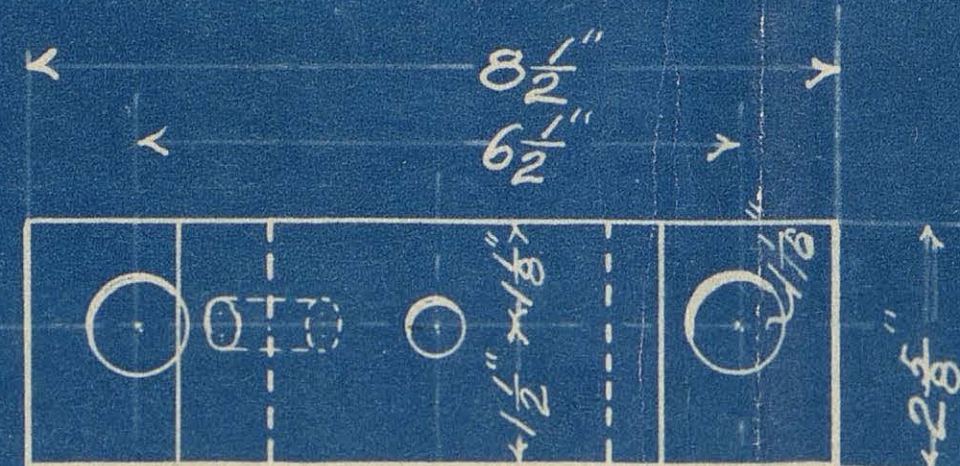
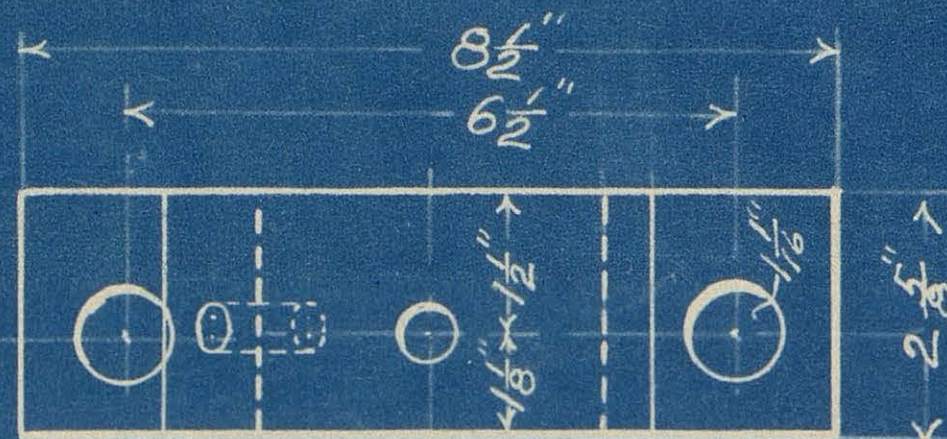
MITER GEAR  
6-P. 36 TEETH  
BUSH TO 1 1/4" WITH CAST BRASS.



BOTTOM HALF  
BOX FOR CAM SHAFT.



TOP HALF  
BOX FOR CAM SHAFT.

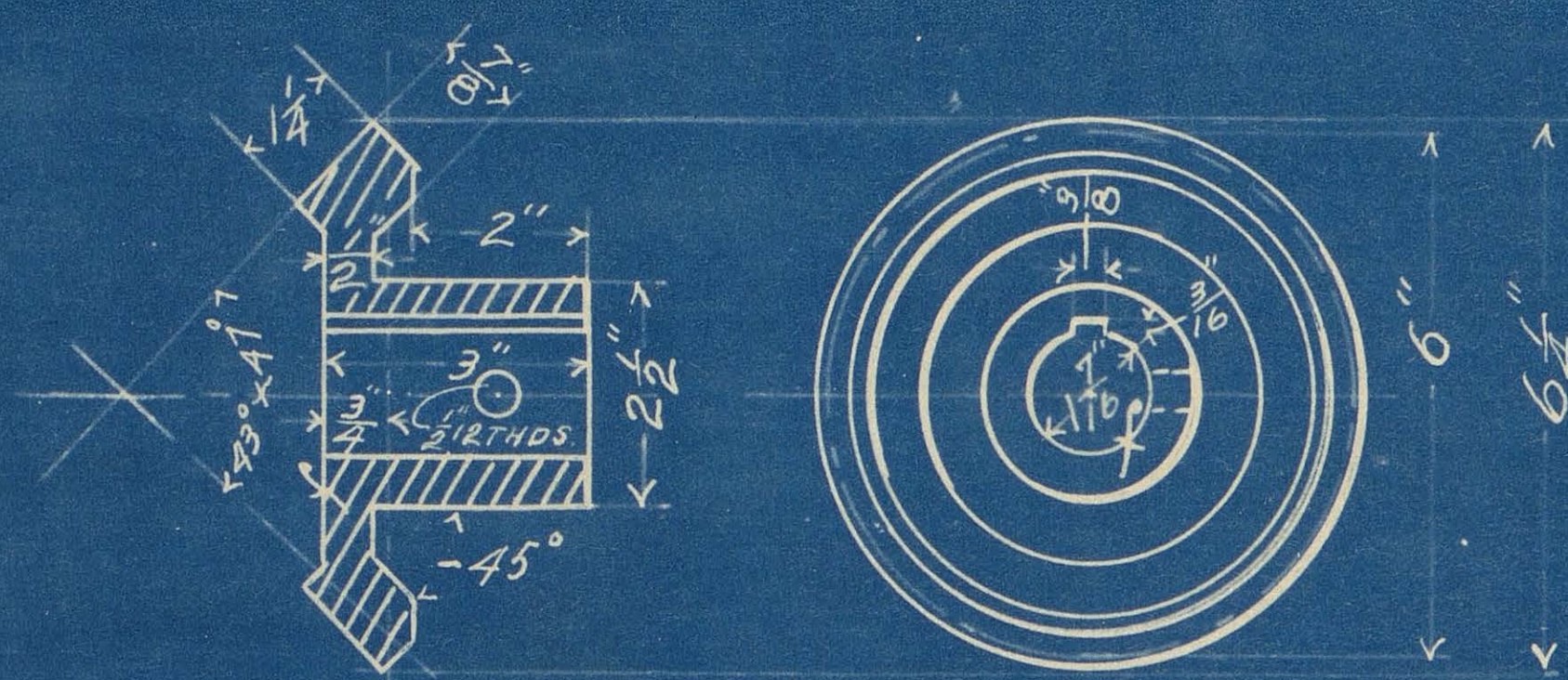


LEFT HAND CAP



RIGHT HAND CAP.

CENTER BOX CAPS,  
4-CAP SCREWS, 4x1"-8THDS.



MITER GEAR  
6-P. 36 TEETH.

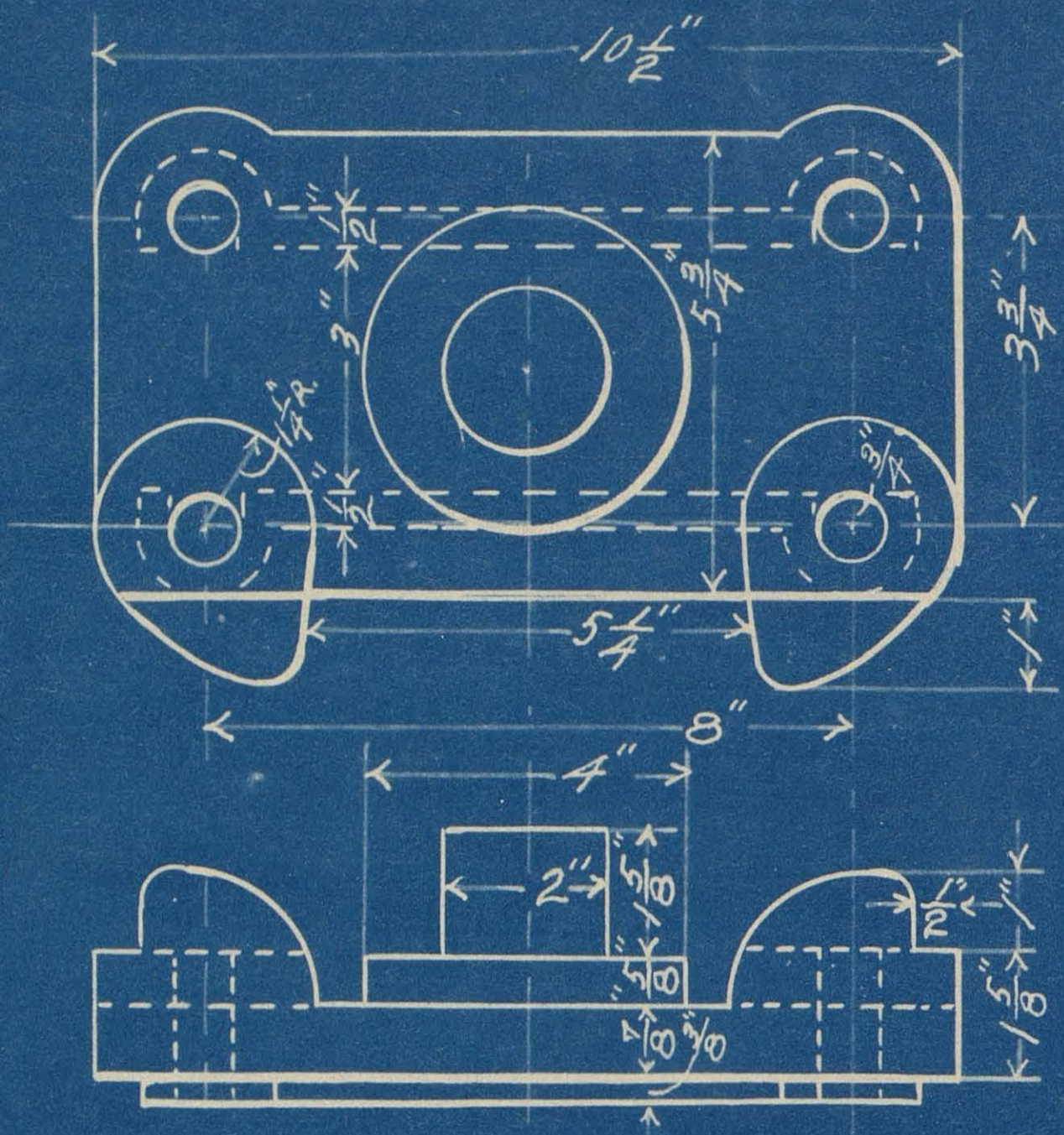
DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: 3"=1'

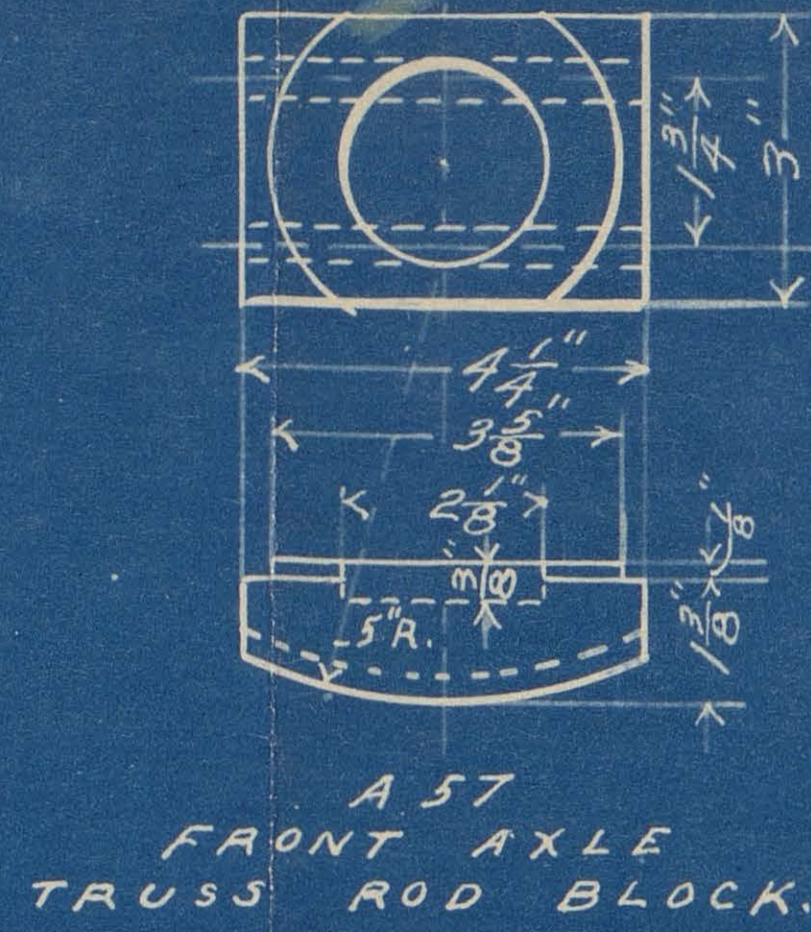
f MEANS FINISH.

DEPARTMENT OF  
MECHANICAL ENGINEERING  
K.S.A.C.

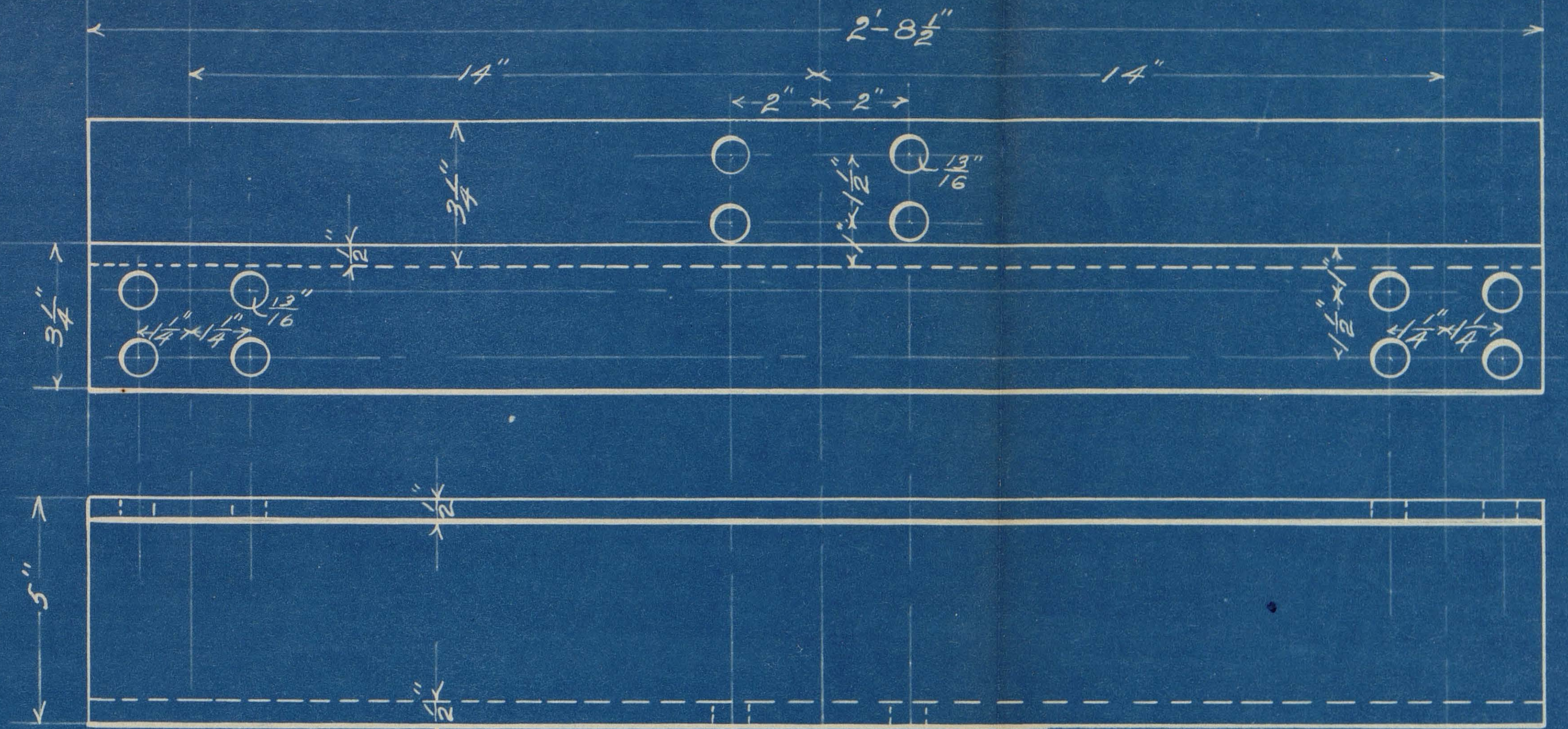




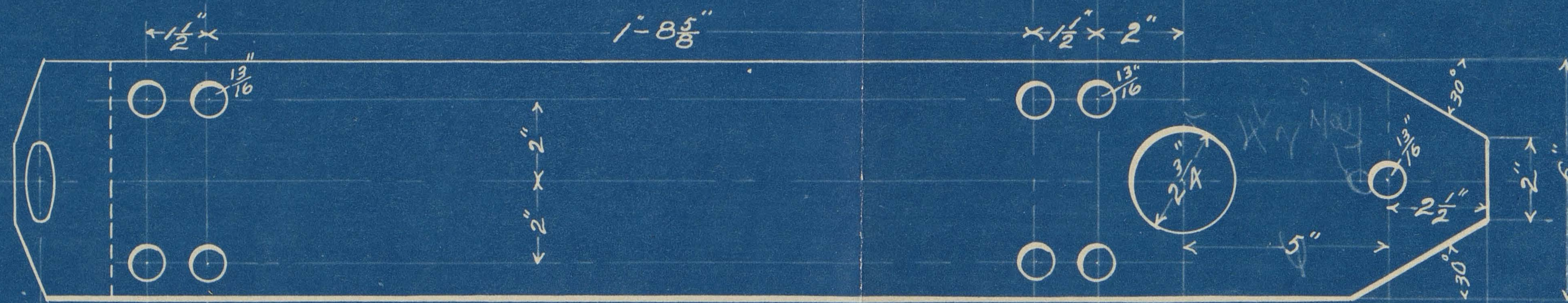
ZZ 22  
FRONT AXLE CASTING.



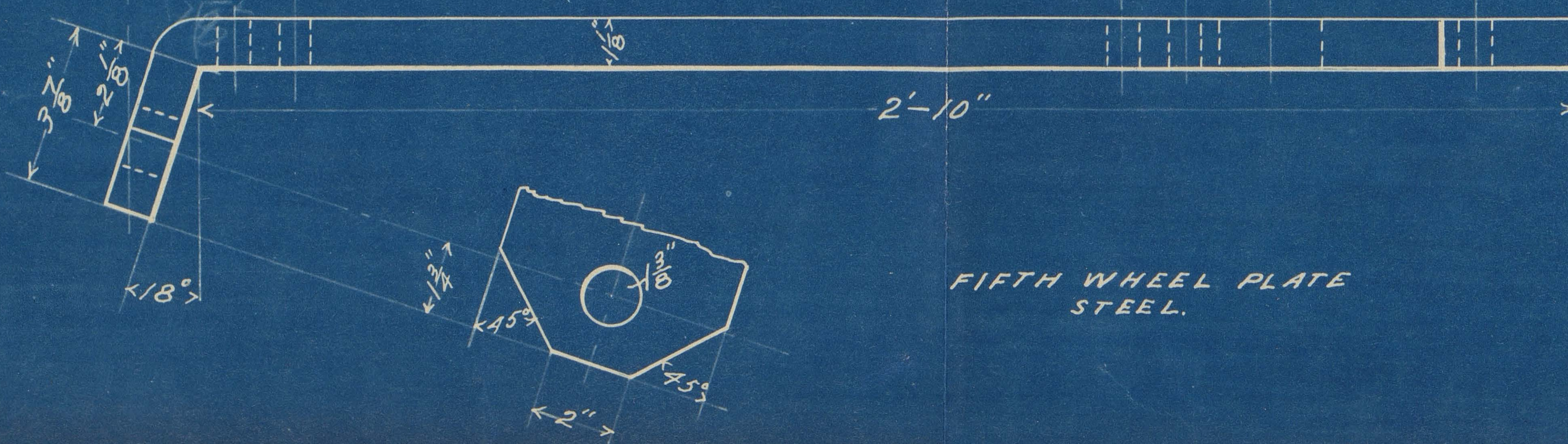
A 57  
FRONT AXLE  
TRUSS ROD BLOCK.



FRONT END SUPPORTS  
Z BARS  
2 REQ'D.



FIFTH WHEEL PLATE  
STEEL.

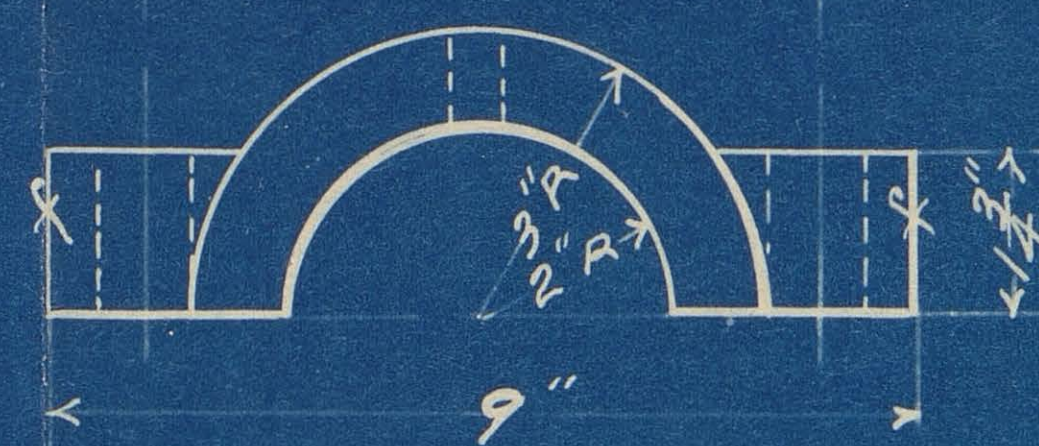
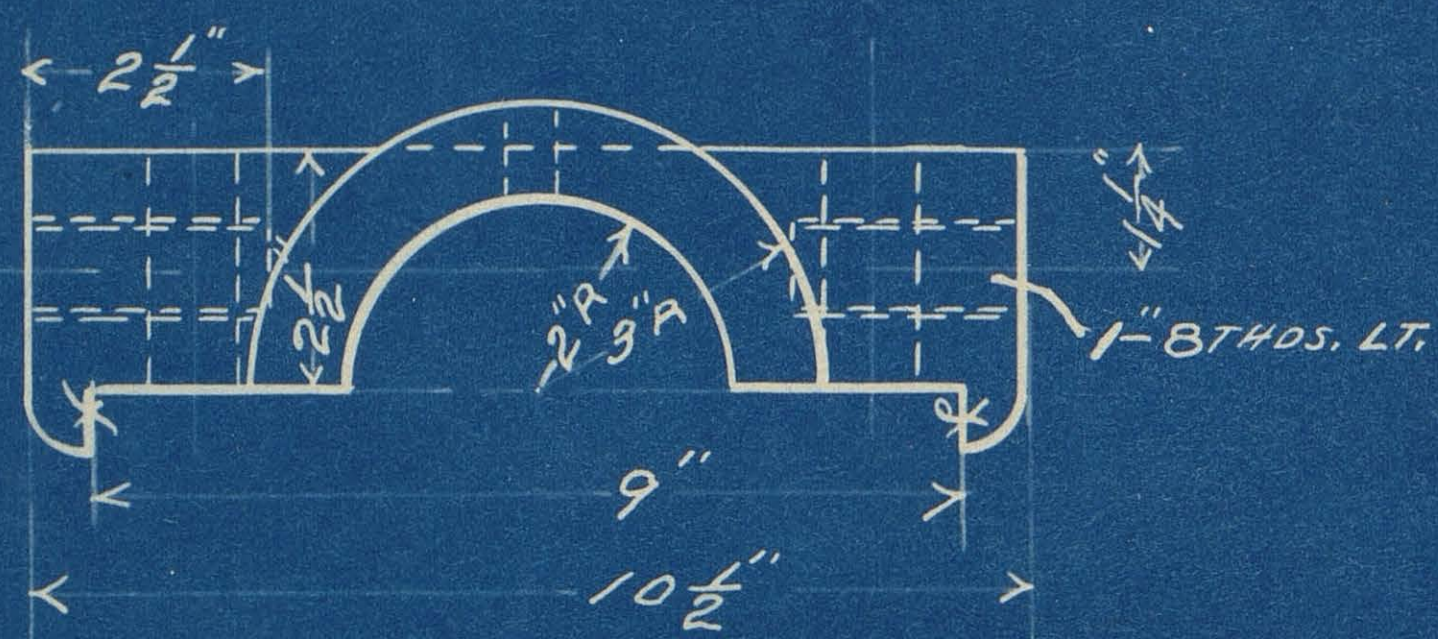
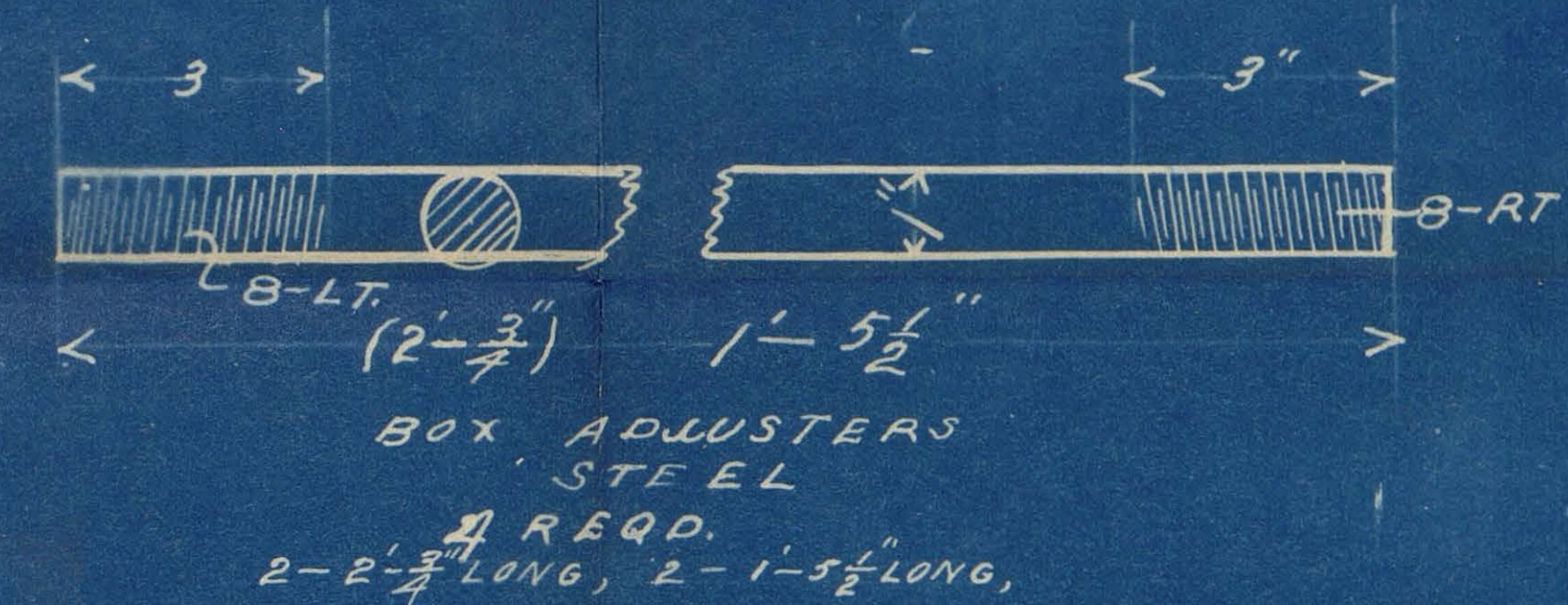
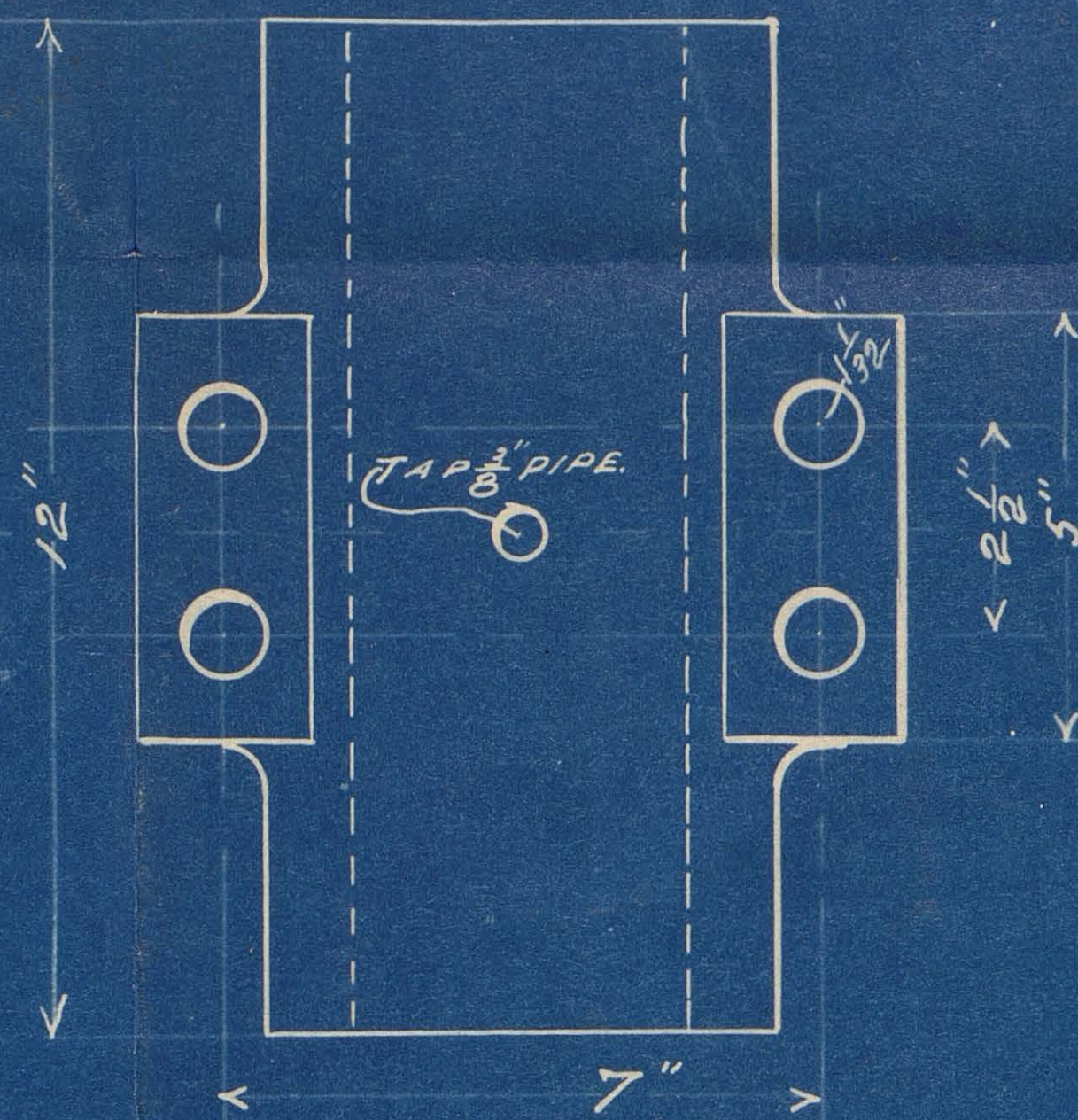
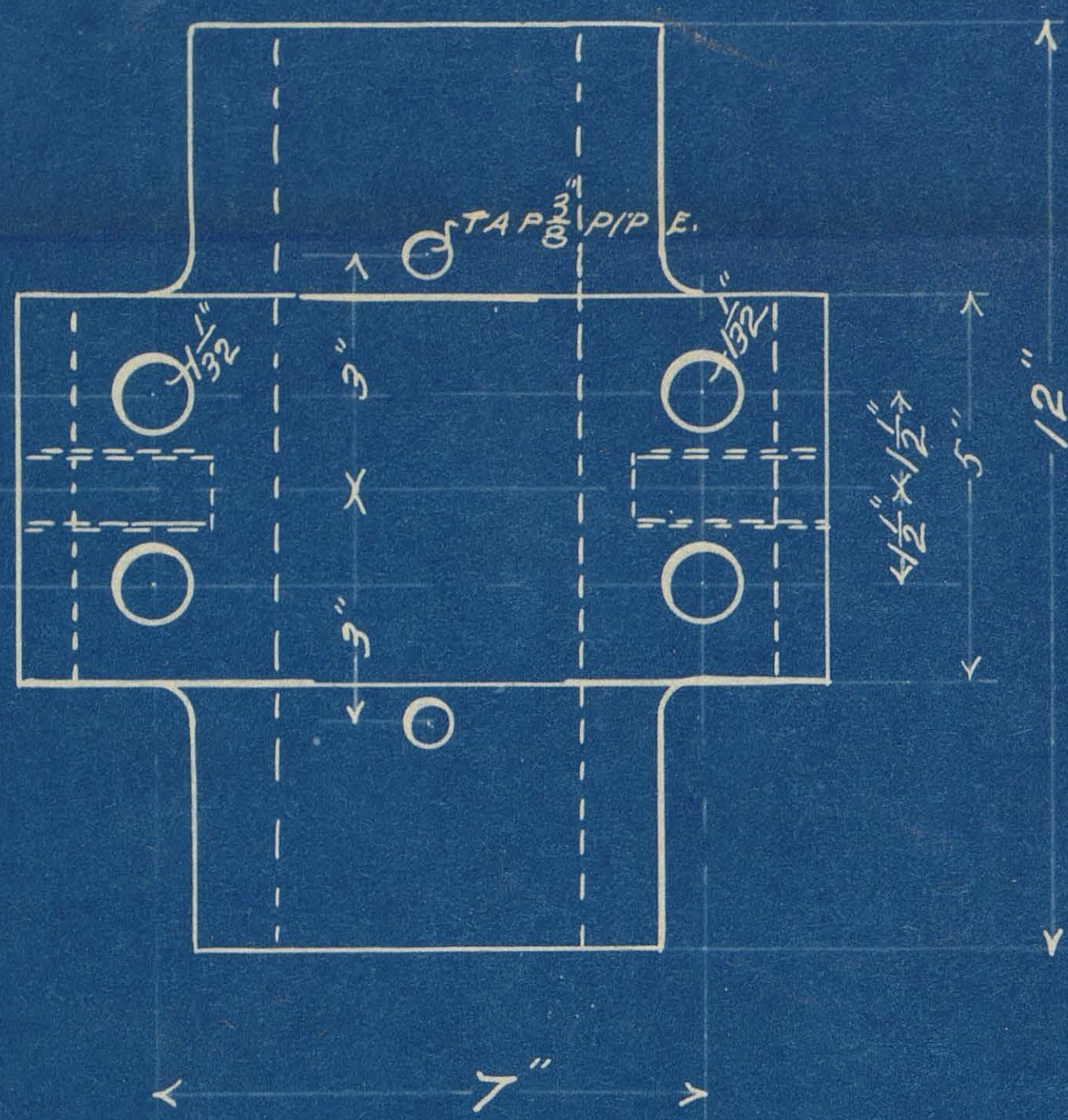


DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: 1" = 3" = 1 FT. f MEANS FINISH.

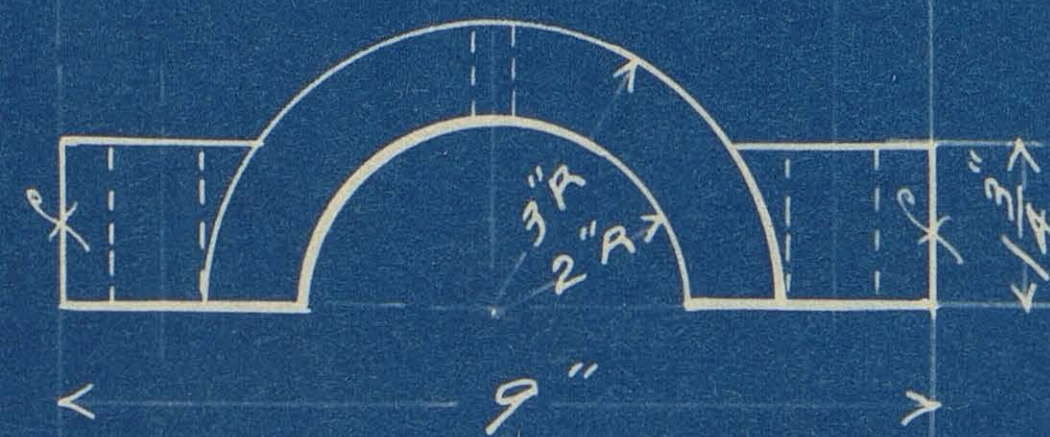
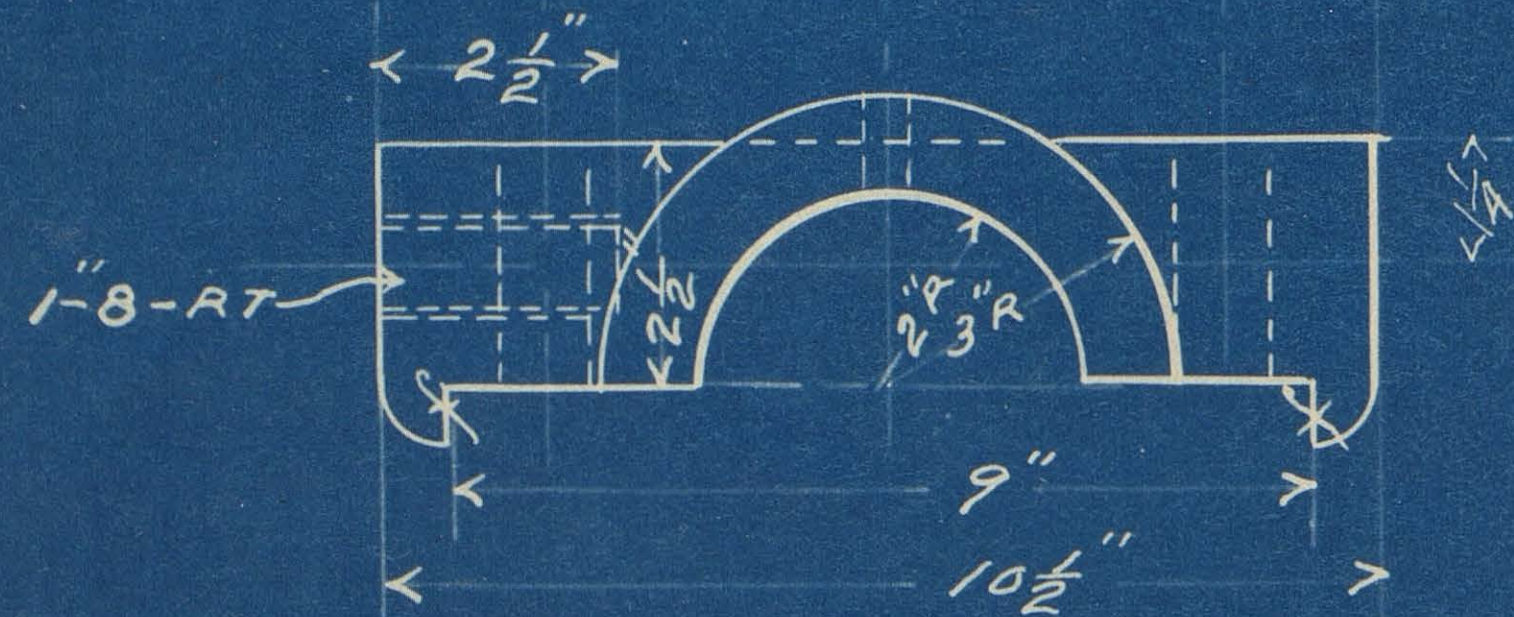
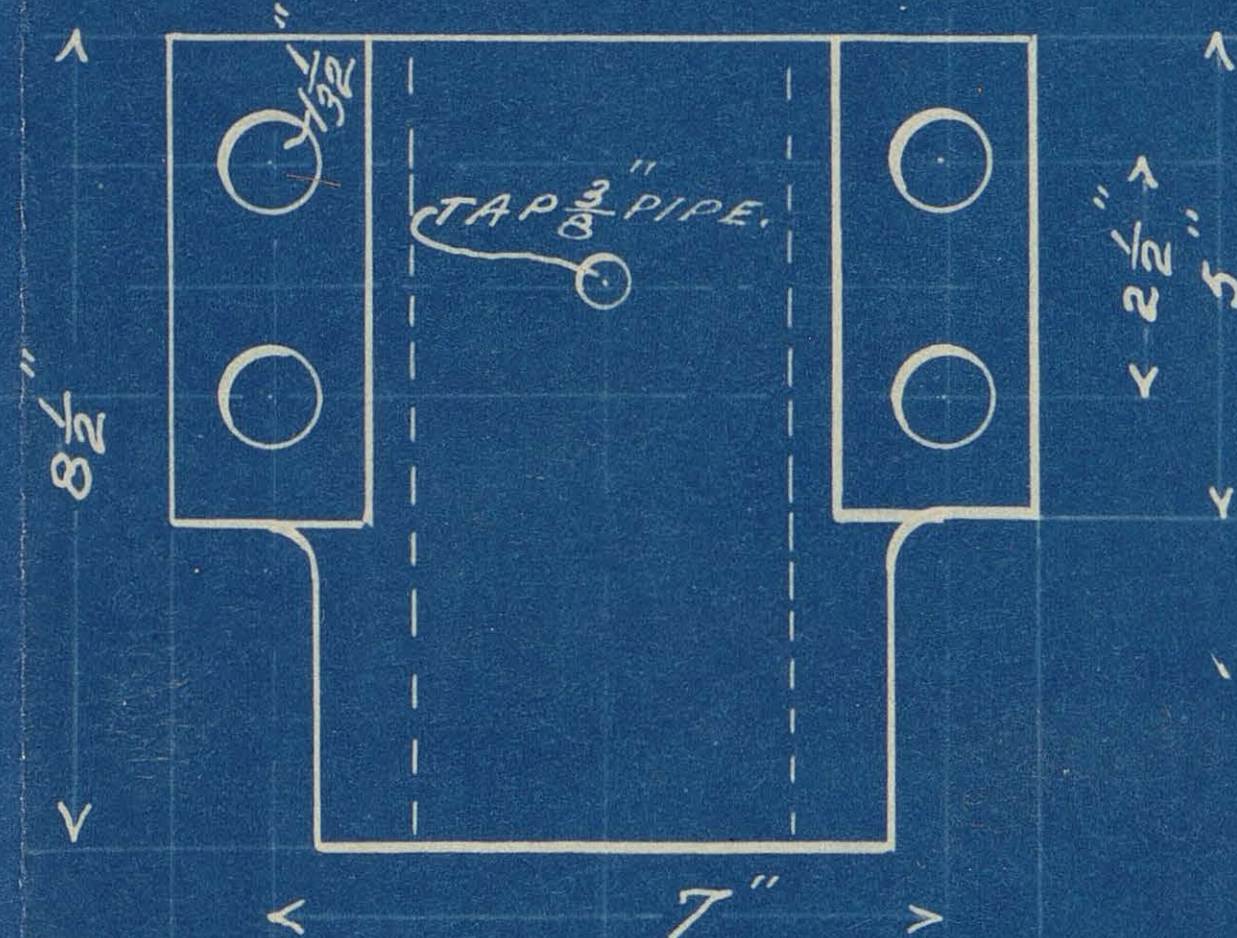
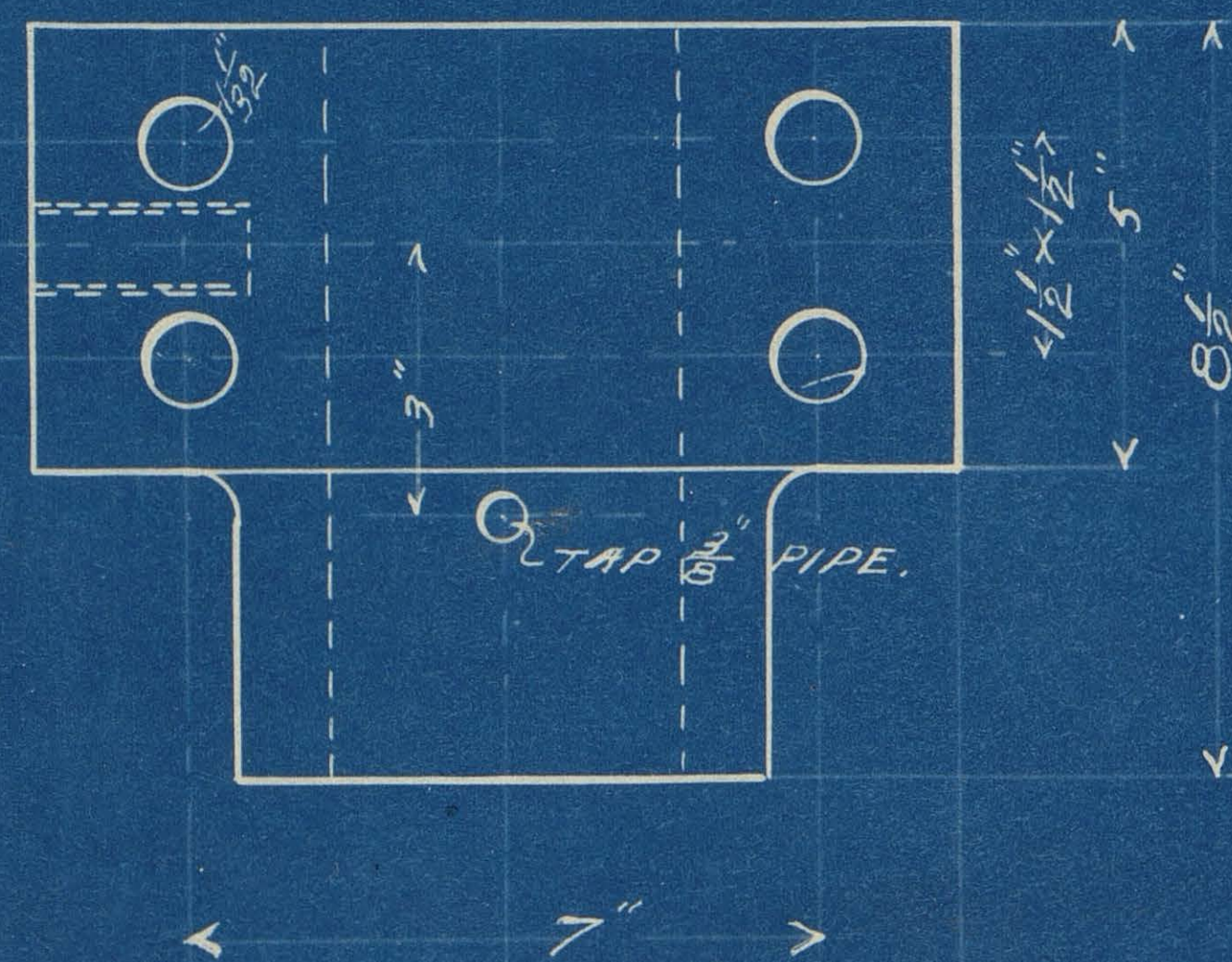
DEPARTMENT OF  
MECHANICAL ENGINEERING  
K. S. A. C.





RT HAND CROSS SHAFT BOX  
BOTTOM HALF.

RT HAND CROSS SHAFT BOX  
TOP HALF.



RT. HAND DRUM SHAFT BOX.  
TOP HALF.

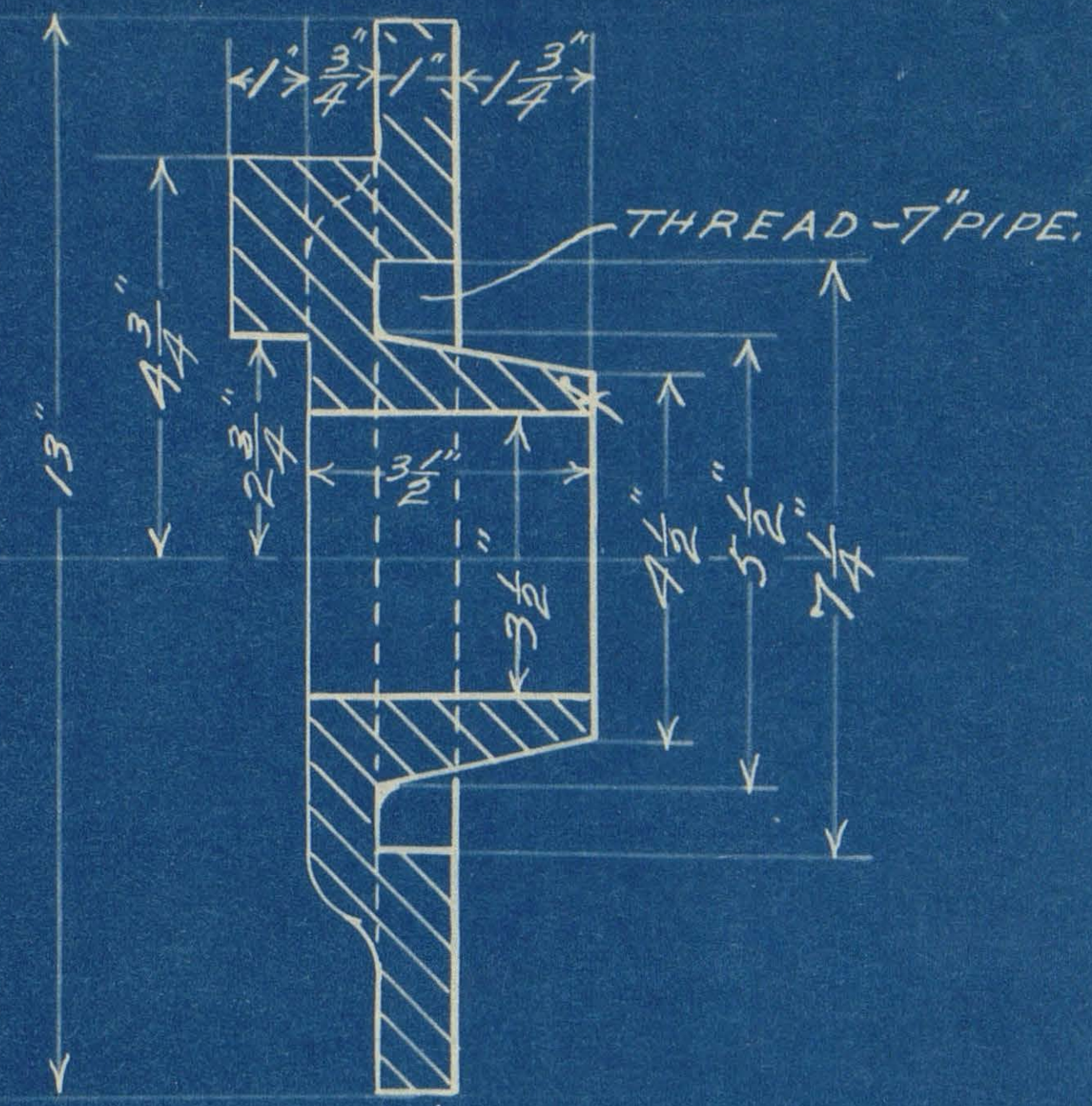
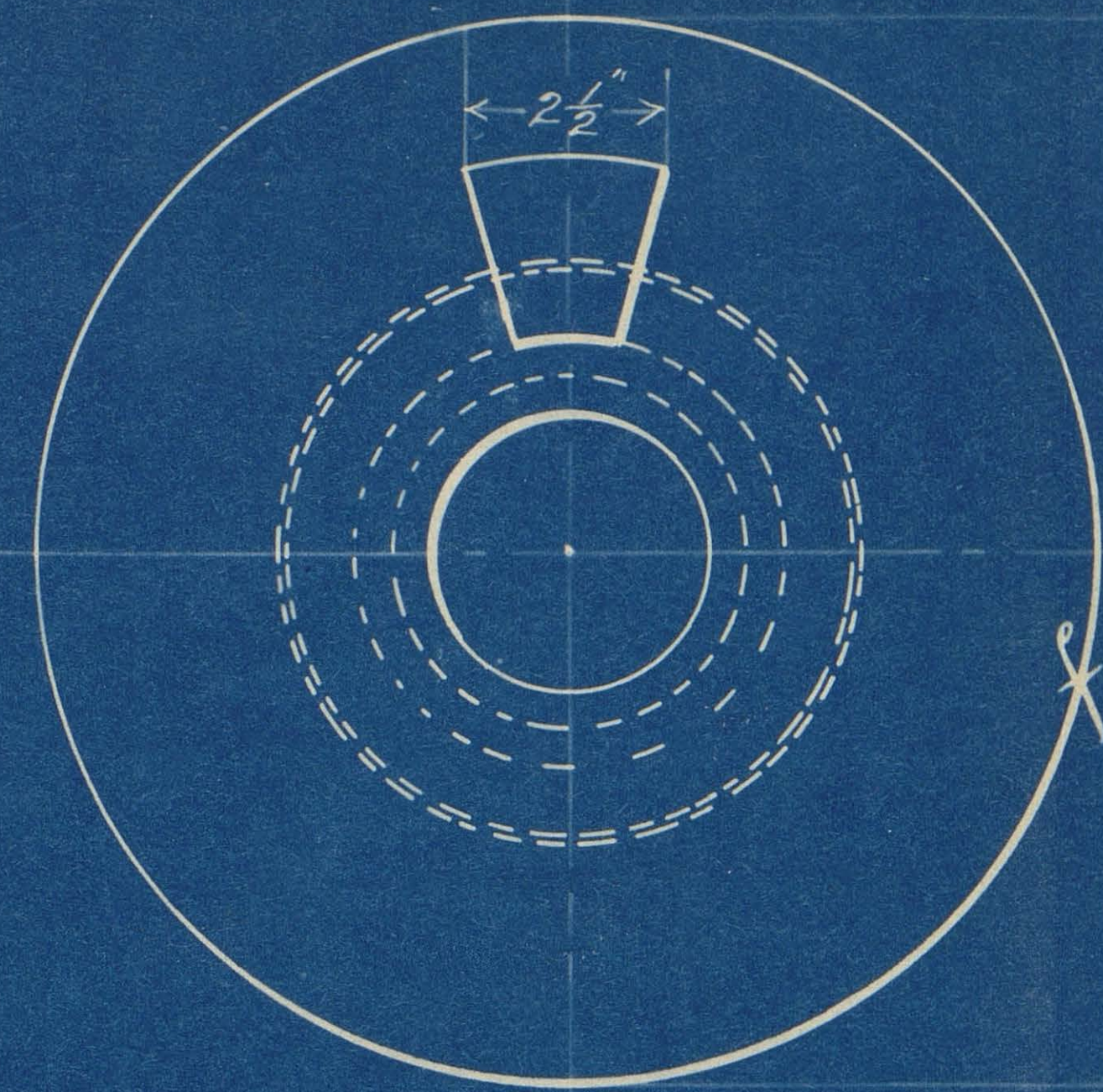
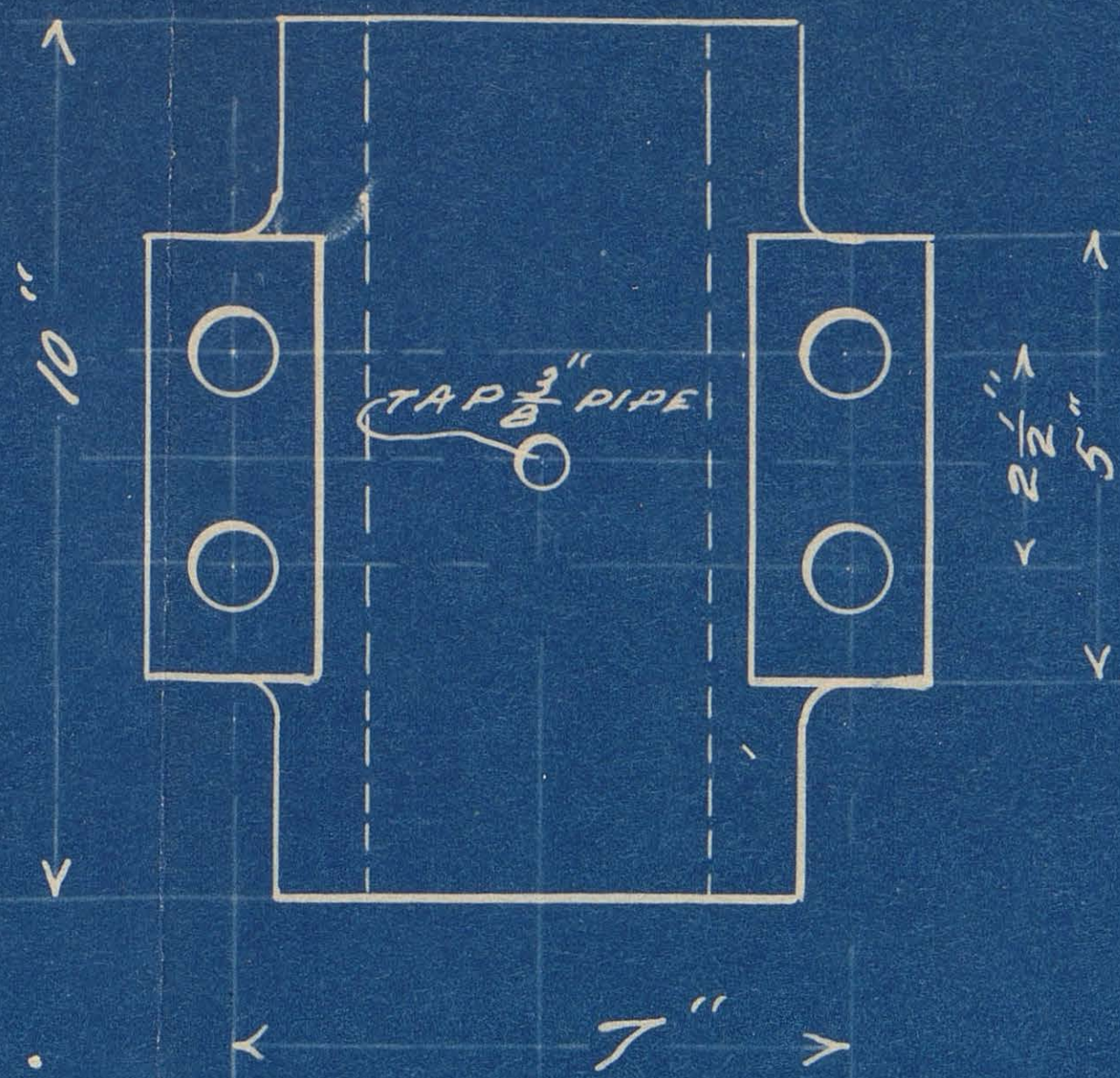
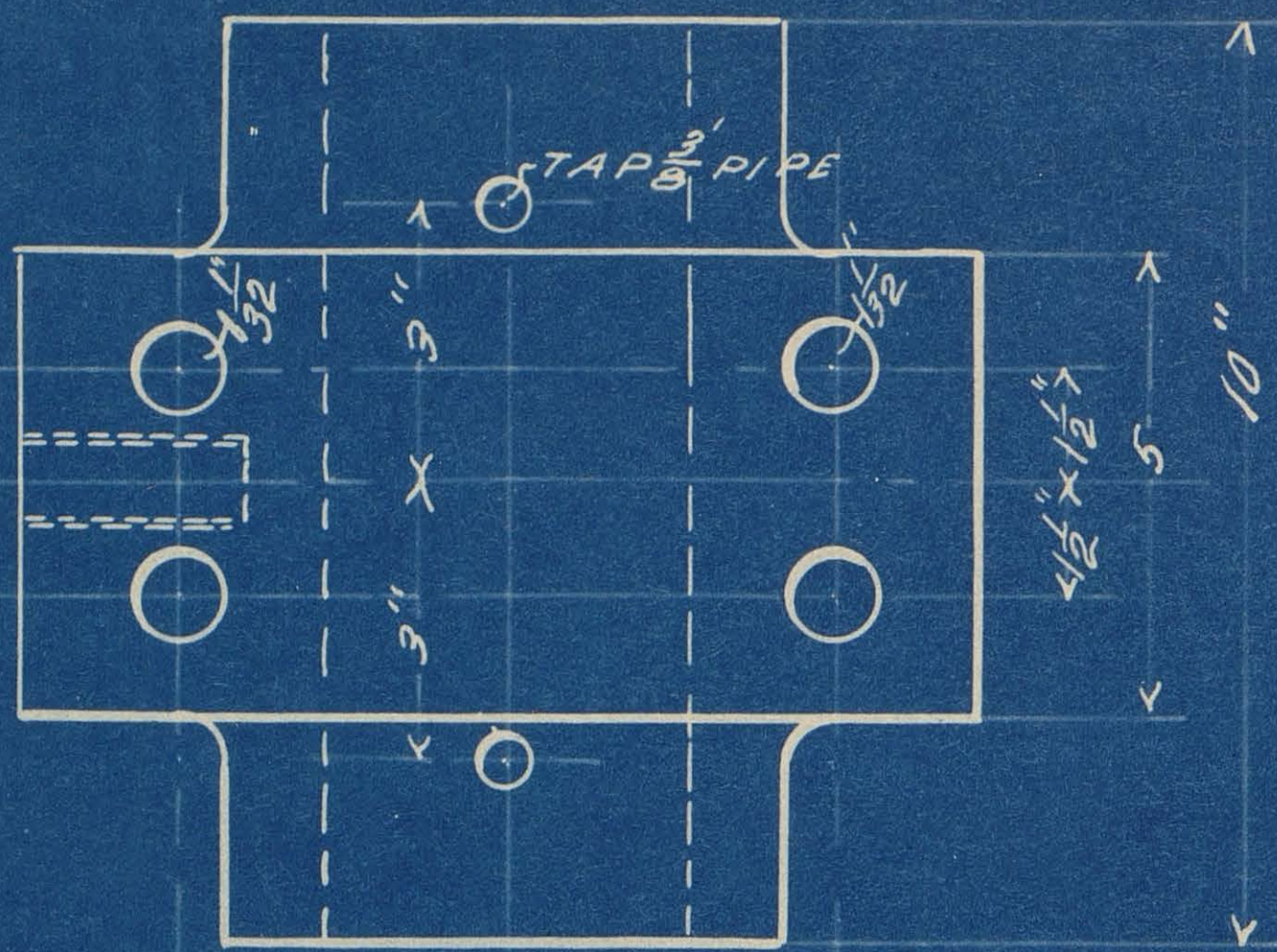
RT HAND DRUM SHAFT BOX  
BOTTOM HALF.

DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

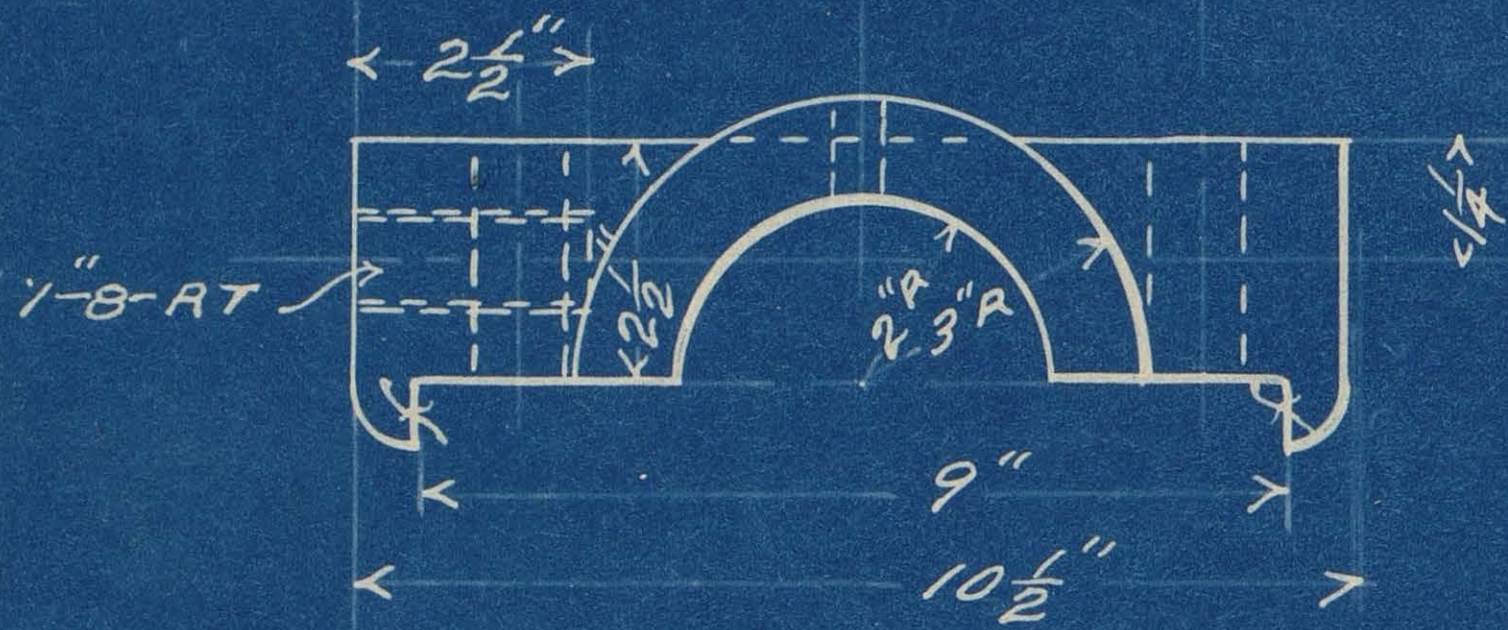
SCALE: 1-3"=1 FT. f MEANS FINISH.

DEPARTMENT OF  
MECHANICAL ENGINEERING  
K. S. A. C.

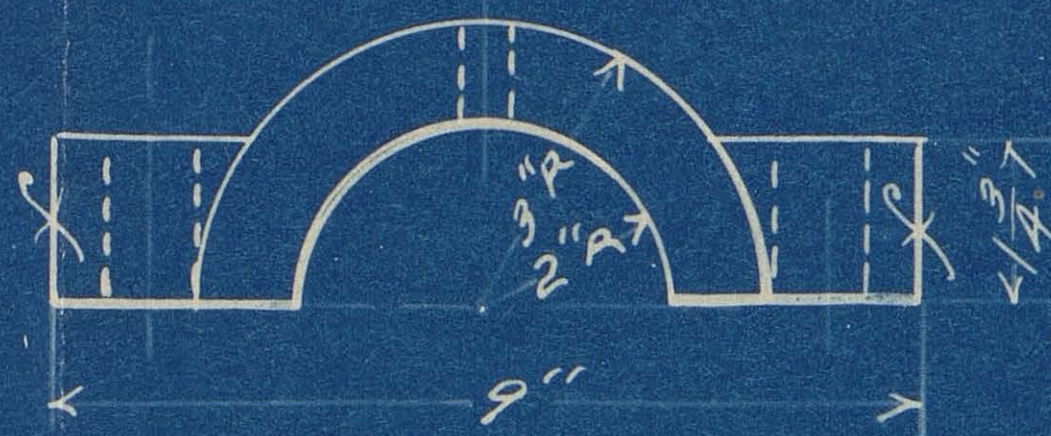




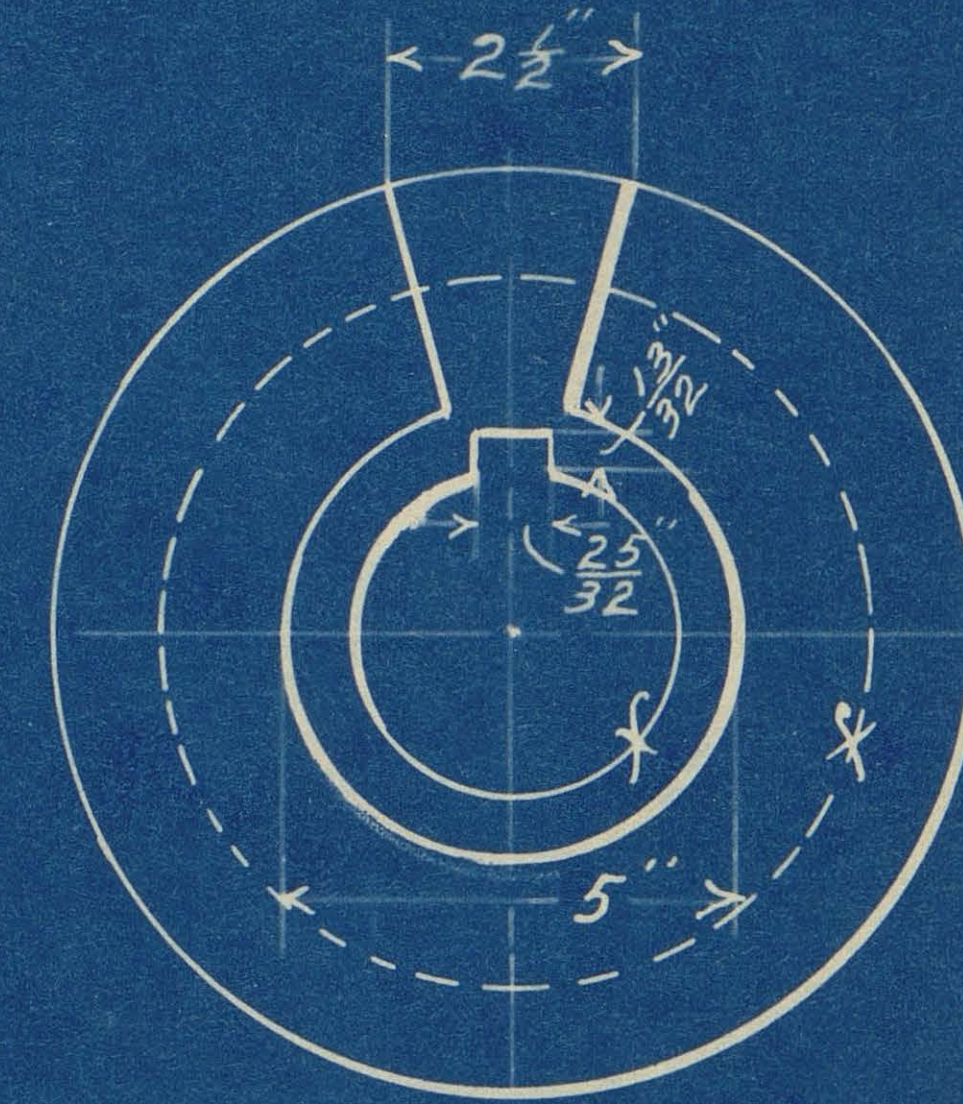
DRUM HEADS  
GRAY CAST 6  
2 REQD



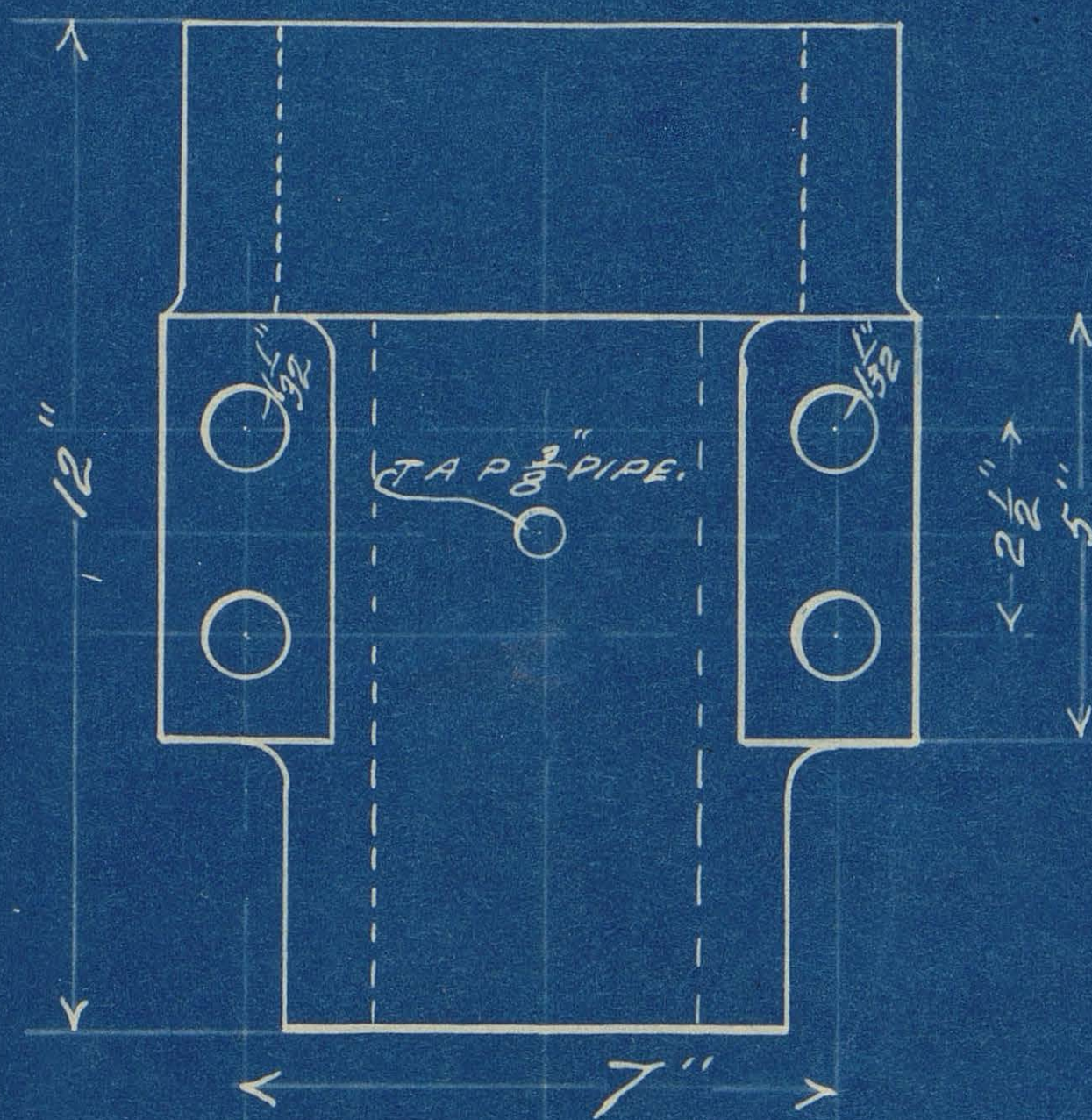
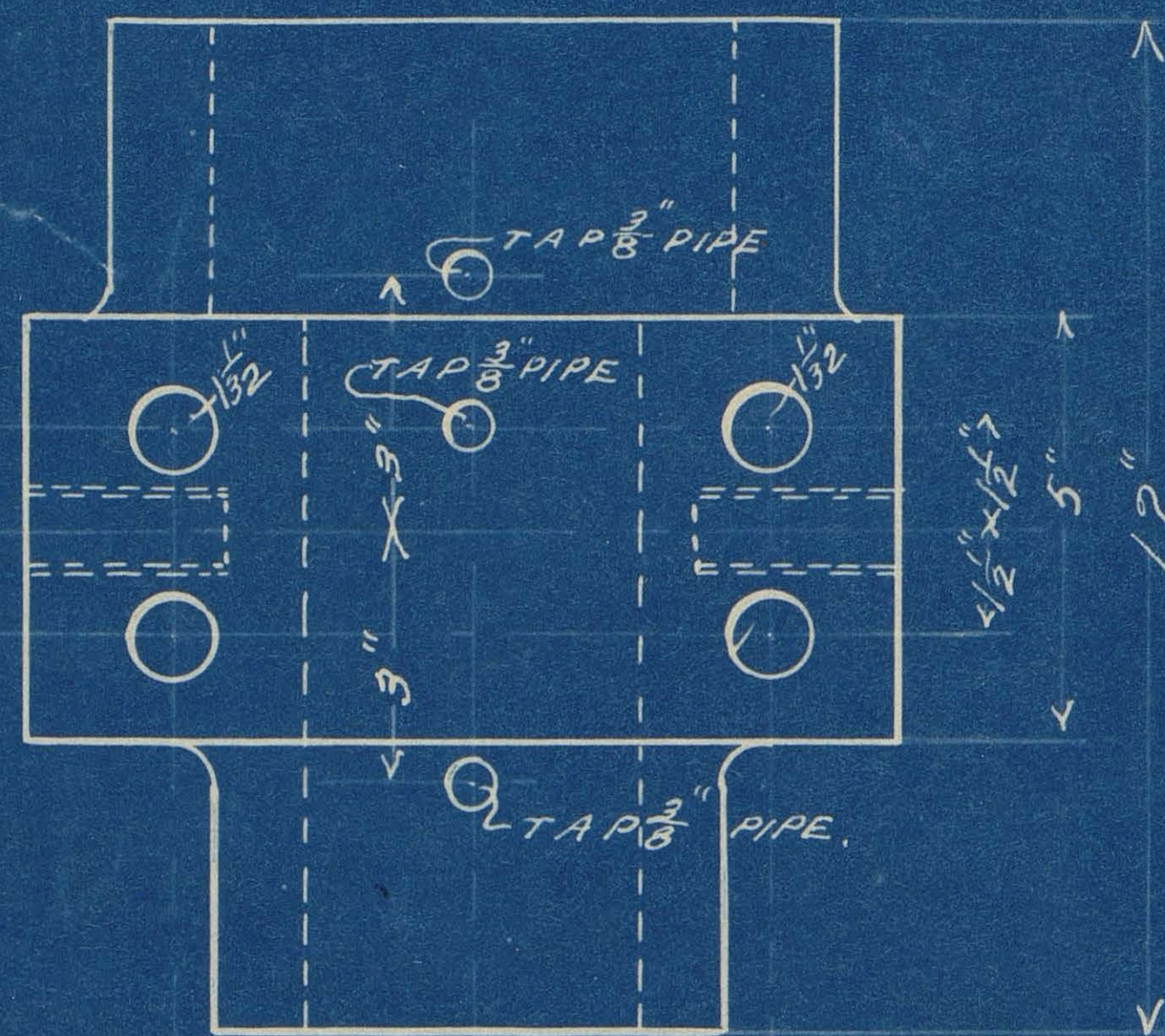
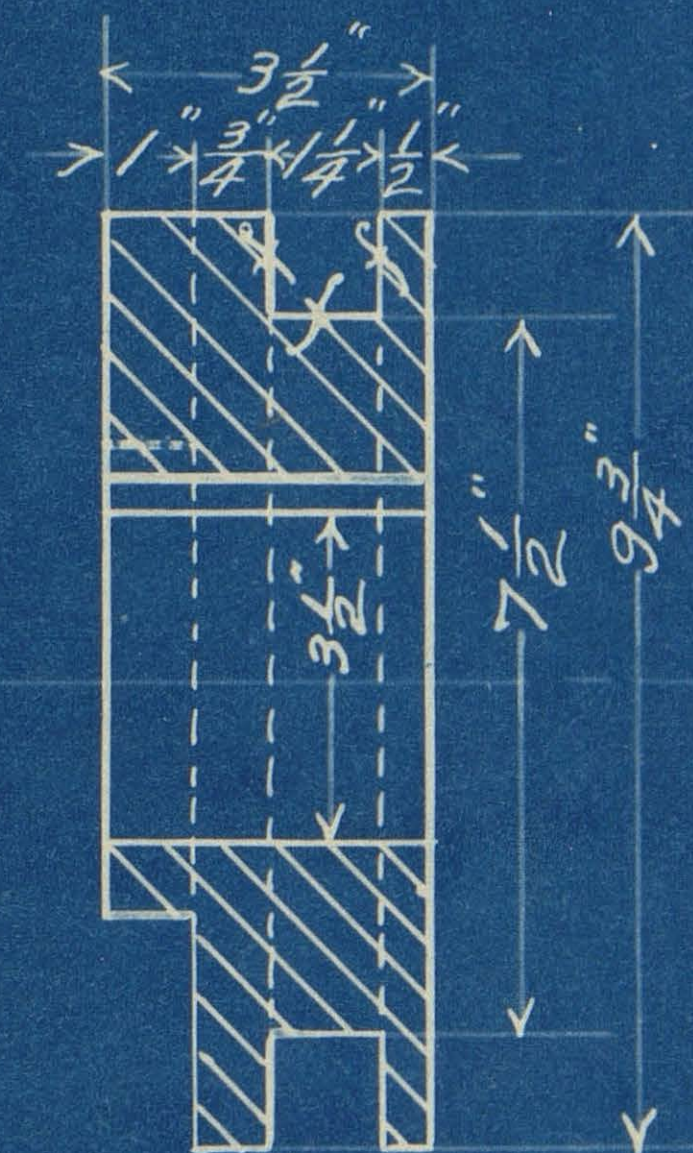
LT HAND DRUM SHAFT BOX  
TOP HALF.



LT HAND DRUM SHAFT BOX,  
BOTTOM HALF.



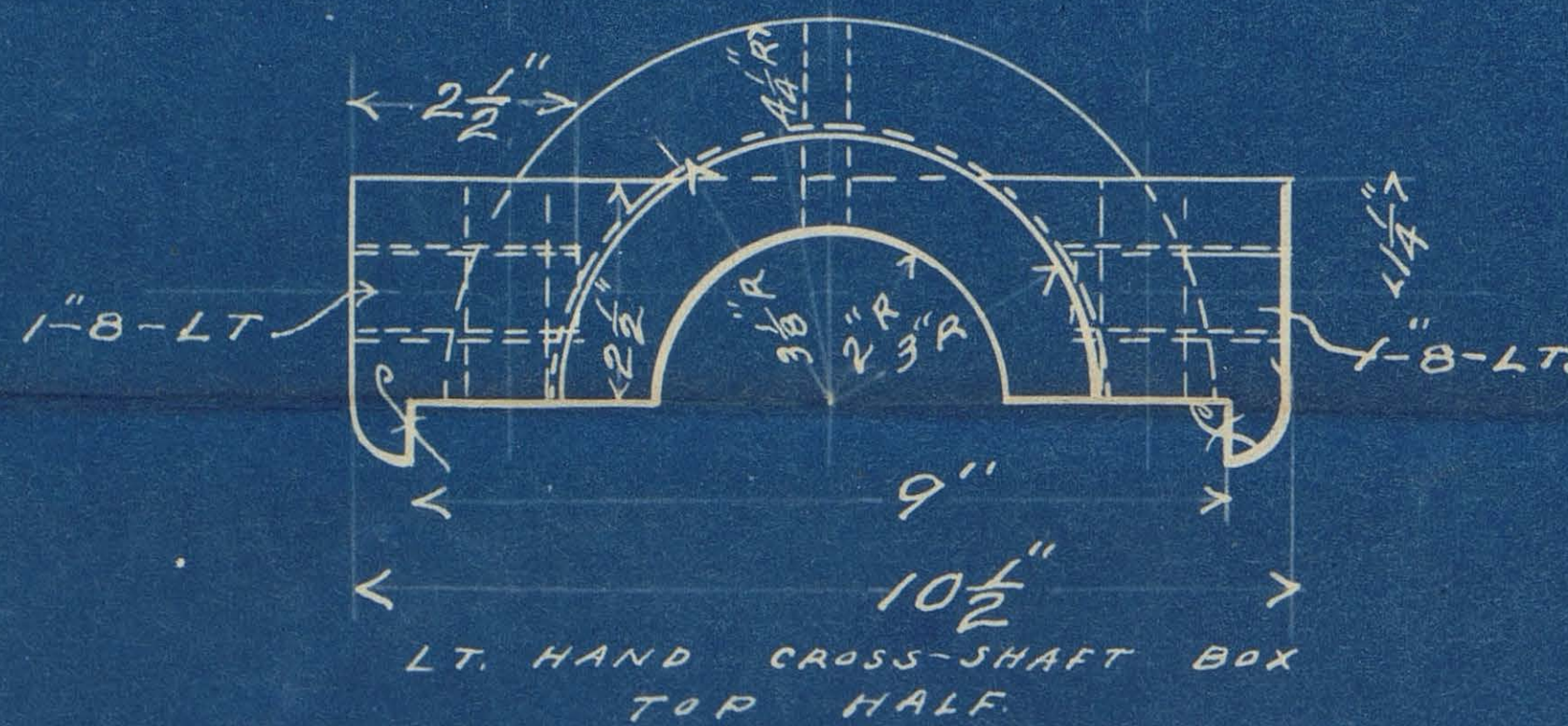
DRUM CLUTCH  
GRAY CAST 6.



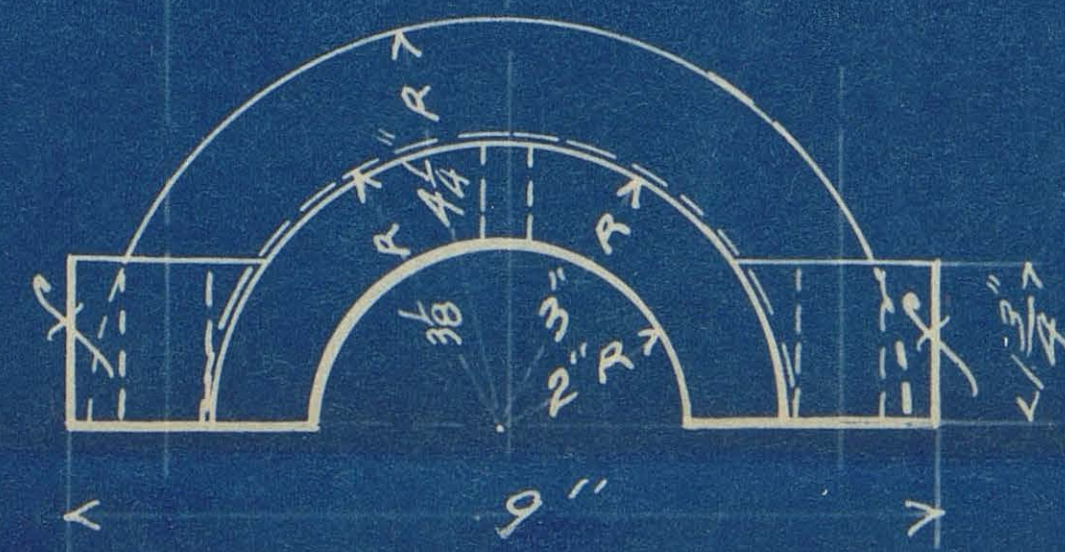
DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: 1-3"=1 FT. f MEANS FINISH.

DEPARTMENT OF  
MECHANICAL ENGINEERING  
K. S. A. C.

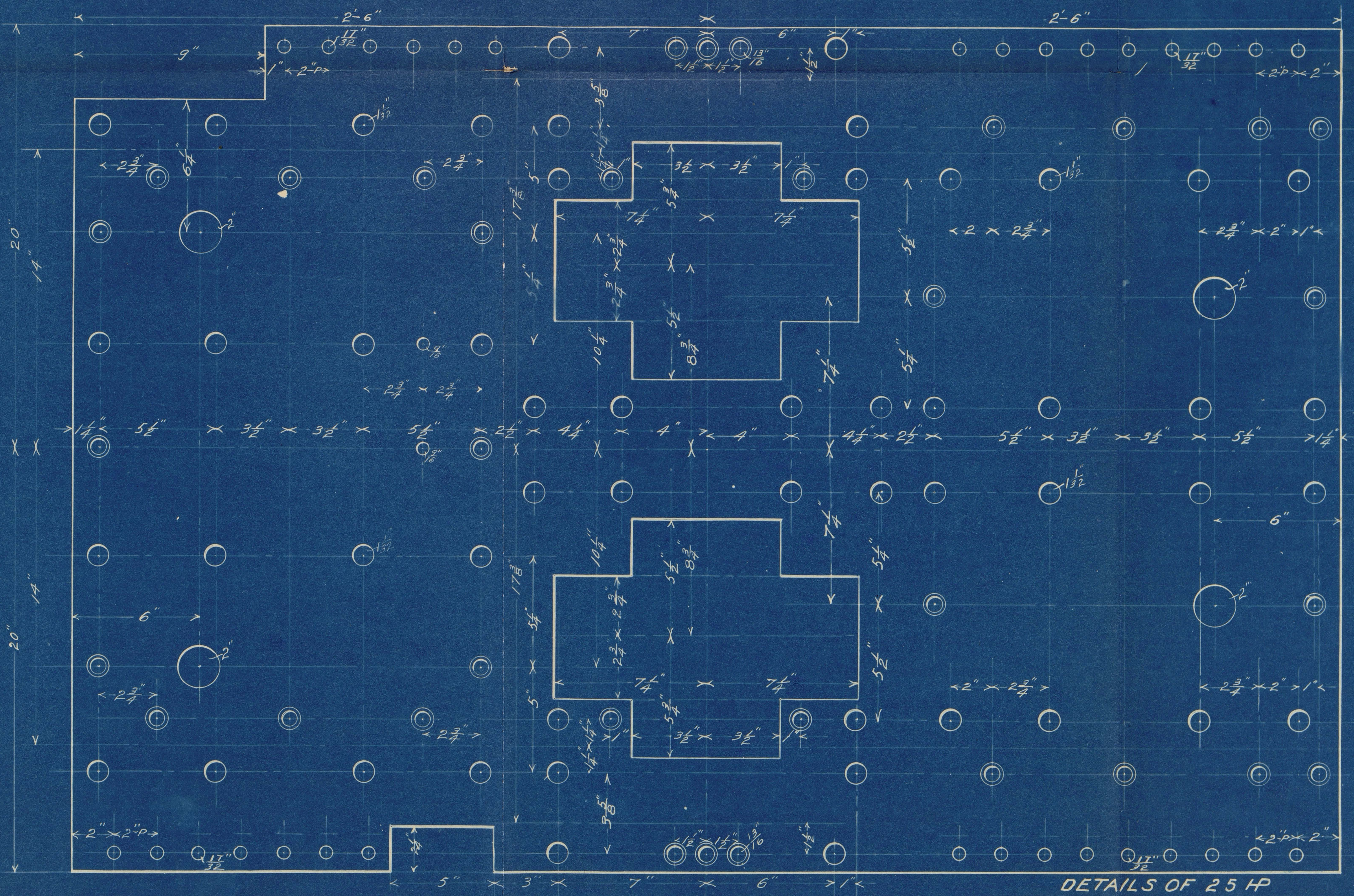


LT. HAND CROSS-SHAFT BOX  
TOP HALF.



LT HAND CROSS SHAFT BOX,  
BOTTOM HALF.

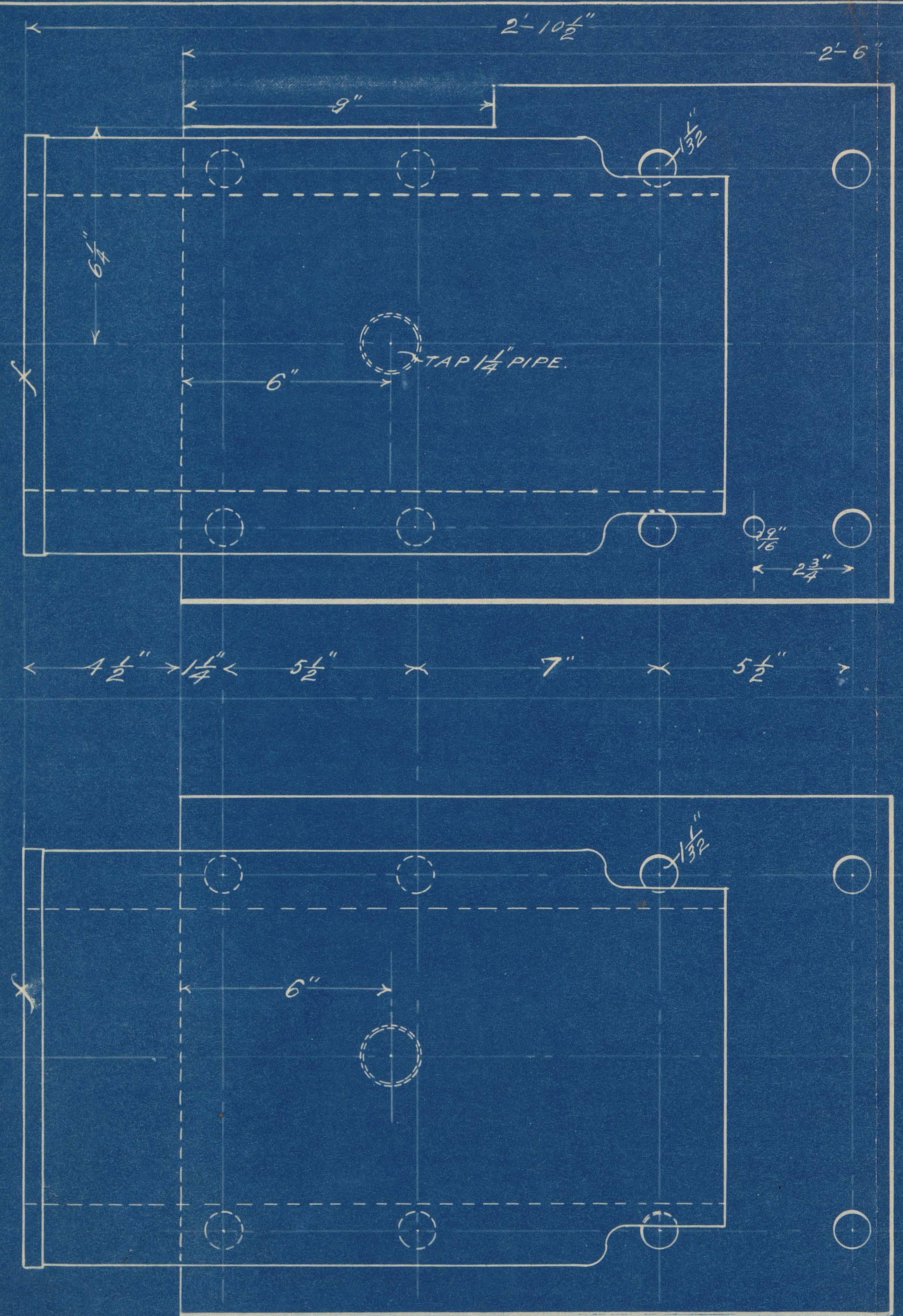




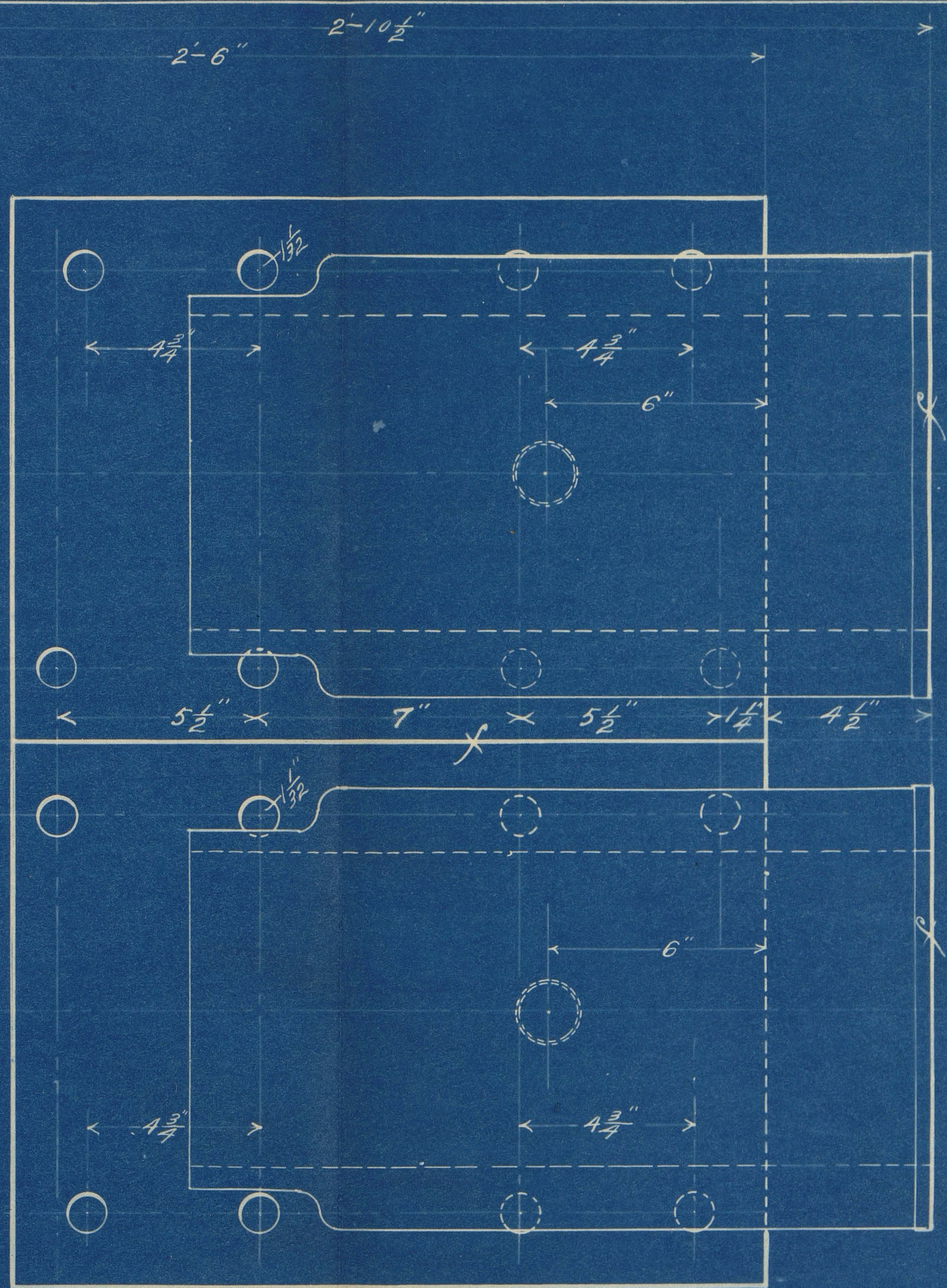
CYLINDER BED PLATE,  
MILD STEEL  
5/8" THICK

DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE  
SCALE: -3"=1FT. f MEANS FINISH  
DEPARTMENT OF  
MECHANICAL ENGINEERING  
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LOCATION OF CYLINDERS  
AND  
LOCATION OF FOUNDATION BOLTS.

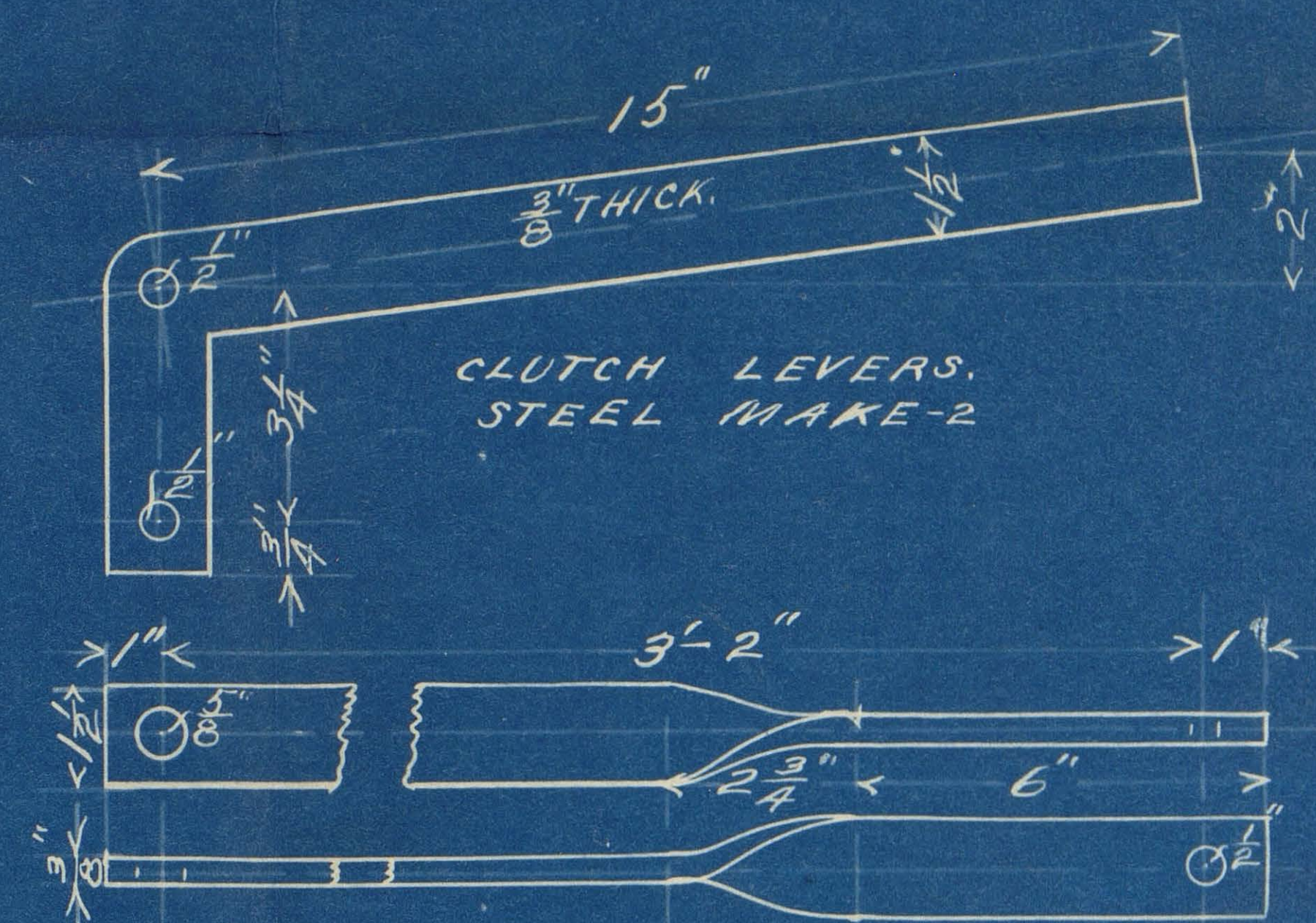
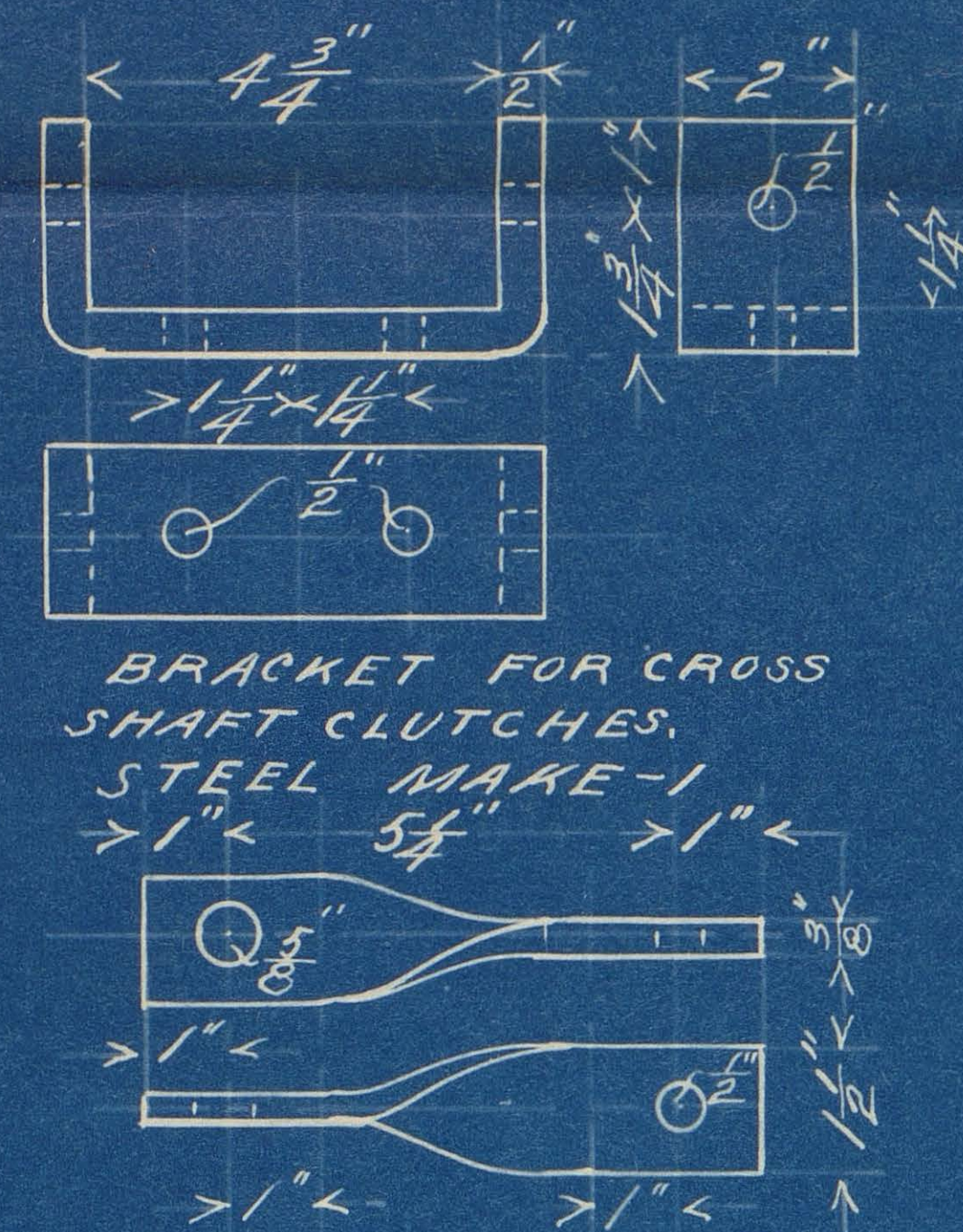
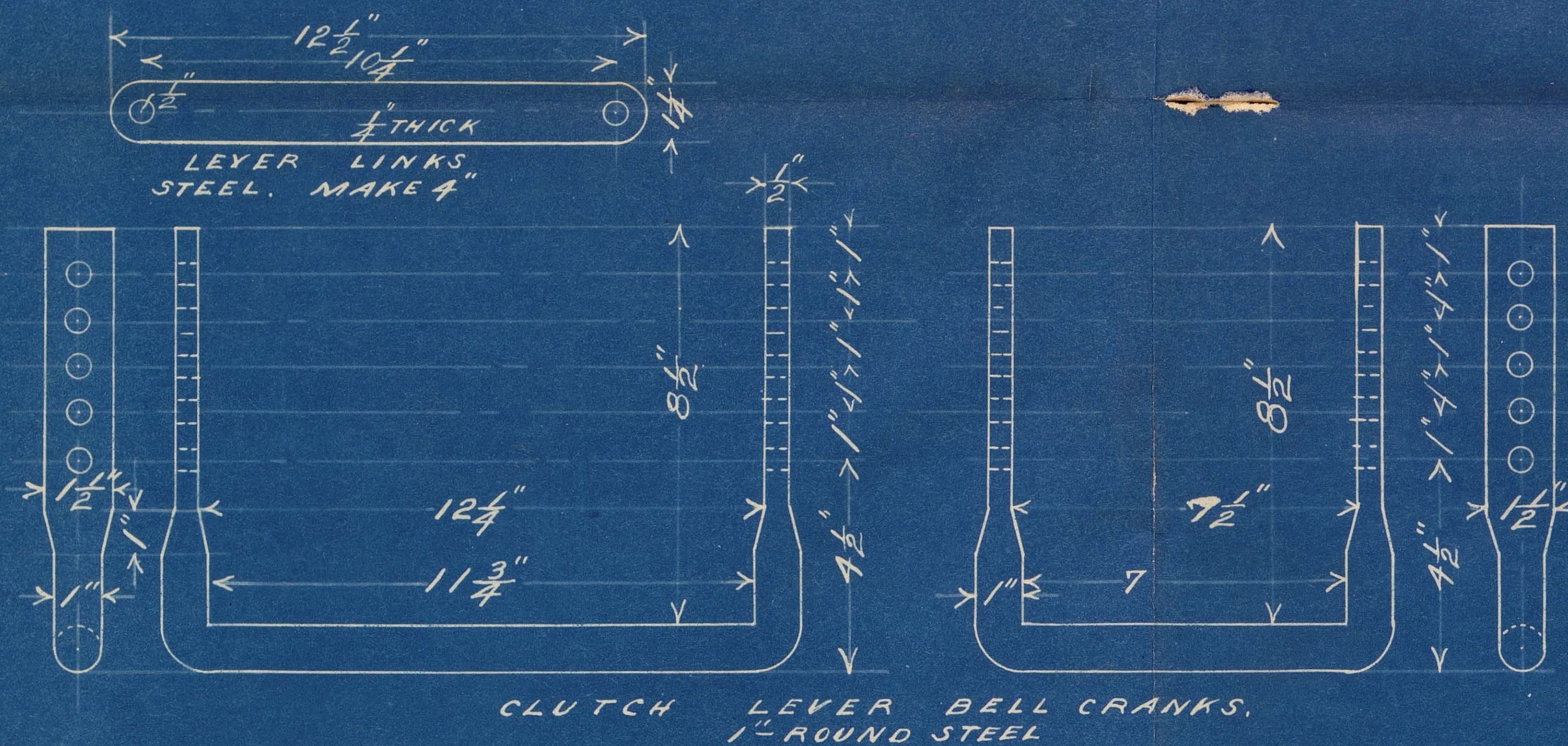


DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

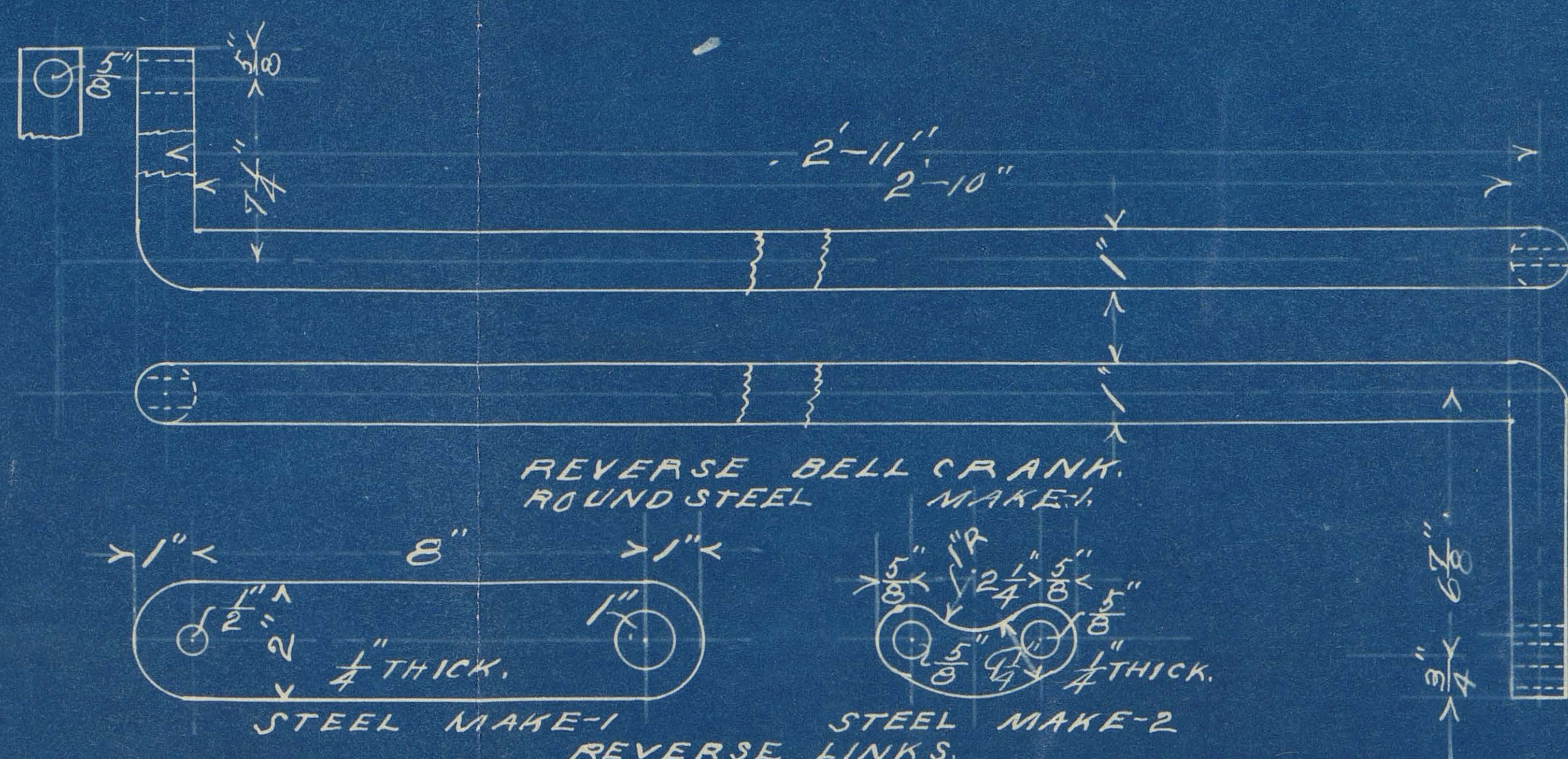
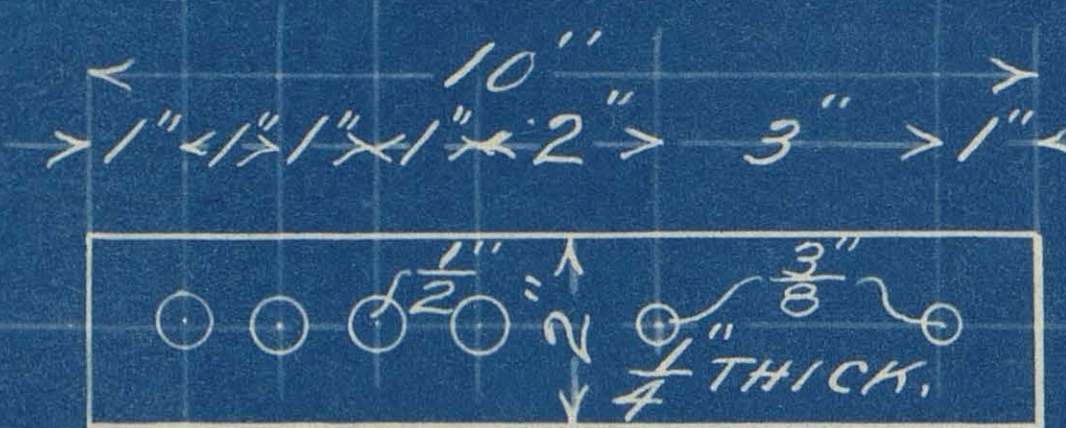
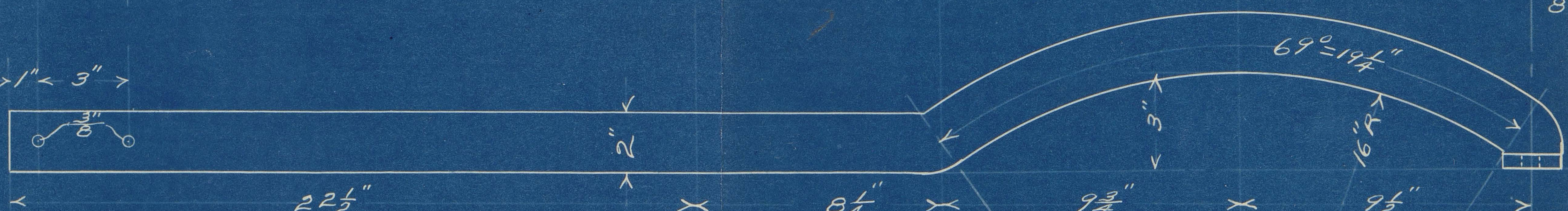
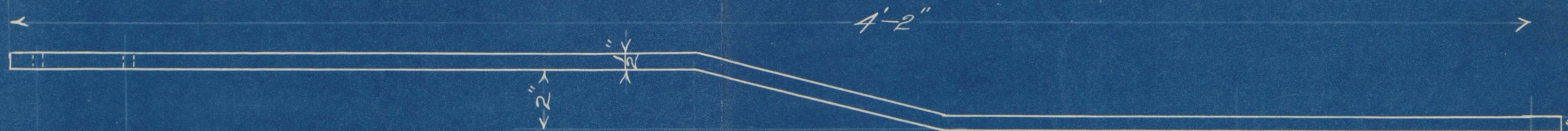
SCALE: -3"=1FT. f MEANS FINISH.

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K. S. A. C.





CROSS SHAFT CLUTCH LINKS, STEEL 1-OF-EACH.



DETAILS OF 25 HP GASOLINE TRACTION ENGINE

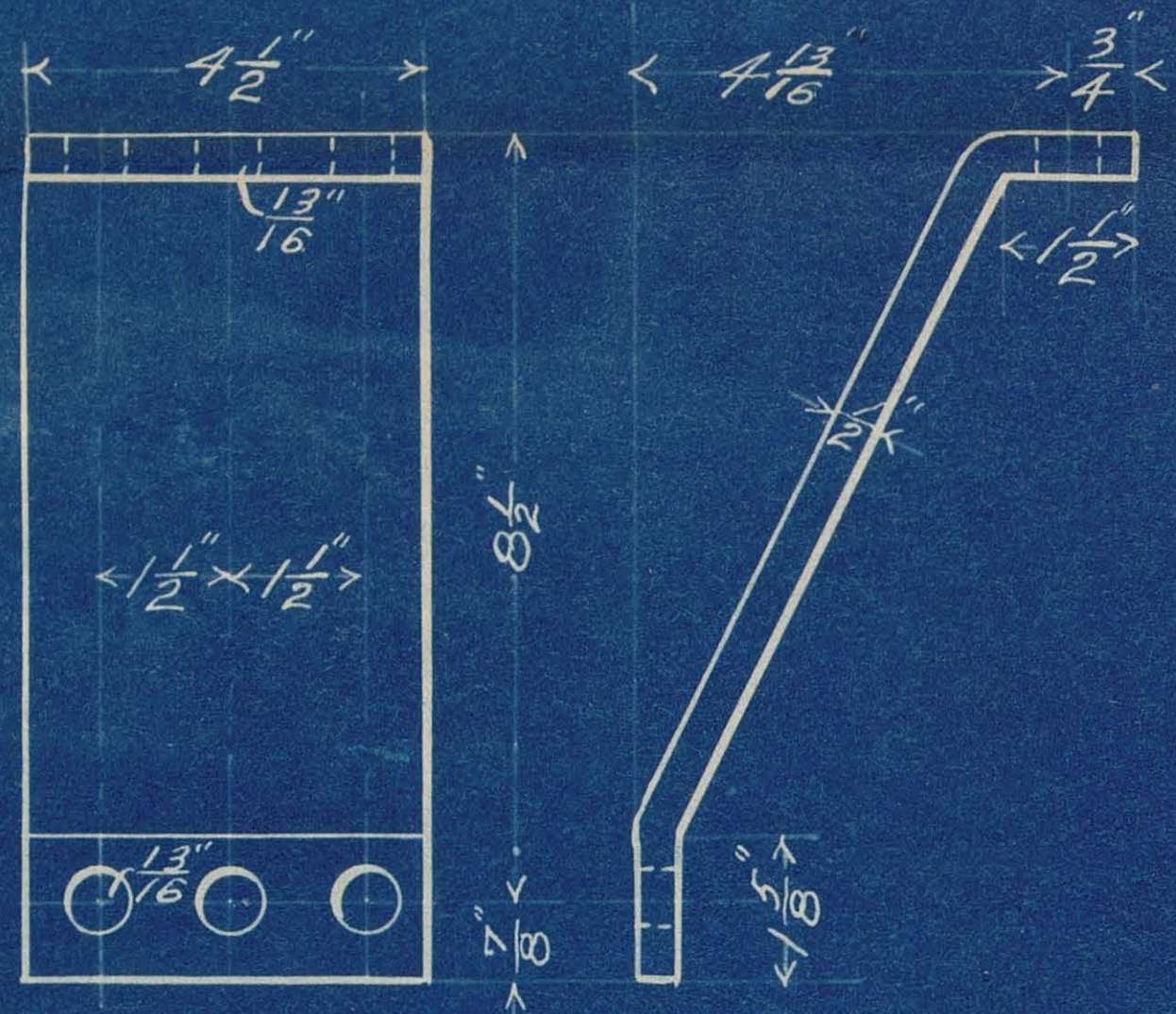
SCALE: 1/3" = 1 FT. f MEANS FINISH.

DEPARTMENT OF MECHANICAL ENGINEERING K.S.A.C.

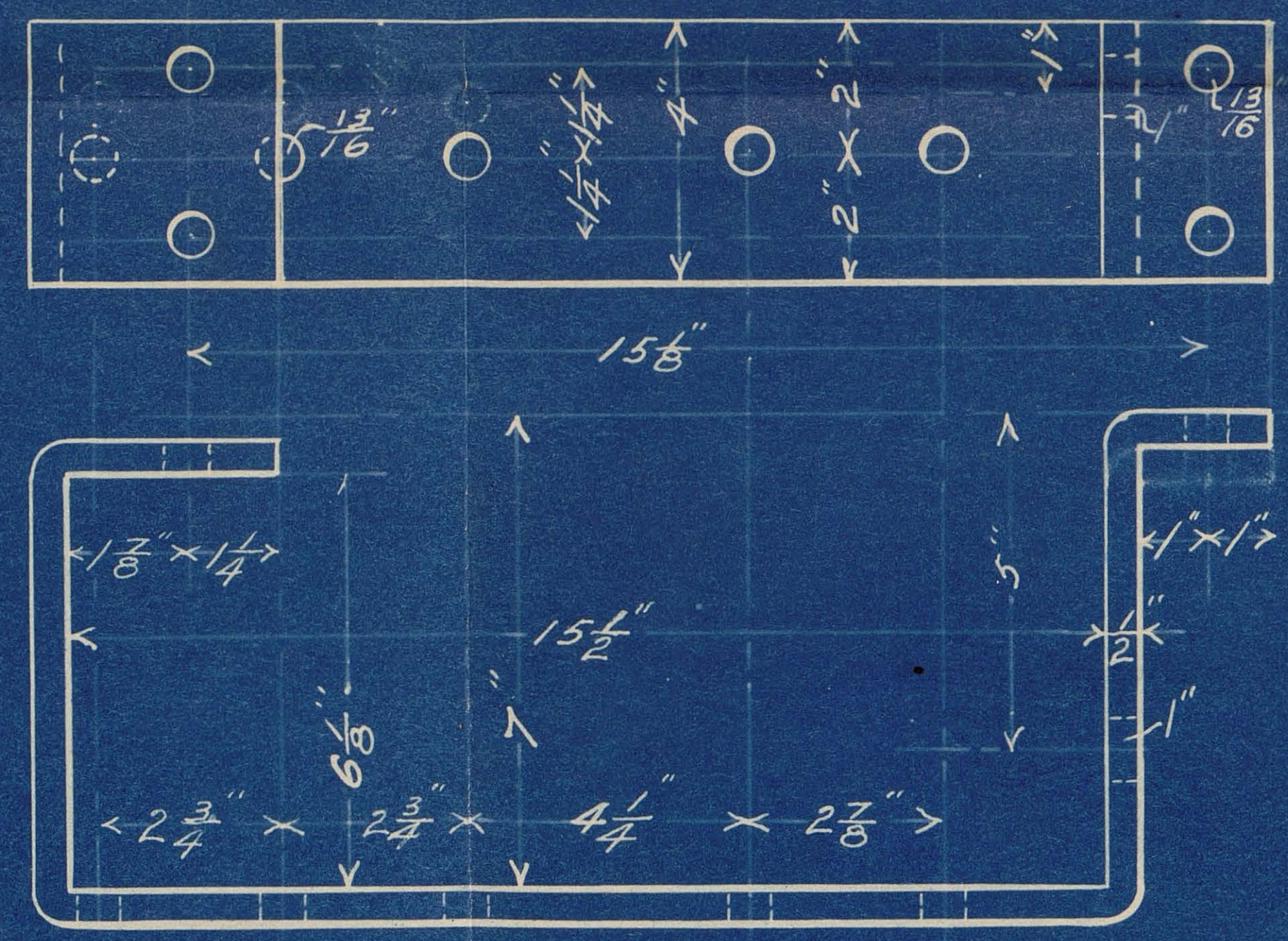




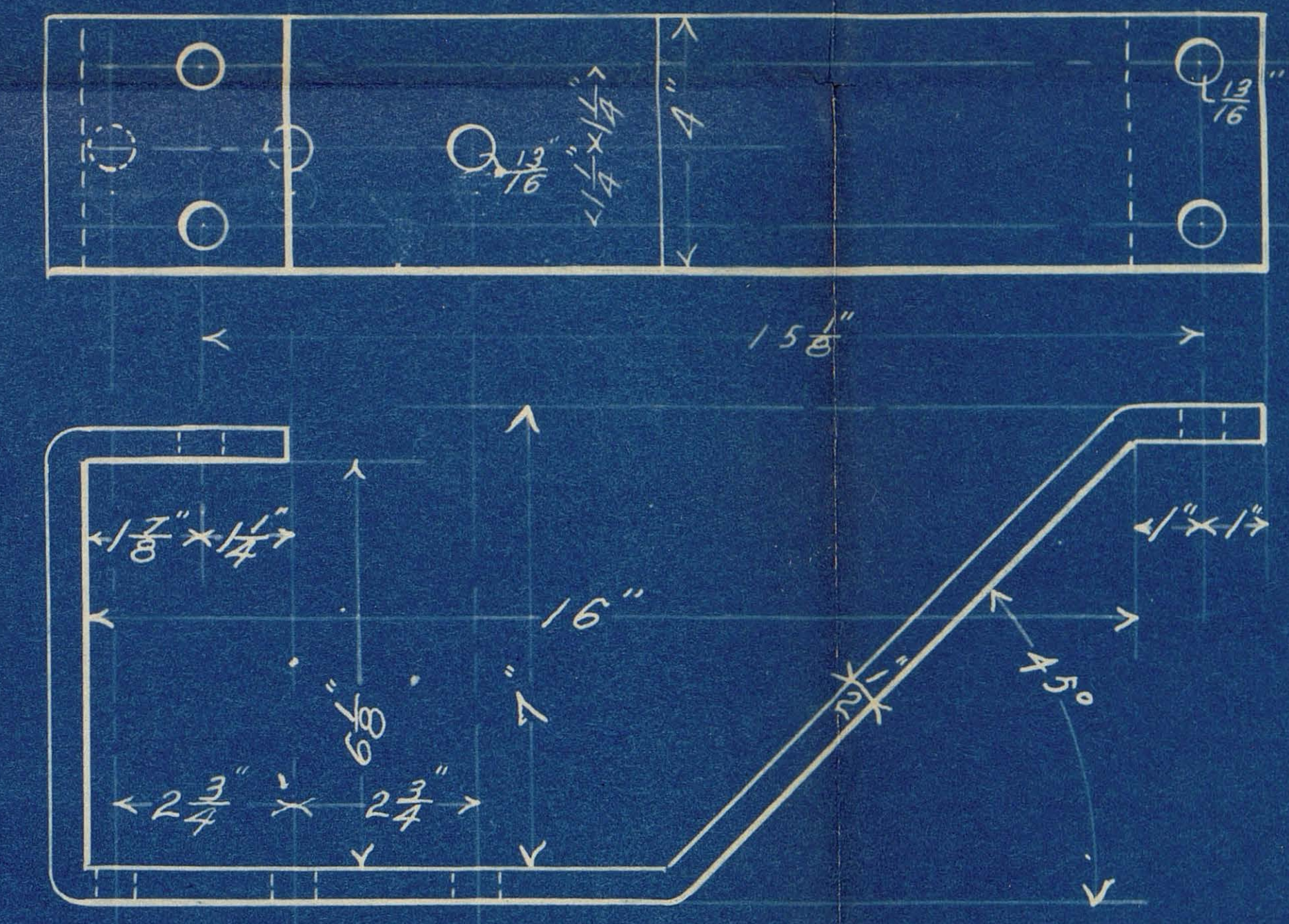




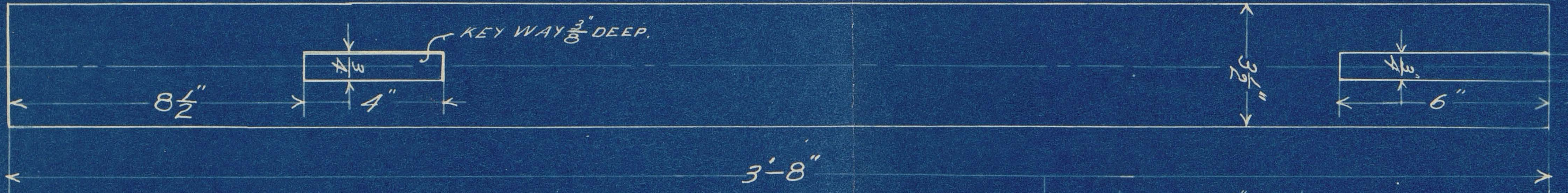
CRANK BOX BRACE  
MILD STEEL  
2-REQD.



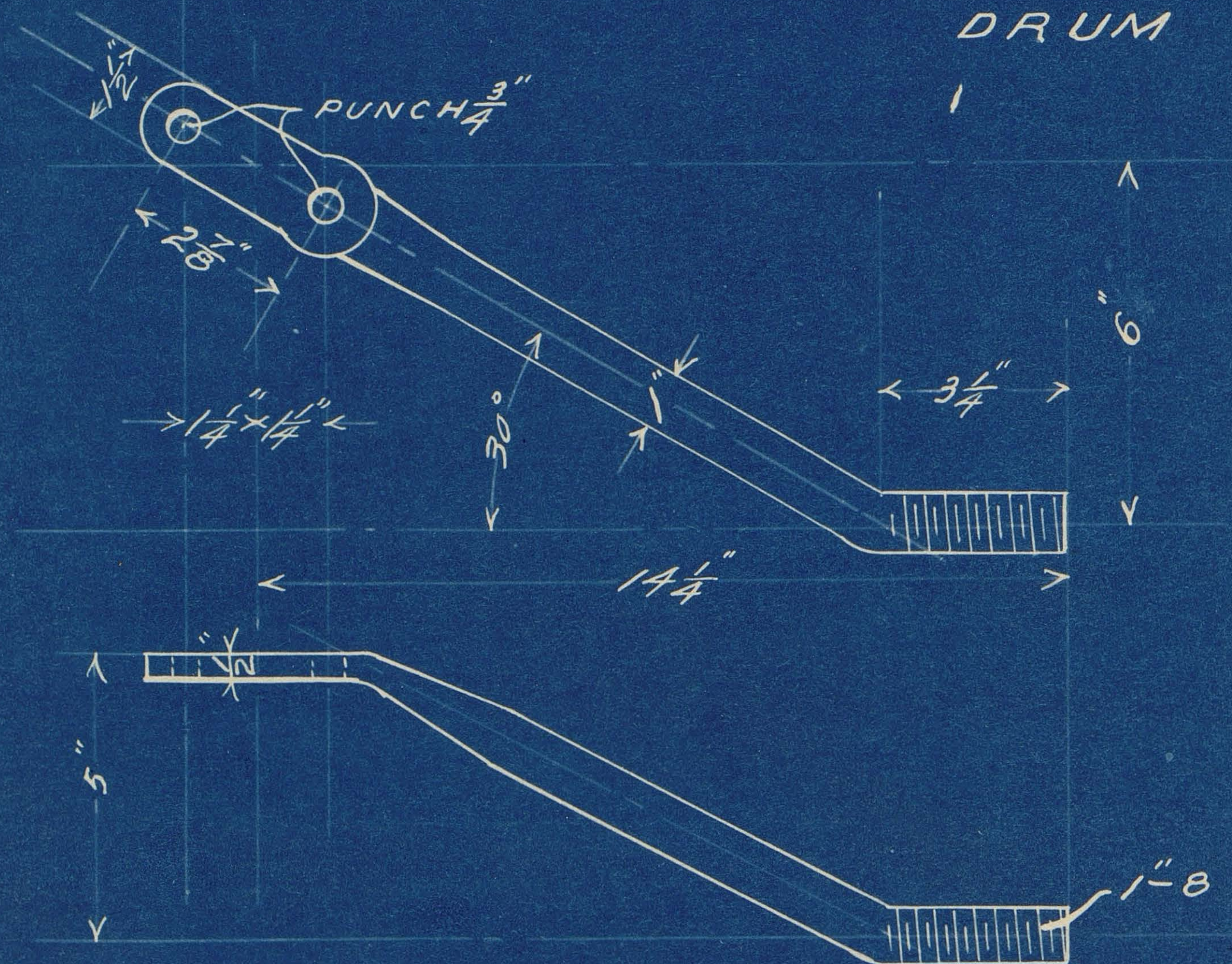
STEERING ROLLER SUPPORT  
MILD STEEL  
1-RIGHT.



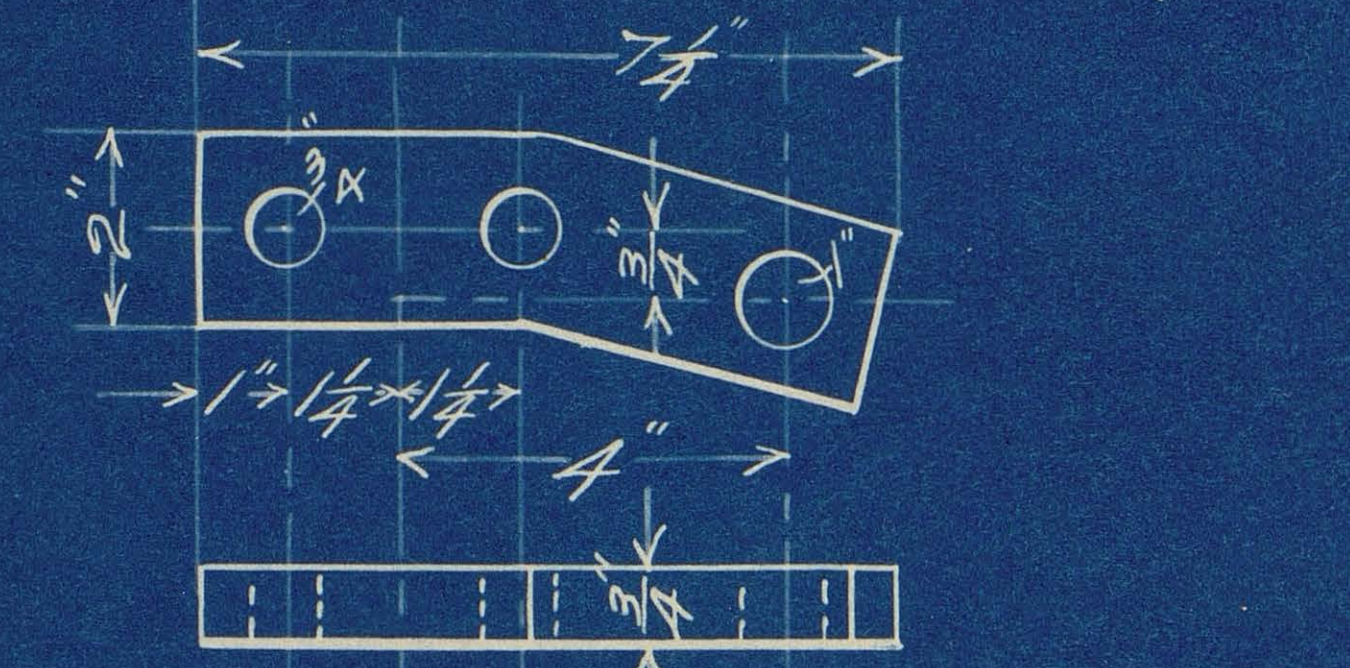
STEERING ROLLER SUPPORT  
MILD STEEL  
1-LEFT.



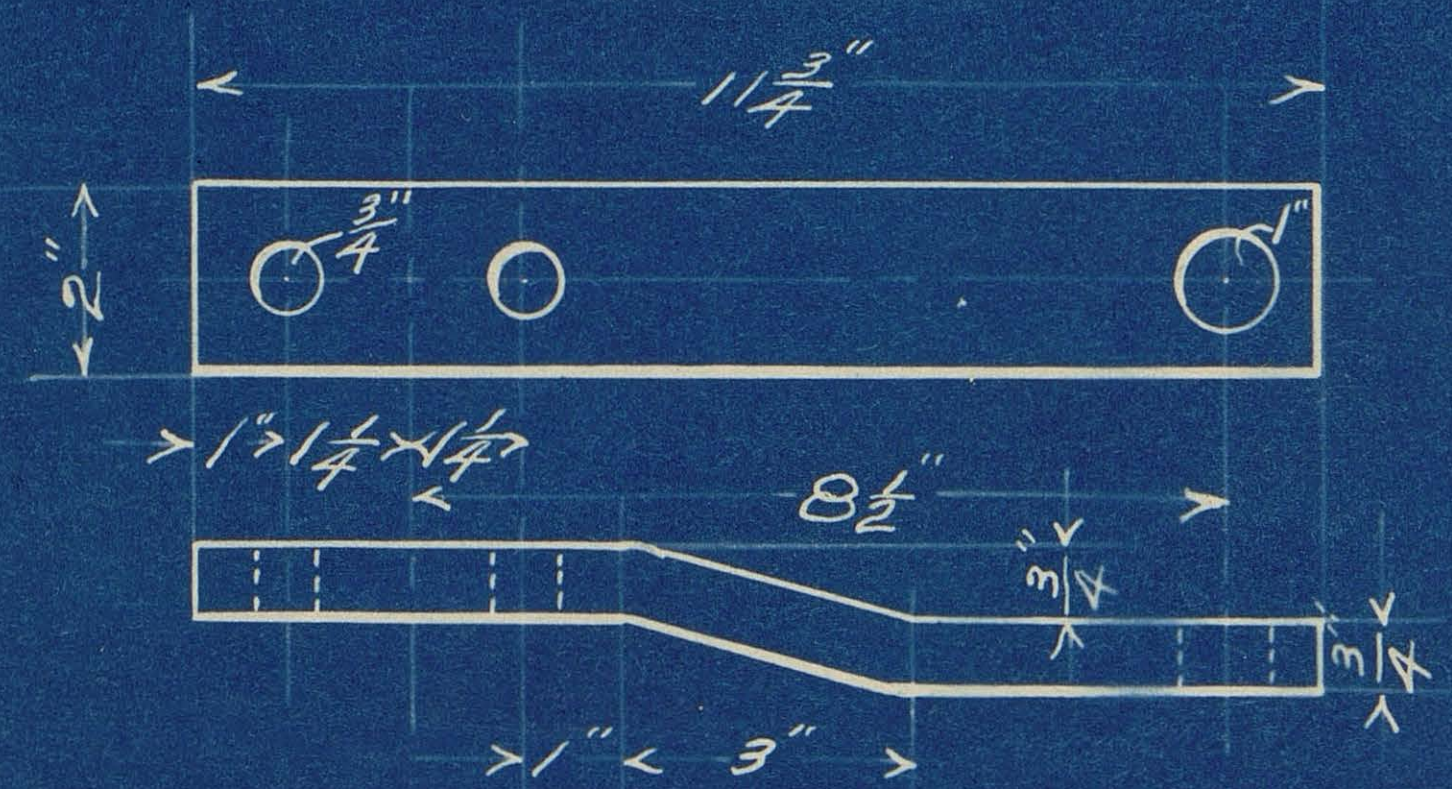
DRUM SHAFT



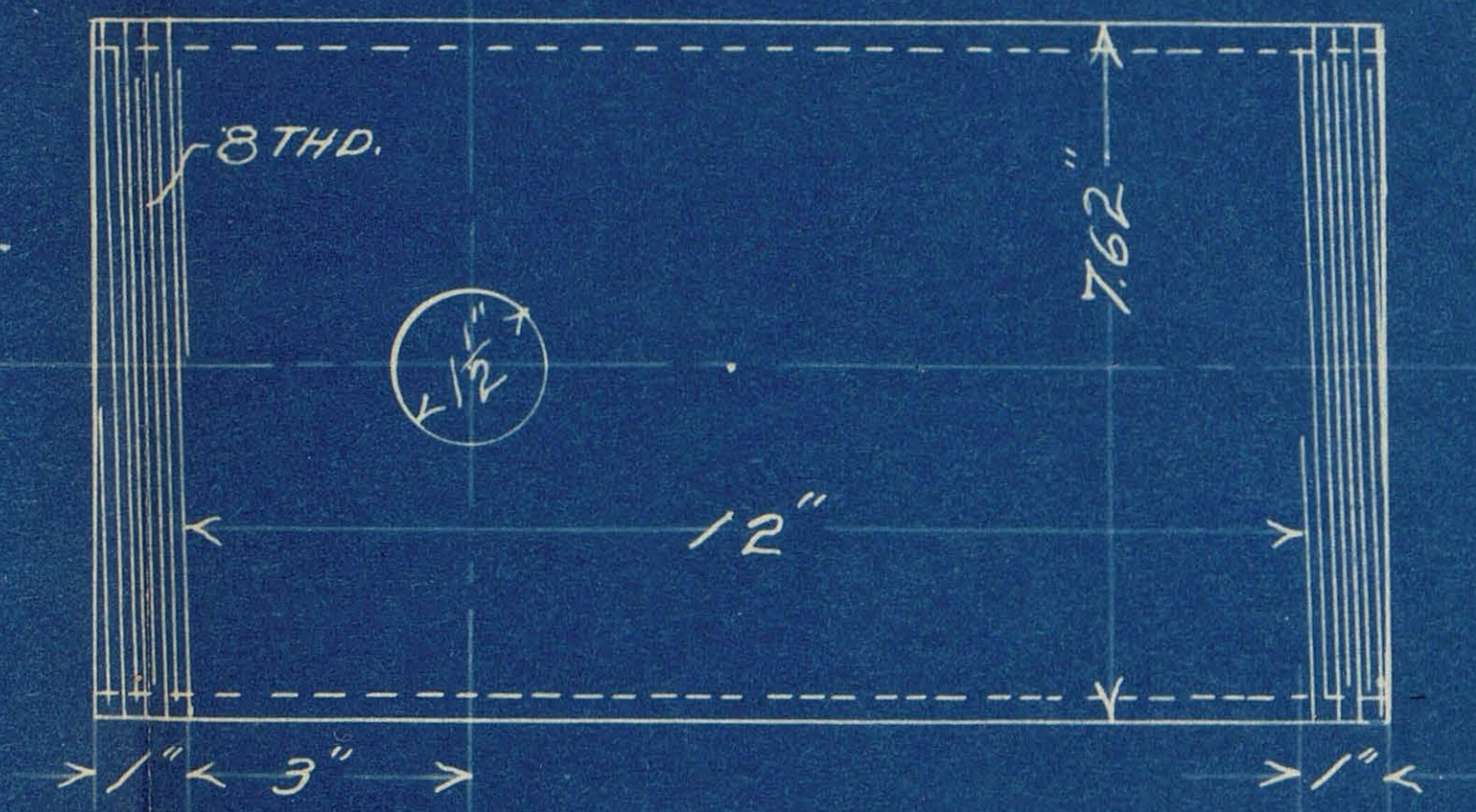
STEEL BRACE FOR  
STEERING ROLLER SUPPORT.



DRUM CLUTCH YOKE SUPPORT  
STEEL



BULL PINION CLUTCH YOKE SUPPORT,  
STEEL.



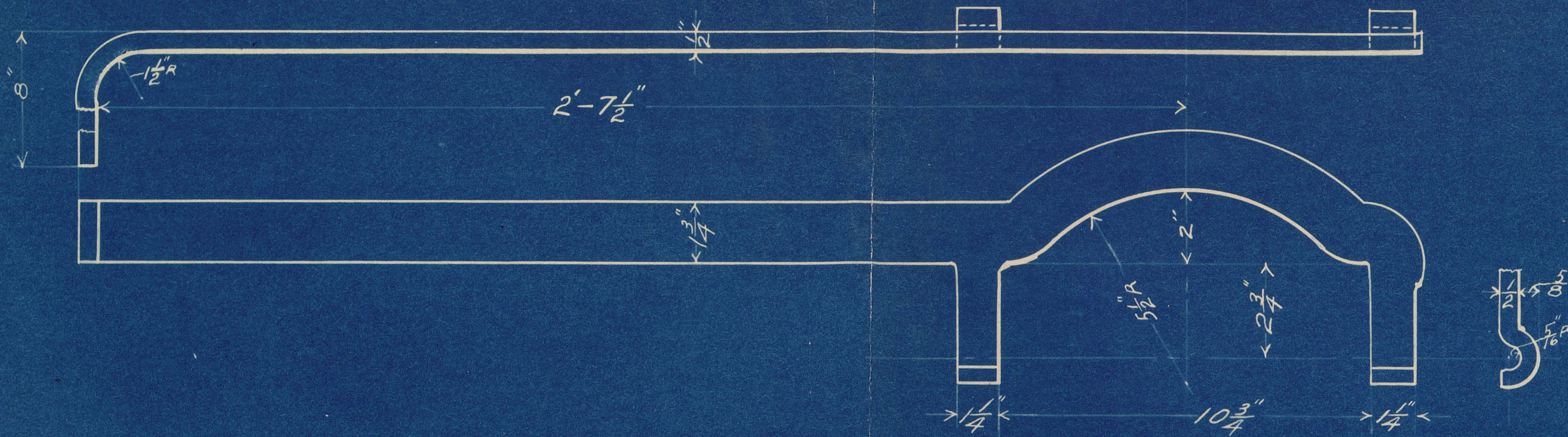
DRUM SHELL  
USE 14" OF 7" PIPE.

DETAILS OF 25 HP  
GASOLINE TRACTION ENGINE

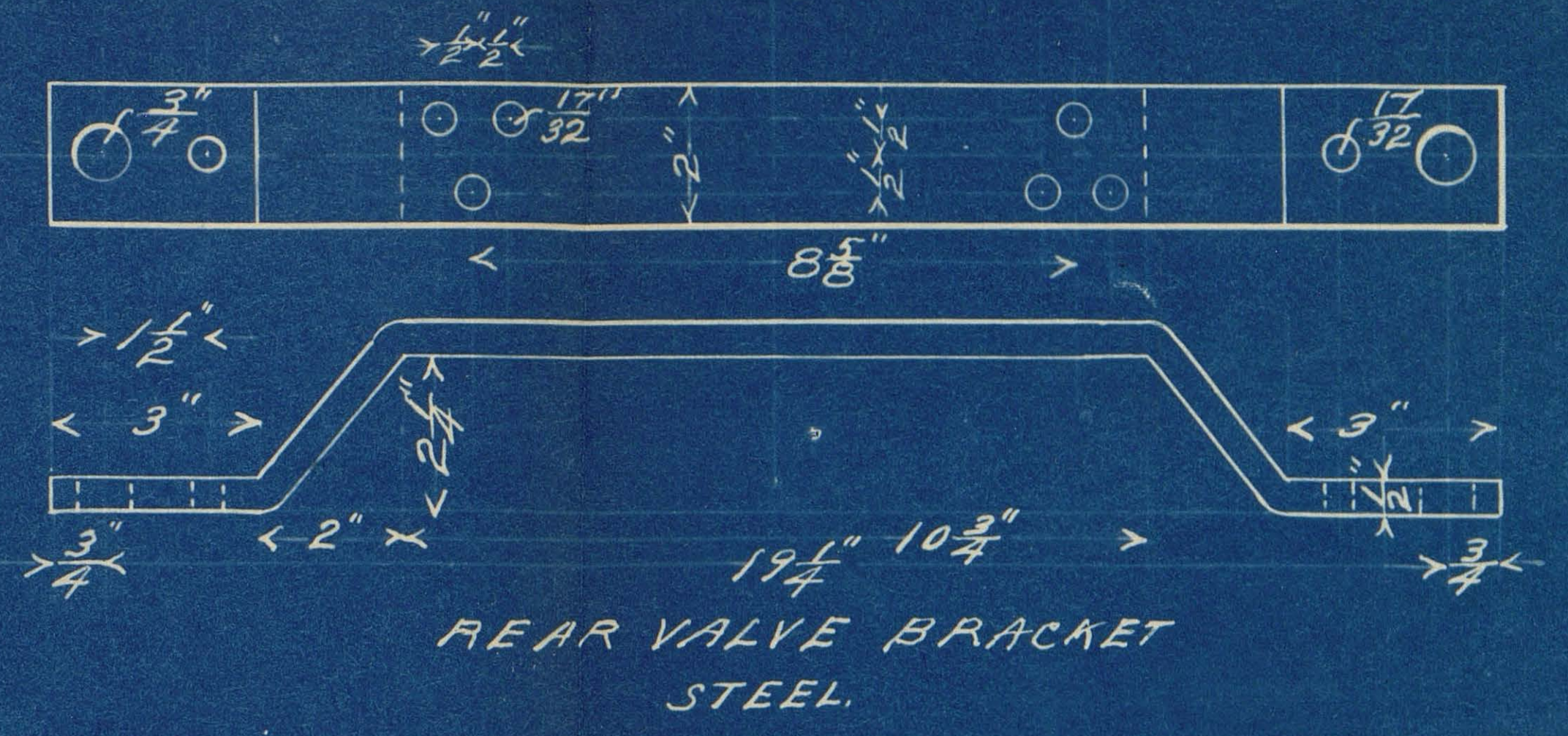
SCALE: 3" = 1 FT. f MEANS FINISH.

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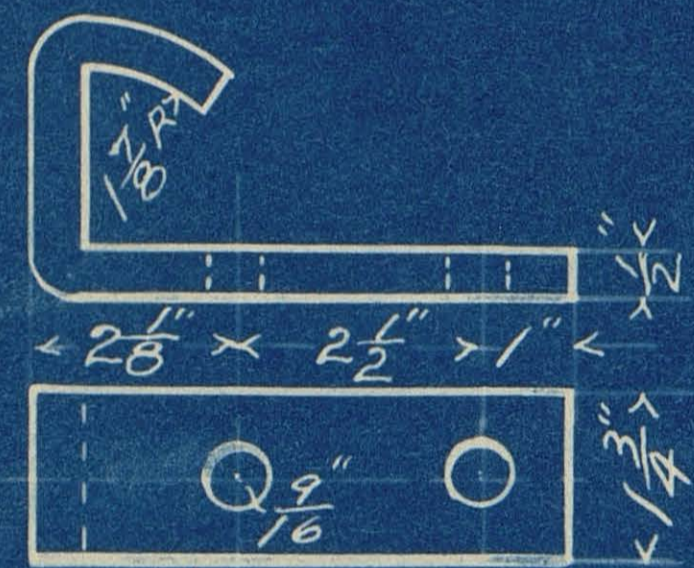




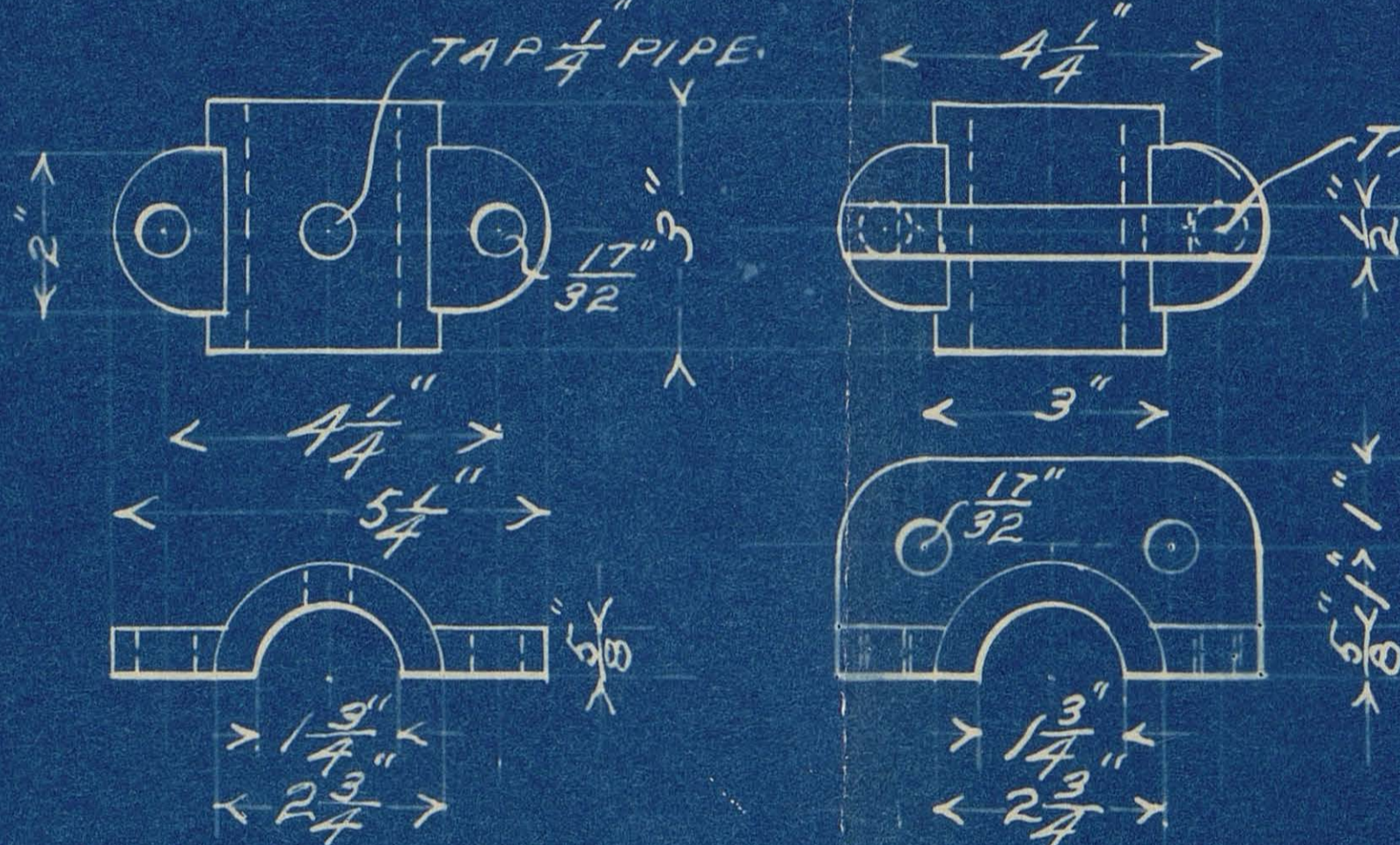
STIFF-GEAR LEVER STEEL.



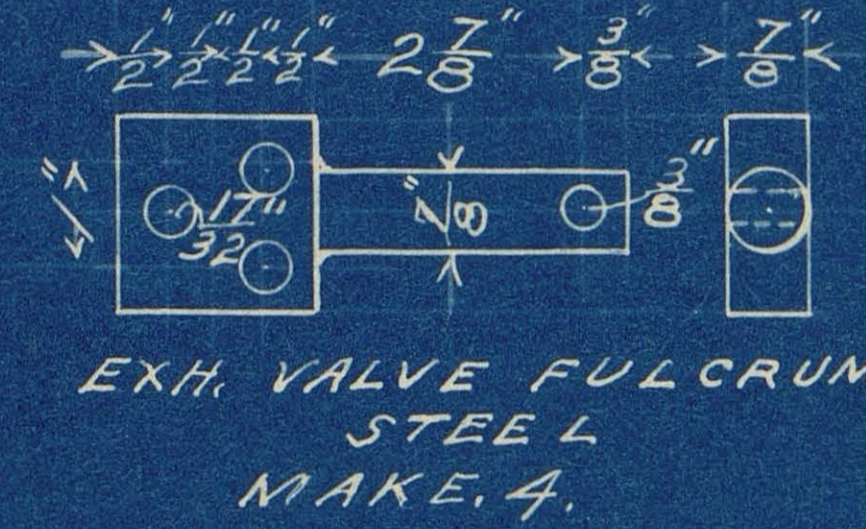
REAR VALVE BRACKET STEEL.



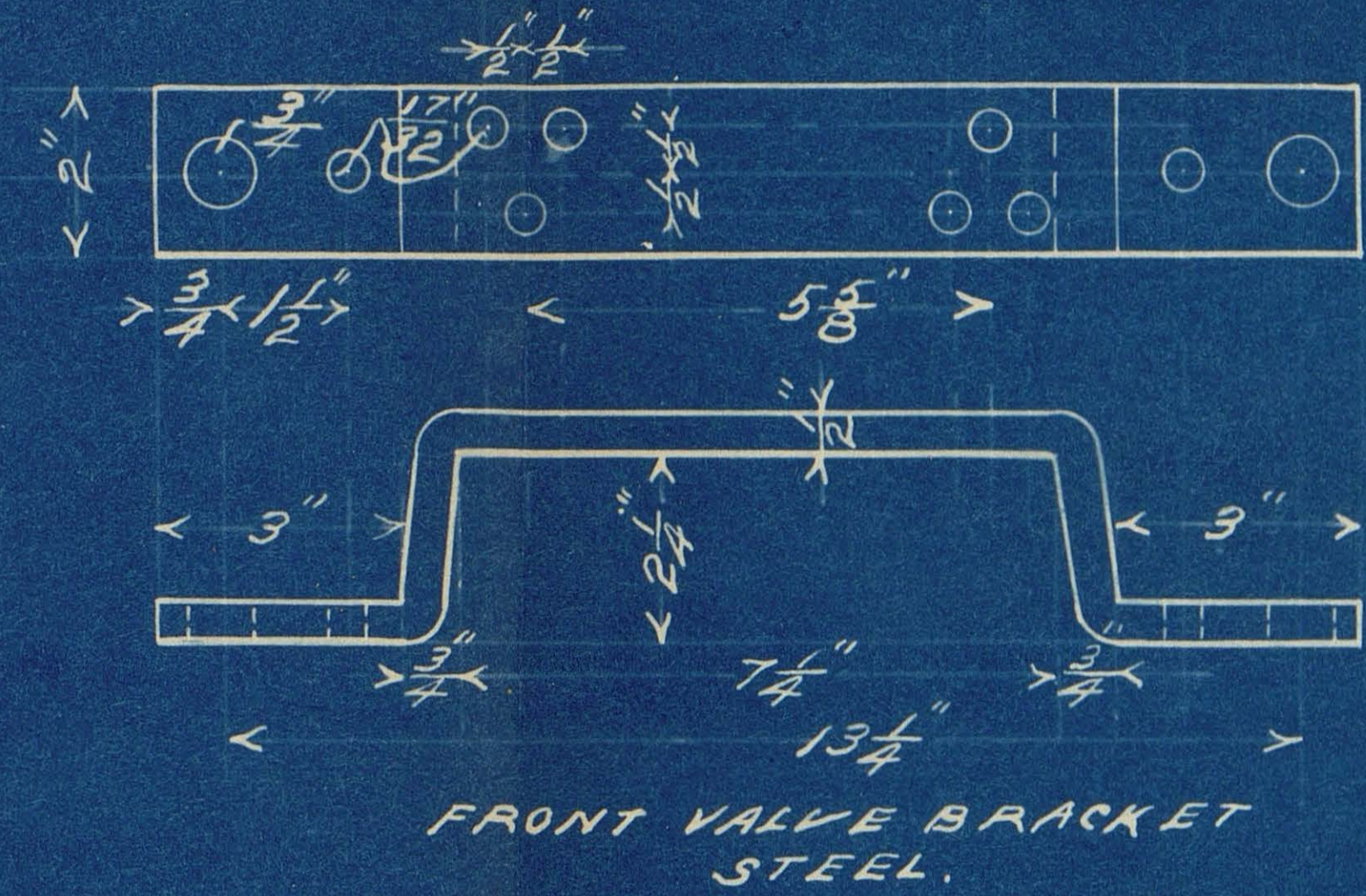
LEVER SUPPORT 2-REQD STEEL.



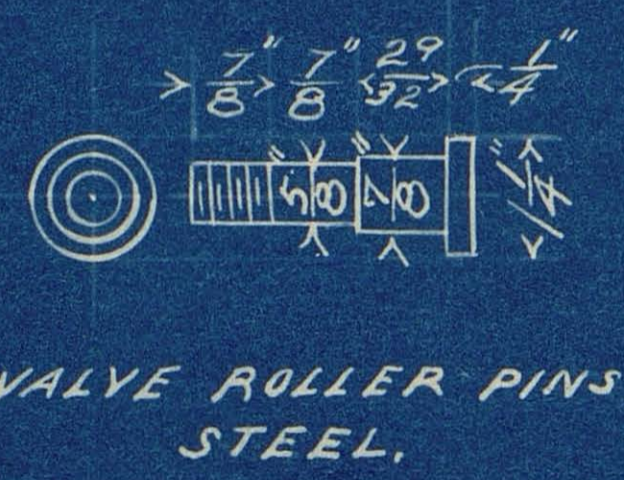
TOP HALF BOTTOM HALF CAM SHAFT BOXES.



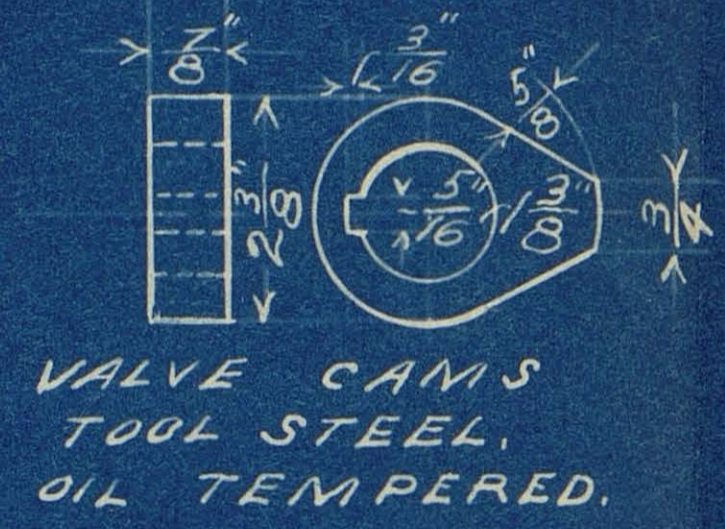
EXH. VALVE FULCRUM STEEL MAKE 4.



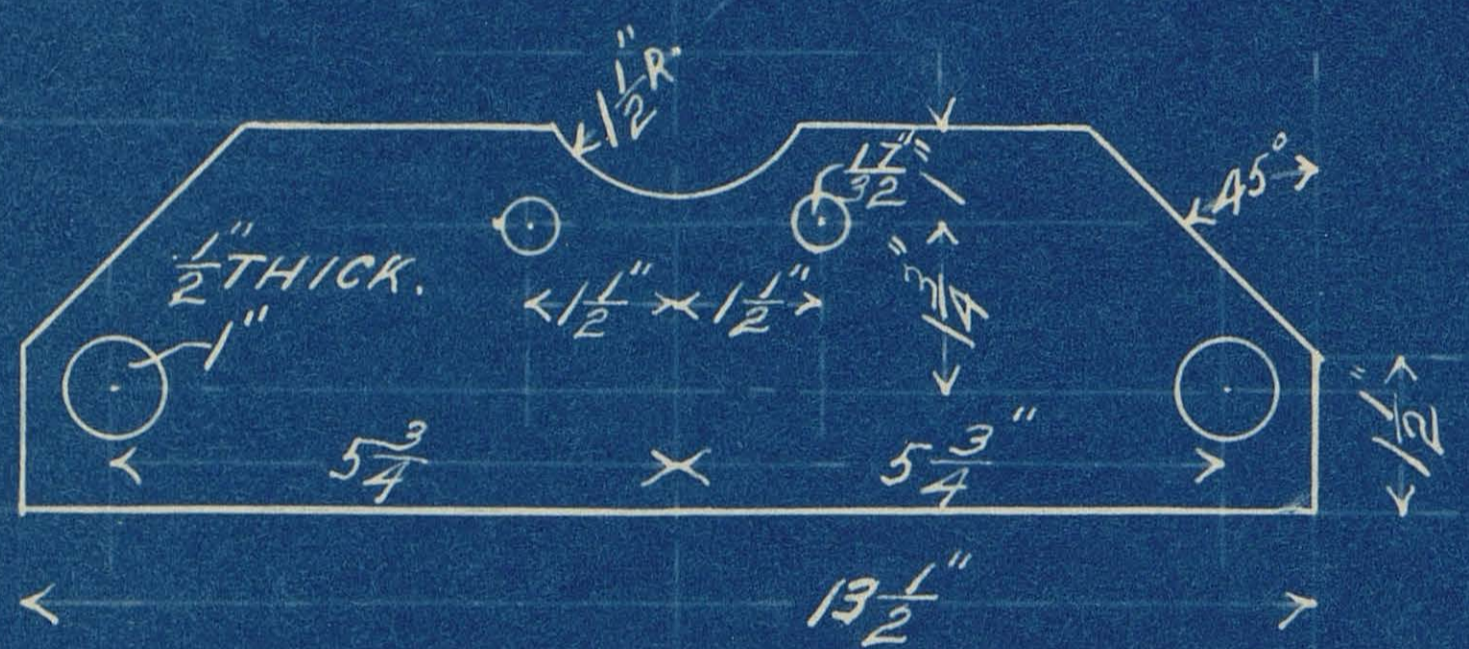
FRONT VALVE BRACKET STEEL.



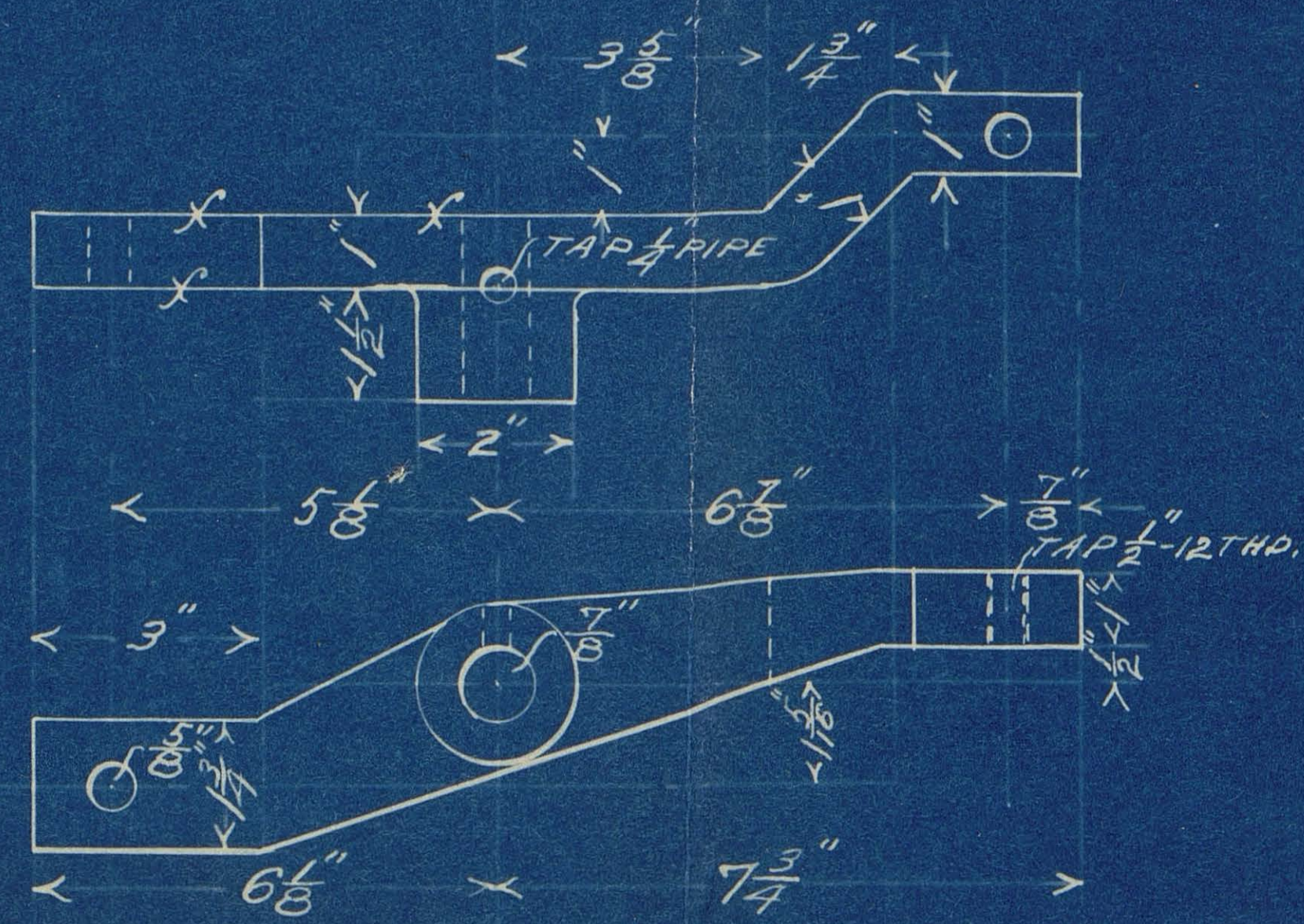
4 VALVE ROLLER PINS STEEL.



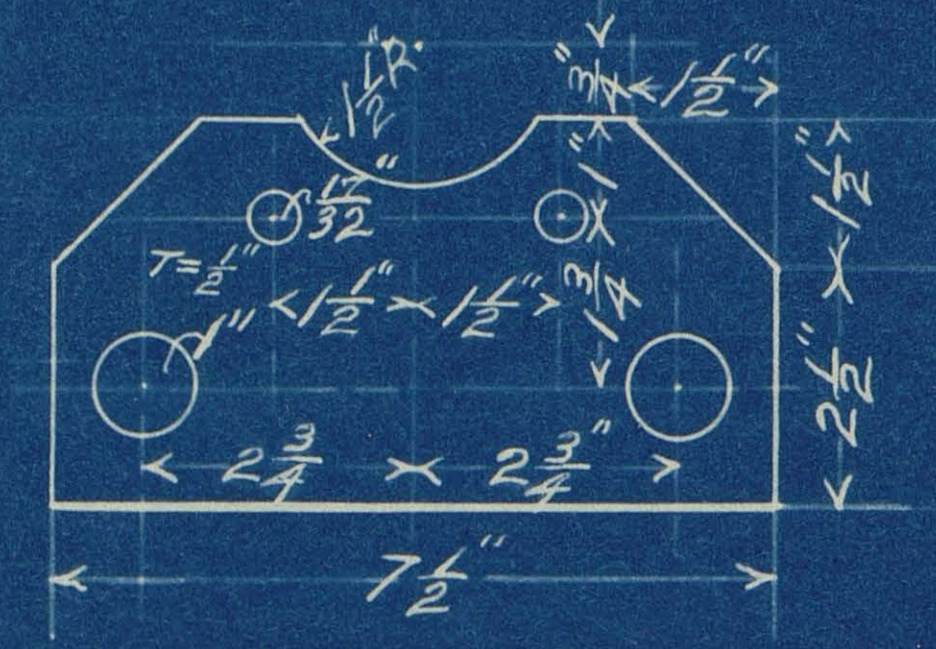
VALVE CAMS TOOL STEEL OIL TEMPERED.



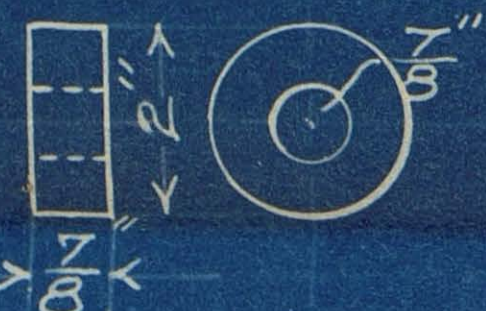
CAM SHAFT SUPPORT STEEL.



EXH VALVE LEVER REAR GRAY CASTG



CAM SHAFT SUPPORT STEEL.



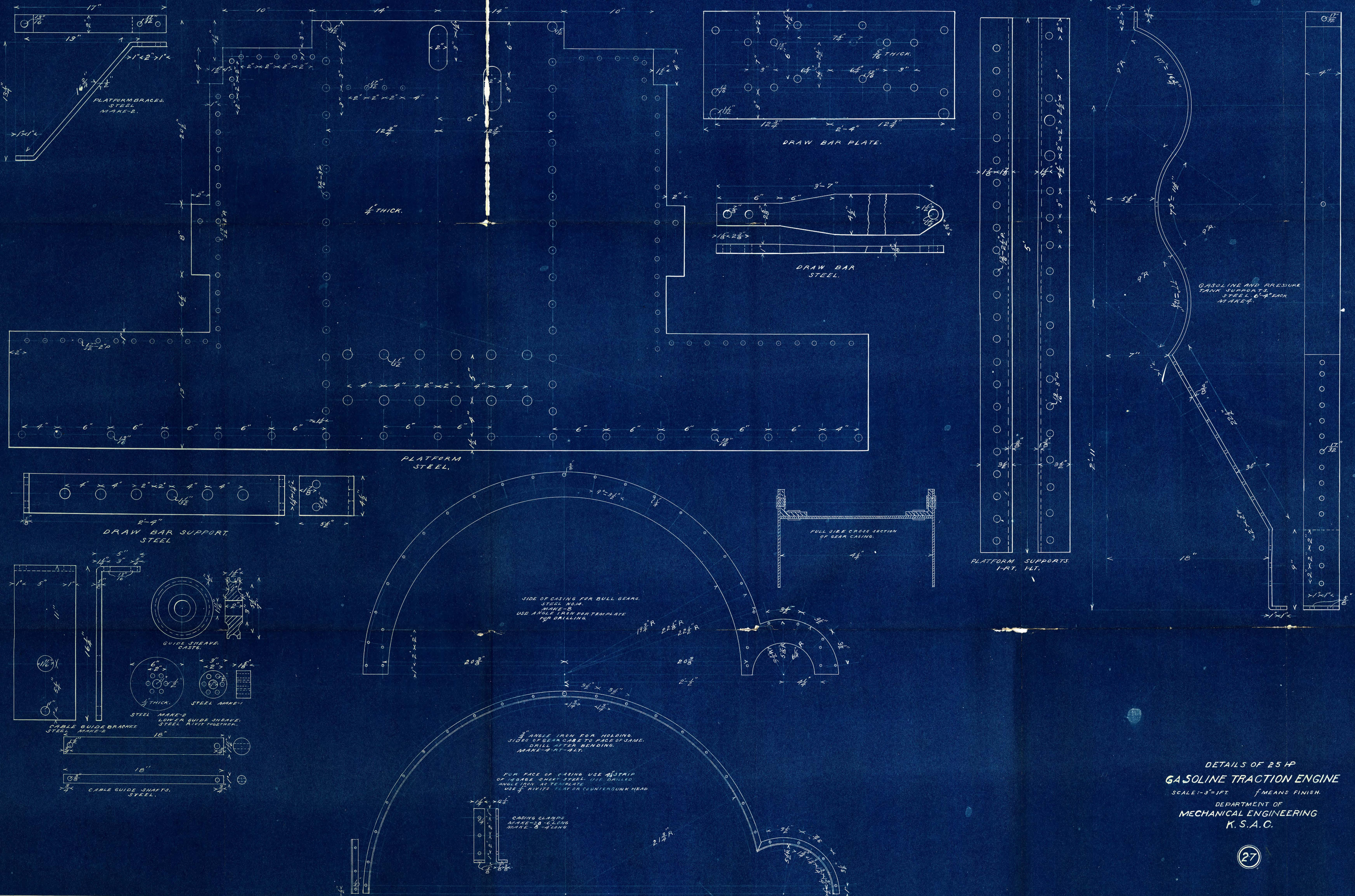
4 VALVE ROLLERS TOOL STEEL OIL TEMPER.

DETAILS OF 25 HP GASOLINE TRACTION ENGINE

SCALE: - 3" = 1 FT. f MEANS FINISH.

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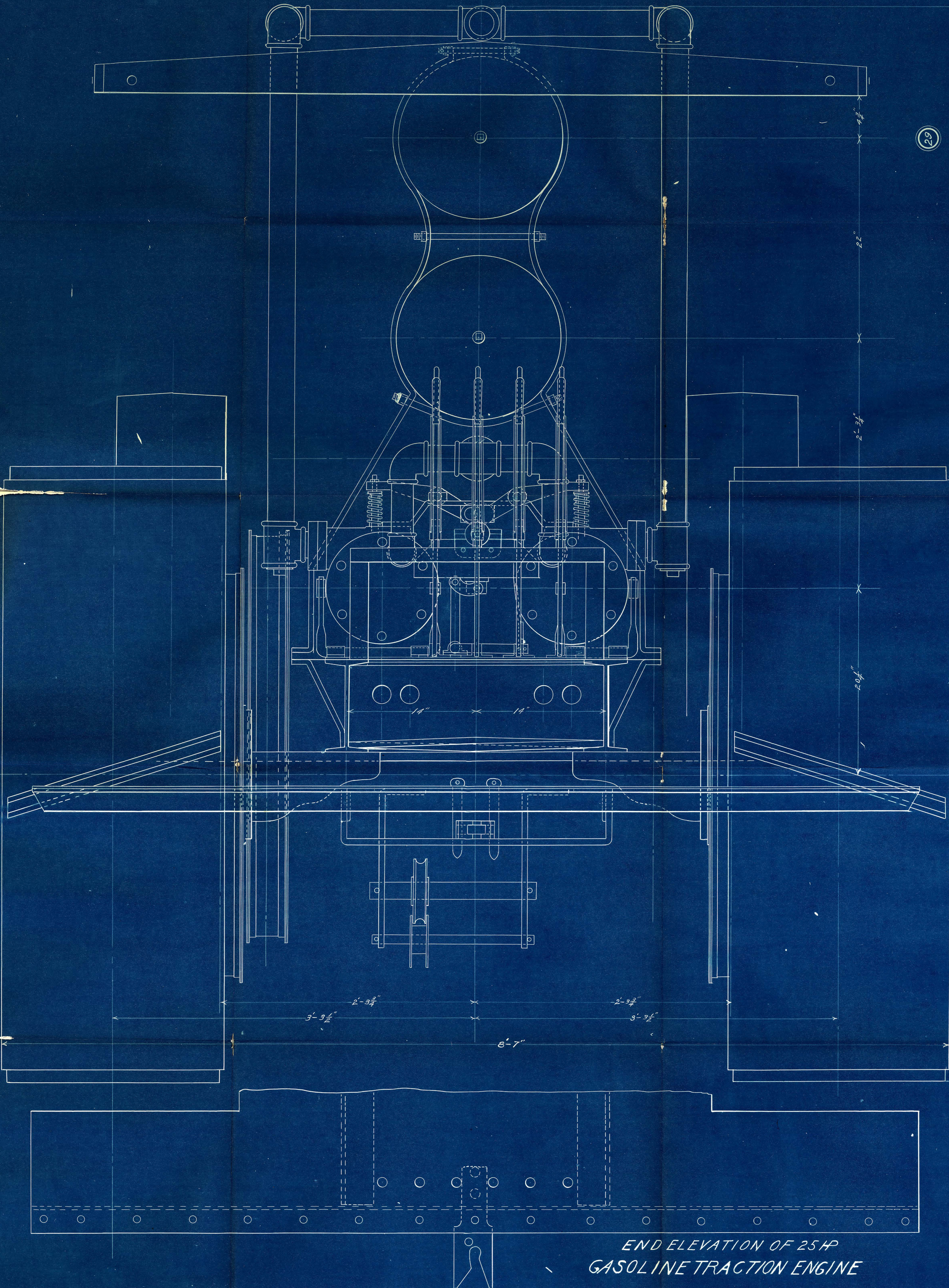


DETAILS OF 25 HP  
**GASOLINE TRACTION ENGINE**  
 SCALE 1-3"=1FT. f MEANS FINISH.  
 DEPARTMENT OF  
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 K. S. A. C.









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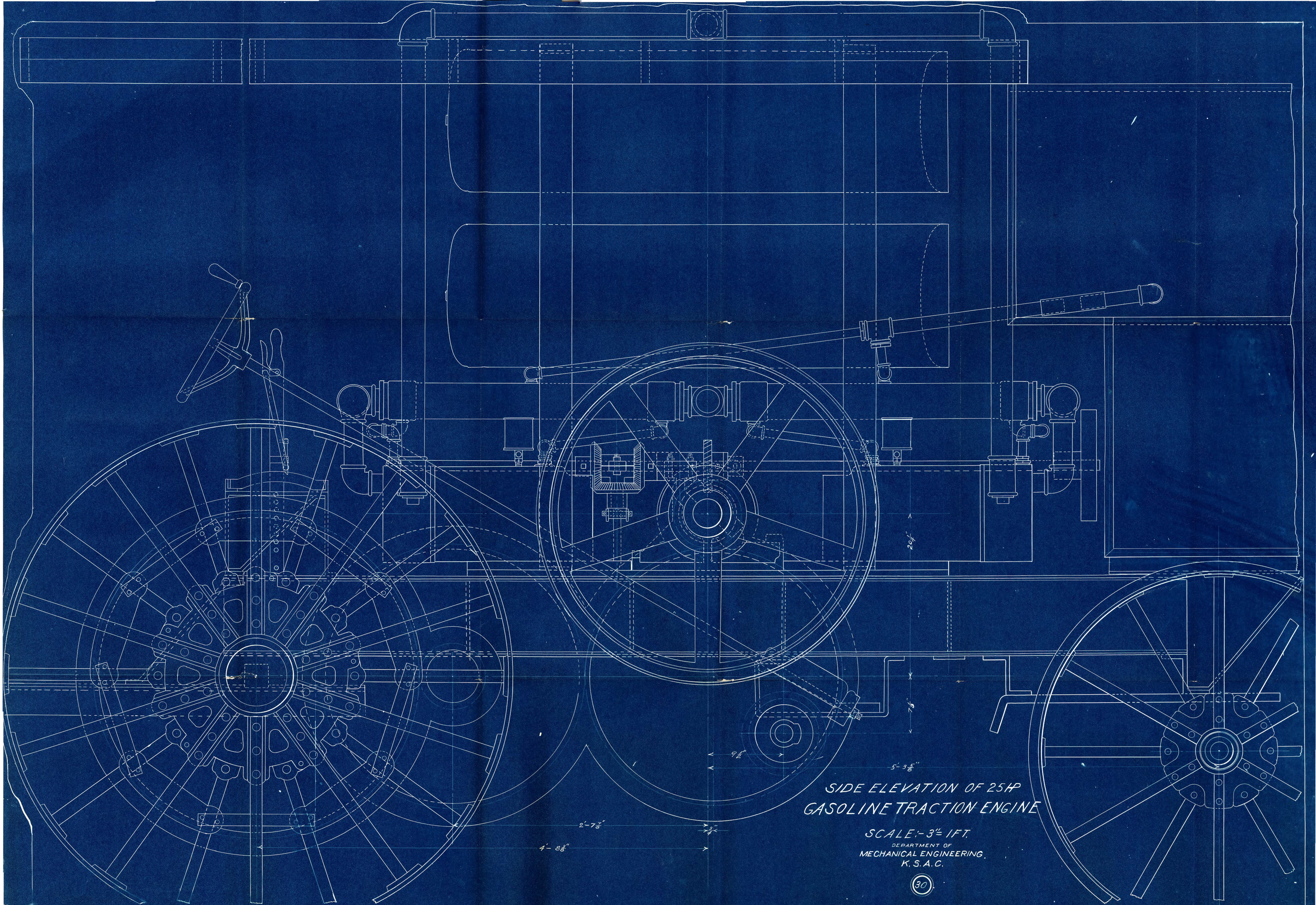
END ELEVATION OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE:- 3" = 1 FT.

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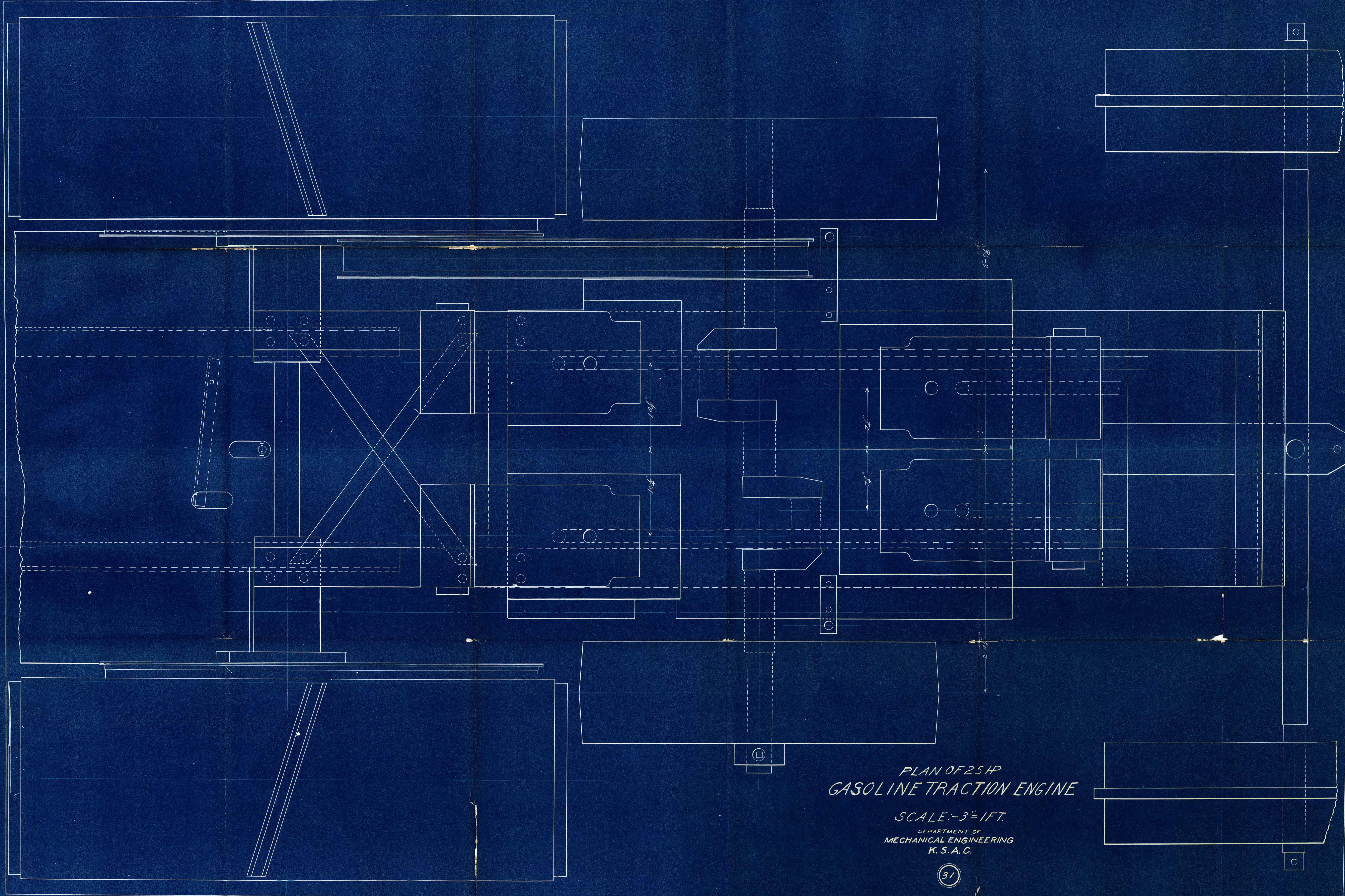




SIDE ELEVATION OF 25HP  
GASOLINE TRACTION ENGINE

SCALE: - 3" = 1 FT.  
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PLAN OF 25 HP  
GASOLINE TRACTION ENGINE

SCALE: 3" = 1 FT.

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