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Report
on
Flexible Rollers and Flexible Roller Path for Main Battery Turrets

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Flexible Rollers and Flexible Roller Path
for Main Battery Turrets.

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FLEXIBLE ROLLERS AND FLEXIBLE ROLLER-PATH FOR A 16-INCH, 3-GUN TURRET

Initial Assumptions

Calculations have been made for the design of a gun turret weighing 1,500 tons. Recoil forces were assumed to be equal to 5,000 tons. The training gear was assumed to be capable of absorbing a turret whip shock of 500 tons and of delivering a driving force of 50 tons. As a part of the design it was considered essential that the bearing should be protected from oscillating forces or any sliding motion of the turret when the turret was not in use.

Calculations and Data

The hollow roller and flexible roller path are apparently a new design. Calculations for the plans submitted are, however, based on commercial roller bearing design practice and equations taken from standard engineering handbooks.

Values for the strength of solid rollers are taken from Precision Roller Bearing Company's catalogue and are in practical agreement with values given by all reliable roller bearing companies. The values used represent allowable loads for continuous low speed operation and since the turret is operated only occasionally, and is subjected to the designed loads only very rarely, the figures should be very conservative.

Kent's Handbook, page 1637, gives the safe load for a solid roller by the equation

$$W = k \ell d$$

where ℓ is the length, d is the diameter and k is equal to 1,000 for hardened steel. It should be noted that values obtained by this formula are about one-fourth those given in the catalogues for slow motion but agree with those given for high speed motion.

Machinery's Handbook, page 343, gives the allowable load on a ring of diameter d as

$$W = 2\pi I/dy$$

where I is the moment of inertia of the cross-section of the ring and y is the greatest distance of any part of the ring from the neutral axis. For a tube with wall thickness b,

$$I = b^3/12 \text{ and } y = b/2$$

the equation for the allowable load then becomes for a tube of length ℓ

$$W = 1.047 \ell b^2 S/d$$

FLEXIBLE ROLLER AND FLEXIBLE
ROLLER PATH DESIGN FOR
TURRET GUN MOUNTS

Design for 16-inch, 3-gun Turret

Design 1. Sheet 1a

This design was prepared to check the mechanical advantage and manufacturability of the flexible roller, flexible roller path and cone worm drive for the main battery of U.S.S. NORTH CAROLINA.

Design 2. Sheet 1c

A conservative design of flexible roller, flexible roller path, with spur gear drive for the main battery of U.S.S. NORTH CAROLINA.

where $\underline{S}$ is the working strength of the material.

If we assume that an $\underline{S}$ value of 100,000 pounds per square inch in the ring equation is comparable with a $\underline{k}$ value of 1,000 in the solid roller equation and solve for a wall thickness that will give a tube strength equal to that of the solid roller, we have

$$1{,}000\, \ell\, d = 1.047\, \ell\, b^2/d \,.\, 100{,}000$$
$$d^2 = 104.7\ b^2$$
$$d = 10.2\ b$$

This solution tells us that a tube with a wall thickness equal to one-tenth of the diameter is as strong as a solid roller of the same length and diameter. This assumption is made in the following discussion.

The Timken Roller Bearing Company has provided the author with two unpublished reports of Stresses and Displacements in Hollow Rollers. The first was submitted in December, 1936, by O. J. Horger and the second, which is a more complete theoretical analysis of the problem, was submitted in May, 1939, by C. W. Nelson.

The equation for the normal displacement of two planes used to load a roller is given in the second paper as equation 257 on page 130.

$$S = (1-\mu^2)/\pi E\cdot F/\ell \left[4 \log (\pi E/2(1-\mu^2)\ \ell^2/F) + 1.545\right] + 2K\ (1-\mu^2/\pi E)F/\ell \quad (1)$$

In this equation, $\underline{S}$ is the change in separation between two parallel loading planes; μ is Poisson's ratio which is assumed to be .3 for steel; F is the applied load; ℓ is the length of the roller; E is Young's modulus, assumed to be 3.10^7 for steel; and K is a constant which is determined by the ratio of the inner diameter of the roller divided by the outside diameter. It is interesting to note that the diameter of the roller does not enter the equation.

Values of K for difference ratios of wall thickness to diameter are given in Table 5, on page 113, of Nelson's report. Values of $\underline{K}$ for use in further discussion and use of the above equation are plotted on a logarithmic scale in plate 1.

Deflections under load were measured for a celluloid disk 3/8 inch thick and 10.75 inches in diameter, and for a ring of the same thickness and diameter with a wall thickness equal to one inch. For a given load the deflection of the ring was twenty times the deflection of the disk.

Loads of 17.6 pounds and .58 pound were required respectively to produce .001 inch displacement.

Poisson's ratio for celluloid is nearly .4, while Young's modulus is about 3.10^5. Substituting these values in equation

(1), deflections of .001 inch would be expected for loads of 10.8 pounds, and .592 pound. These values are within the creep error in celluloid loading measurements. Measurements of the solid disk were subject to a large error.

The expression $1-\mu^2/\pi E$ is a constant for any given material, and for steel this constant is $.965 \cdot 10^{-8}$ when μ is assumed to be .3 and E is assumed to be 3.10^7.

Equation (1) may now be written for steel rollers as

$$S = .965 \cdot 10^{-8} \, F/\ell \, (72.6 + 4 \log \ell^2/F) + 1.93 \cdot 10^{-8} \, K \, F/\ell \qquad (2)$$

The first term in equation (2) represents deflection of a solid roller, and the second term that part of the deflection resulting from the inner boundary or the hollow of the roller. Total deflection is the sum of the two terms, but if the inner diameter is small, the second term is negligible, while if the inner diameter is large, the first term becomes negligible.

The Metallurgy Division of the Laboratory has a 10,000 pound Morehouse proof ring. The width of the ring ℓ is 2 inches; the outside diameter is 5.38 inches and the inside diameter is 4.56 inches. The value of K for equation (2), from Plate 1, is 800. If we assume a uniform ring, the first term of (2) equals .00046 inch and the second term .0154 inch for a total load of 2000 pounds. The total deflection would be .0159, or 35 times the first term, which represents deflection for a solid roller. The loading bosses stiffen the ring so that the measured deflection is .0104.

Three steel loading rings in the photoelastic equipment of the Laboratory are eight inches in diameter and approximately two inches wide with wall thicknesses of .71, .4, and .22 inch. These rings flatten .01 inch for loads of 2,500, 500 and 100 pounds respectively.

The values of K for the first two rings are 430 and 2433 respectively. The third ring is outside the range of Nelson's table.

Deflections calculated for $F/\ell = 1000$ for two similar but uniform rings would give for the first term .00046 as before, and for the second terms, .0083 and .0470. Total deflection for uniform rings would be .009 and .0475. These deflections are equal respectively to 19 and 103 times the deflection of solid disks carrying the same load.

Measured deflections of the two rings are .0075 and .04 inches respectively. As with the Morehouse proof ring, the loading bosses stiffen the rings so that the measured deflections are less than those calculated, as would be expected.

The four cases of measured deflections discussed above all check with calculations made by equations (1) and (2). It is assumed that these equations will give deflections for any type of roller under any magnitude of load within the elastic limit of the material of the roller and the path.

When an outer ring is stretched over an inner ring so that the tension in the outer ring is <u>t</u> the total radial compression between the two rings around the entire circumference is $2\pi t$. This value is used in calculating the relation between load on vertical rollers and tension in the horizontal roller path. 2π times tension in the horizontal roller path and such supporting structure as may be assumed to conform with it is equal to the total compression load on the vertical rollers. In cases where a ring has been shrunk into position, the force holding it from slipping is 2π times the tension times the coefficient of friction. The average coefficient for dry steel on brass is about .2. When this factor is assumed, the force necessary to slip a ring shrunk into place is 1.25 times tension in the ring.

The coefficient of thermal expansion for brass is about $1/1.0 \cdot 10^5$ per degree F. When a ring with cross section equal to <u>a</u> must be heated T degree F. to slip into its seat for a neat fit, tension in the ring at normal temperature will be

$$\text{(Tension on outer ring)} = T\,\frac{1.3\cdot10^7 a}{1.0\cdot10^5} = 130 \text{ Ta pounds} \qquad (3)$$

and the force necessary to slip the outer ring over the inner ring is 160 Ta.

The above calculations and assumptions are the basis of the plan submitted for the flexible roller and flexible roller path ring for the 16-inch three-gun turret mount.

<u>16-INCH, THREE-GUN TURRET ROLLER PATH</u>

Sheet 1 of Plan shows a 1/2 scale sketch of a cross-section of a proposed flexible roller path ring assembly at the left <u>a</u>, a 1/4 scale sketch of the present roller path in the center <u>b</u> and an alternate flexible path <u>c</u> on the right, also 1/2 scale.

<u>Design a</u>

The turret wall 1, shown as a circular plate 1.5 inches thick, is welded to the upper path ring 2. It carried an accurately ground face as an outside guide for the horizontal roller 6 which carries the gravity load of the turret. The turret wall extends about one inch below the top of the lower roller path 5 in order to inclose the rollers in a grease-tight cavity. The upper path 2, which is rolled strip 2.5 by 3 inches in cross-section, is welded to the upper path side wall 3, which is likewise rolled strip and before grinding is 3 by 10 inches in cross-section. The bolts shown connecting 3 to 2 are used only for two loose sections of 3, which are normal to the gun girders, 180° apart, and designed to permit the assembling and inspection of rollers 6 and 7. The lower edge of 2 and the upper edge of 5 are accurately ground and hardened to provide the upper and lower paths for rollers 6, which carry the gravity load and part of the vertical recoil shock forces of the turret. The race for roller 7 is machined and ground partly out of 5 and partly out of 3. Slots for the holding down key 8 are machined half in 3 and half in 5, 8 is assembled through two movable sections similar to those for rollers 6 and 7. The key 8 is of high strength brass made up as short arcs of a circle. The sliding contacts 1 to 5, 3 to 5, and 3 and 5 to 8, are assumed to have clearances of .01. Rollers 6 and 7 are assumed to be hollow and with wall thicknesses equal to one-tenth of their diameter. When subjected to recoil forces, rings 6 and 7 collapse the amount of the clearance and the major part of the shock is taken direct between 3, 5 and 8. At the front of the turret, recoil would drive 1 against 5 to absorb the front share of the recoil.

The pan plate 4 is welded to 3, level with the horizontal roller 7. The shortness of the span, less than 6 inches, from the lower inner corner of roller 7 to the upper outer corner of roller 6, would make this a very rigid construction. The gun girders bridge over from 1 to 4 and stiffen the structure at the points of greatest stress. The weight carried by the pan plate makes the position of 1 outside of roller 6 a better balance for loading 6 than could be obtained by placing 1 above 2.

Ring 5 which has the lower roller path as its upper edge is of rolled steel 3 by 24 inches in cross-section. This is the one part of the assembly which is made heavier than the corresponding part of the present assembly. The three inch thickness is carried down low enough to offer solid reliable support for the bases of the training worm drives shown in sheet 2, or as a foundation for the training rack as shown in sheet 1c.

The stool carried accurately ground surfaces for both roller paths and an accurately machined groove for key 8. The ring collar 10 is welded to 5 and is accurately machined on the upper surface as a base seat for block 9.

Block 9 is of brass and is approximately 1.5 x 2 inches in section, and is machined in arcs of a circle about 2 feet long. It is not quite rectangular in section, but is slightly wider on the inner side so that when it is in position it forms a secure lock between 3, 5 and 10.

Assembly of the Roller Path

The routine of assembling the roller path ring would be as follows: The horizontal path for 6 on 5 would be ground plane marked on ring 10 and lugs B similar to C are welded to 10 to be opposite C for this position, as shown on sheet 2b. Lugs A, which are beveled as shown, are now welded to 10 at the original position of lug C.

With the turret in the position shown in sheet 2 the 24 blocks 9 are slid through the 24 slots and left resting on 10 and against 5. If the turret is now rotated counter-clockwise, block 9 is held by lug B, while the turret wall slides over it until contact with the left end of a lug C with the right end of a block 9 locks it on the center line of the ship. If the turret rotates clockwise, the right end of lug C strikes the left end of block 9 and by means of the beveled face on lug A shoves it out of the slot, clearing the turret for action.

Each individual block 9 should be ground to fit its individual position so that when all blocks are in position, and the turret is at rest, any block can be slid in its seat by a two-ton thrust. That is, a 24-ton thrust would start the turret from rest in its locked position.

Each block 9 would probably be chained to its position on ring 10.

16-INCH- THREE-GUN TURRET TRAINING GEAR

A reduction of the main battery roller path assembly from the dimensions shown in sheet 1b to the dimensions shown in sheet 1a would make a change in training assembly very advantageous.

Design

A new design for training assembly substituting worm for spur gear drive is shown in Sheet. 2. This is a radical change not only in appearance and mechanical design, but also in its function with reference to shock stresses. Experience in turret design has shown that the spur gear, in spite of its tremendous weight, must be protected from recoil stresses, and cannot be used normal to the direction of gunfire. The driving mechanism is therefore placed forward, as near the line of fire as possible, so that recoil distortion separates the driver from the rack. It could not be given this protection if it were mounted on the stool. The spur gear is protected from turret whip due to the firing of a single wing gun and from shock resulting from the impact of an enemy projectile by a release in the driving motor. The worm gear, as shown in the plan on page 2, is designed to absorb the shock of recoil whip, or the impact of enemy projectiles with no release. The driving motors are mounted on the turret stool. Two motors are used as in the present drive. Each motor drives two worms and each worm is designed to absorb a shock of 500,000 pounds. If the worms are to absorb this shock, the gear must be capable of delivering that load, therefore the worm gear is designed as a continuous ring. It is believed that flexibility of the entire assembly is great enough so that loads greater than one half million pounds would not be put on any one worm, or a load greater than one million pounds on the worm gear.

Assembly of the Training Gear

Sheet 2a shows how two driving motors are mounted in the center line of the ship fore and aft on the turret stool. Sheet 2c shows how these motors through pinion 23, shaft 22, and two bevel-cut pinions 21, drive the worm pinion 16. The worm 14 is supported for radial forces by needle bearing 15 and for end thrust by roller bearing 17. The worm must be lifted into place from below, therefore castings 24, 25, 26, and 27 must be designed to allow the worm to be lifted upward and in toward the gear at an angle of 60° as the worm and gear mesh. This would include a backing for bearing 15 that would allow for adjustment of the worm into its final seated position. Plates 18 would be machined and ground for longitudinal adjustment of the worm and bearings 17. Clearance between the end of the shaft on worm 14, and spacing disk 18 should be made very small, less than .01, in order that excessive shock, such as whip from gunfire, would be taken by direct contact between the shaft and the washer. The comparatively small thrust load 10 to 30 tons of training control would be supported by thrust bearing 17, but flexibility of the rollers would allow thrusts in excess of 50 tons to be taken directly from the shaft through spacer 18 to the base castings 24 and 26.

The worm No. 14 would drive the circular worm gear 13, which is shrunk into place on circular support 11, which is welded to the underside of the pan plate of the turret.

The dimensions of 11 and 13 could be varied over a fairly wide range to obtain the distribution of recoil forces considered most satisfactory. In this plan 13 is assumed to have a cross-sectional area, free of the gear teeth, of 25 square inches and to be able to sustain a tension load of 1,000,000 pounds. It is assumed to be shrunk into position with a temperature difference of 150° F. between 11 and 13. If one-third of the yielding were in 11, and two-thirds in 13, the tension in 13 at normal temperature would, by equation 3, be 14,000 pounds per square inch. The total tension would be 175 tons and the force necessary to slide 13 over 11 would be 220 tons, assuming there was no binding action resulting from the way in which the shearing force was applied. The shearing force would be applied through worms 14 and binding action would be great so that the force needed to slip 13 over 11 would be much greater than the 200 tons calculated. The area of contact between 13 and 11 would be about 5,500 square inches. A shearing force of 220 tons would therefore give an average force of 80 pounds per square inch. There would be little danger of this resulting in seizing between the two surfaces. However, the two contact surfaces should be given a polish finish in order to decrease the danger of seizing.

11 is shown 2 inches thick and extended 5 inches below 13. The support of 13 is both strong and flexible, while the weight is not excessive.

The castings 24 and 26 would need to be strong and solidly welded to the stool through lugs 20 since they would be required to resist shock forces as great as 1,000,000 pounds.

Strength of the Design

Data for the roller paths are shown on Sheet 1. It is assumed that the rollers will require no guides since they will be closely confined between polished walls and can be packed in grease. Figures for the strength of the rollers are taken from Precision Bearing Company catalogue and are in agreement with commercial practice.

Horizontal Path

The normal load on the horizontal roller is less than one-fourth the rated capacity of the bearings. If the clearance between 3 and 5 through 8 were .01 inch, the turret would move on roller bearings until the vertical force was 85 per cent greater than the normal gravity load. The maximum load on the rollers would be 8,500 pounds, or only 43 per cent of the load which solid rollers should stand under continuous service. Vertical contact between 5 and 3 through 8 would be 6 square inches to the linear foot. The strength of this contact should exceed 150 tons per linear foot.

Vertical Roller Path

The vertical rollers are designed to hold close tolerances wherever movable parts of the turret face stationary parts. For example, between 1 and 5, between 3 and 5, and between the worm and the worm gear in the training mechanism of the turret. These tolerances can be kept low because pressure on the vertical rollers will stretch or warp the lower path ring to a near perfect fit with the upper path ring. The lower ring in the plan submitted is designed .05 small. Pressure of 1,680 pounds each on the vertical rollers would stretch lower ring to fit. If the upper ring were slightly elliptical, the lower ring would, of course, conform to its curvature. A difference of .25 inch in the major and minor axis would be negligible. A clearance of .01 in the vertical separation of 3 and 5 would mean that the turret would have roller bearing until the side thrust on the turret was above 500 tons. This is the lateral thrust for a 20° roll or list of the ship, and is thought to be sufficient. In this condition there would be direct contact between 3 and 5 and the area of this contact would be 36 square inches per linear foot. The strength of this contact would be more than 1000 tons per foot. The maximum load on the vertical rollers would be 4,780 pounds, or 40 per cent of the allowable continuous load for a solid roller.

Maximum stress for the hollow rollers calculated by the equation

$$S = Ld\ y/2\pi I$$

where L is the load, y the maximum distance from the neutral axis, and I is the moment of inertia of the cross-section of the ring, is as follows for the horizontal and vertical rollers respectively; 88,000 and 54,000 pounds for the normal loads and 162,000 and 152,000 pounds per square inch for the maximum loads.

The lower roller path ring base 5 is 3 inches thick and about 24 inches wide. This provides a wide stiff collar well designed for distribution of recoil stresses to the top of the turret stool.

The key 8 is designed as a holding down clip. It should not need more than .01 inch clearance but should be ground to float freely. It would be loaded only a few hundred times during its life and should withstand a shock load of 25,000 pounds per inch, or 150 tons per foot. This should give it ample strength.

Block 9 was included in the plan because commercial experience has shown that vibration stresses are far more dangerous to bearings than shock stresses. Turret bearings remain locked in one position while they are subject to continuous vibration for months at a time. Blocks 9 are set into their seats on ring 10, as shown in Plate 2b, counter-clockwise rotation of the turret, then lock it into rest position on blocks 9. It is assumed in the plan that the force necessary to start the turret from rest would be one ton per block, or 24 tons total. This is one-half the driving force assumred for the training motors. If the co-efficient of friction to start the turret turning on blocks 9 were .04, a high factor for greased surfaces, the vertical load on each block would be 25 tons and the total load on the blocks would be 600 tons, or 40 per cent of the total load of the turret. The pressure on blocks 9 would be 833 pounds per square inch. With care this would not cause seizing.

The brass worm gear 13 would have a cross-section of 25 square inches, and is considered to have a tensile strength of 1.2 million pounds. It is assumed to be under a structural tension of 14,000 pounds per square inch. At any time when one part of the ring is subjected to a very high tension there will be no tension in the part of the ring diametrically opposite, therefore, the total tension strength of the ring will resist the shock forces.

A million pounds tension in the ring is equal to 40,000 pounds per square inch. The deformation associated with this stress would be one twenty-fifth of an inch per foot. Since the clearance between 3 and 5 is only .01, it would not be possible to load the gear 13 with stresses of 40,000 pounds per square inch. The gears could be given a heavy load only by turret whip. In this case all four worms would be equally loaded and excessive forces would result in slippage between 11 and 13.

It should be noted that there would be local slippage between 11 and 13 each time when a turret gun is fired. It is therefore necessary that the contact surfaces be given a very smooth finish.

The worm 14 is 52 inches long with circular pitch of 4.5 inches. It will therefore have contact with 11.5 teeth and since each tooth is 5 inches wide, the total length of tooth contact will be 57 inches for each worm. Lewis' formula for 14-1/2 gear teeth with circular pitch of 4.5 inches gives the tooth strength as 13.5 tons per linear inch for material with a working strength of 50,000 pounds per square inch. This would give an overall strength for the 57 inches of tooth contact of 770 tons for each worm, or 3,080 tons for the four worms. These calculations are made for the brass teeth of the worm gear and should be conservative, since the teeth are on the outer circle of the worm. The worm itself would be of steel with much greater tooth strength.

The figure 50,000 for brass is high, but that stress would only be required to resist the turret whip resulting from the impact of an enemy projectile. Recoil whip resulting from the firing of a single wing gun would amount to about 500 tons at the training gear. This load divided by four would give a force of only 125 tons to each worm or barely over one-fifth of the rated capacity. The worm which has a diameter of about 8 inches at the bottom of the teeth could resist a radial thrust of 270 tons, but in taking this load it would bend .4 inch. The small clearance at the roller path and flexibility of the supporting ring 11 would make it impossible for a load of any such magnitude to be communicated to the worm gear as a direct thrust coming from the pan plate.

Recoil whip would of course place a radial thrust on the worm. If the standard 14.5° slope is assumed for the gear teeth, radial thrust would be a little more than one-fourth of the tangencial load along the gear if we assumed zero friction between the gear teeth.

Recoil whip from the firing of a single wing gun would, neglecting friction, place a radial load of 32 tons on each worm. This is less than one-eighth the strength of the worm and would give a bending of the worm away from the gear of .047 inches. Friction between the gear teeth would reduce the radial thrust, and the actual displacement of the worm away from the gear would be less than .04 inch.

A needle roller bearing would be used at 15 because of the small clearance. This bearing is on a 7-inch shaft and would be about 12 inches long. It could take a shock load of half a million pounds and should be seated in a foundation capable of resisting that load.

The worm teeth on 14 should have a lead of 10°. This makes them practically self-locking against shock. Gear 16 is designed 5 inches wide and 14 inches in diameter. If we assume a tooth strength of 15 tons per inch, 16 would stand a shock load of 75 tons. This would be equivalent to 110 tons at the pitch diameter of the worm. Spacer 18 and base casting 26 should be designed for a shock load of 500 tons.

The rate of slippage of worm 14 over gear 13 for full speed training would be about 6 feet per second, or 360 feet per minute. The worm might be lubricated from a splash pan placed under the worm, but it is believed more effective lubrication could be obtained by enclosing the worm in a hood and introducing an air pressure atomizer which would fill the entire hood with a spray of very light oil. The atomizer could draw the oil from a splashproof tank in the bottom of the hood.

Conclusions

The simplicity, ruggedness, and economy of both weight and space show a very great advantage of flexible rollers and flexible roller path over the present design. The use of small rollers and simple cross-section for the roller path offers a possibility of higher quality of material and better heat treatment. Commercial bearings set a standard of over 600 Brinell for hardness of both rollers and path. 200 is considered satisfactory for the present ordnance rollers. Rolled welded strip for the paths can be produced with greater uniformity and greater tensile strength than the present paths. Continuous circles such as are used for all parts, upper path, lower path, holding-down clip, vertical roller path, training gear and training gear foundation, will give better distribution of forces and greater flexibility with greater economy of material than can be obtained by any other arrangement. Flexibility is imperative where very severe shock forces are to be encountered. Finally the entire assembly can be packed in grease, thus giving better protection from dirt or moisture and better lubrication than can be obtained with an open bearing.

The principal advantage of the vertical rollers and the flexible lower roller path is that this design will reduce clearances and relative displacements between the upper and lower roller path rings, thus holding recoil forces to a minimum. The vertical roller transmits horizontal thrust from the pan plate to the lower roller path ring without torque. This was the problem which initiated this entire study.

The principal advantage of flexible rollers is the change in the need for close machine tolerances. The flexible roller will allow recoil forces to be taken by direct contact between the upper and the lower roller path ring. This advantage is secondary. Solid rollers are stronger than hollow rollers and solid rollers could be used where hollow rollers are shown in the drawings if the load could be symmetrically distributed. They could take the full shock of recoil forces without the help of direct contact between the upper and the lower roller path rings if those rings were true planes and remained true planes. Equation (2) shows that a load of 1000 pounds per inch gives a distortion of .00046 inch and that a load of 20,000 pounds gives a distortion of .0071 inch to a solid roller. Twenty times the load gives only fifteen times the distortion. Equation (2) shows that for a given load the distortion of the hollow roller with wall thickness equal to one-tenth the diameter is fifteen times the distortion of the solid roller. Therefore, a given irregularity of the roller paths will produce in the solid roller 20 times the stress it would produce in a hollow roller.

If the only irregularities between the upper and the lower roller path were the tolerances of the original machining, small solid rollers could be used and the tolerance of .005 inches between upper and lower roller paths would not overload them, but temperature irregularities and movement of the ship introduce changes much greater than that. A line 400 inches long, about the diameter of a turret, on a deck which is alternately stretched and compressed 10 tons per inch by hogging and sagging stresses in the ship would change its length .52 inch with each swing of the stresses. This is 104 times the allowable tolerance. Such deformations are at all times changing the shape of the turret stool. A temperature difference of 10° F. for 100 inches would give a deformation of nearly .007. This would be sufficient to load certain solid rollers 20,000 pounds per inch. Stresses of this magnitude might damage the solid rollers, and they would be very apt to damage the roller track. The same irregularity or deformation would place a load of only 1,000 pounds per inch on hollow rollers and this load would be within the design stress limits for both the roller and the tracks. As a particular example, the 2.5 inch roller shown in the plan would be loaded 1800 pounds per roller by track irregularities of .005 inch, while a solid roller of the same size would receive a load of 36,000 pounds from the same variation. Flexibility of the hollow roller will protect both itself and the track from excessive loads due to irregularities in the track.

Conical worm gears are accepted everywhere in industry as the gear best adapted to withstand shock. It seems to be ideal for turret training drive.

All parts of the roller path ring assembly, including the holding-down clips, where there is possible friction are packed in grease. The coefficient of friction for a good grease lubricated bearing is .003. The entire weight of the turret moving with this coefficient would require only a 4.5 ton force to maintain sliding friction. The ideal clearance between the upper and lower roller path rings would be zero. Rollers give low starting friction, but after surfaces are in motion the difference between good lubrication and roller bearing is negligible. The design shown has the low starting friction of a roller bearing but the compactness and ruggedness of a grease bearing.

Recommendations

An estimate of the possibilities of flexible rollers and flexible roller paths requires more exact knowledge of the manufacturing possibilities, the physical properties, and the fatigue characteristics of flexible rollers and of the entire flexible roller and flexible roller path assembly.

Test of Flexible Rollers

It is recommended that tests be made with rollers 2 inches in diameter by 2.5 inches long. This is about the standard size roller for extra heavy duty.

It is suggested that tests include failure under static load, deflection under static load, and fatigue tests under oscillating load for the following wall thicknesses, solid, 1/4", 1/5", 1/6", 1/8", 1/10".

Test of 5-inch, Two-Gun Turret

Model tests under conditions approximating service usage are very desirable. It is suggested that a 5-inch two-gun turret be mounted on a roller path similar to that shown in the plan but with all dimensions reduced to about one-fourth those given. The base of the present turret is a solid casting. Flexibility of the lower support is a necessary part of the design, therefore, it is suggested that tube support at least 12 inches high be substituted. The present mount for this turret consists of a double roller path, but the vertical rollers for the horizontal thrust are above and outside of the horizontal rollers for the vertical thrust. This arrangement introduces torque in the supports and should be avoided.

The purpose of the design which is being considered is that of combining the low starting friction of the roller bearing with the compactness and ruggedness of the grease bearing. The flexible roller tests outlined above could, with little additional expense, be extended to include grease bearings by machining a bronze bearing ring to replace each set of rollers. These rings should have a master grease groove on the back around the entire circle so that one grease gun fixture would serve the entire ring. From this master groove, holes should be bored through the ring leading the grease to distribution spider grooves on the bearing surfaces. This is suggested as a practical means of testing clearances which should be used between 2 and 5 and at 8.

Discussion

The above plan has been discussed with a number of different people and organizations as follows:

January 31, 1931, at S.K.F. Plant in Philadelphia
First Assistant Engineer J. S. Tawresey

Argument

The flanged roller is a very bad design.

Commercial practice in roller and roller path design call for a Brinell hardness of over 600. Softer rollers can not be good design. Designs for all ordnance bearings should be standardized by bearing engineers.

Shock forces are very unimportant as a cause of roller failure. Practically all roller bearing failures result from vibration or oscillating forces while the bearing remains fixed in one position for some considerable time. This has been called fretting abrasion.

Examples offered:

Automobiles shipped by train arrived at their destination with ruined bearings.

Eight sewing machines were attached to the steel top of one factory table. One machine was used while seven were idle. The bearings of the seven idle machines were all ruined.

Conclusion reached:

The bearings were destroyed by abrasion and not by work hardening. The abrasion is caused by movements that are of molecular dimensions and has little relation to ordinary sliding friction.

The most important problem of ordnance bearing design is a better understanding of the problem of "fretting abrasion."

Suggested methods of attacking the problem:

A committee of three engineers experienced in the inspection of used roller and ball bearings in commercial service should be given a group of bearings which had been subjected to service tests. They could tell what changes in design were needed to give longer better service by measurable defects in the used bearings.

March 3, 1941 - Timken Roller Bearing Company, Canton, Ohio
Vice President T. V. Buckwalter

Argument

The flanged roller is very poorly designed for resisting horizontal thrust.

200 Brinell hardness for roller bearing material is, for either the rollers or the track, entirely out of line with all modern practice. There is a 1000 to 1 improvement in going to 600 Brinell.

All bearing design should be in the hands of specially trained bearing engineers. Companies such as Baldwin's Locomotive, New York Central, or Santa Fe Railroad, even the automobile companies, do not design the bearings they need. They take the design from the different bearing companies. Ordnance designers call for hundreds of special bearings which are unnecessary. Ordnance bearings would be more efficient, they would be lighter and they could be manufactured in much less time, if they were standardized by trained bearing engineers. The Navy is requiring that certain bearings be chromium plated. Commercial experience would indicate that the plated bearing will be less efficient and have a shorter life than the unplated bearings. The danger of fretting abrasion injury to bearings can be very greatly decreased by giving a smoother finish to the bearings and by using specially prepared lubrication. The greases sold at the average filling station are entirely unsatisfactory for extra heavy duty bearings, because they contain too much soap. The life of a heavy duty bearing with standard automobile grease lubrication can be increased ten to one by correct lubrication.

A new bearing has certain irregularities in the finish of the surfaces. The first effect of fretting abrasion is that of rubbing off these irregularities. This leaves a smoother surface, but the particles rubbed off at once begin to act as an abrasive, thus starting fretting abrasion. A smoother initial surface may prevent the start of the action.

Mr. Buckwalter offered several examples of commercial roller bearings.

A roller 4 x 4.5 after five years' in a steel roller mill showed no signs of wear. The bearing fit on the roll neck of a 90-inch mill which was designed for a load of 100,000 pounds per inch, or 9 million pounds. This was the average load for each pass. Accidental trouble in the rolling as when a "fish tail" doubles back had subjected the bearings to several times the design load without injury to the bearing. The 4 x 4.5 roller exhibited was tested between the hardened bed plates of the testing machine with a load of one million pounds with no injury to the roller.

A one-inch roller 1.5 inches long was exhibited which Mr. Buckwalter was sure would carry the recoil of 16" gun trunion. The claim was made that this roller would stand a shock load of 100,000 per inch of its length. The bearing in which this roller was used was designed for use on old locomotives. Plane bearing of the old locomotives were cut away and new roller bearings were inserted in the space required for the plane bearing. The suggestion was made that the same change could be made on gun trunions with very little trouble.

These two examples are offered to illustrate the difference between ordnance and commercial bearing practice.

Static tests made in the Timken laboratories showed that solid rollers are stronger than rollers with even a small hole through them. The hollow rollers always failed in tension from the inside out and in line with the applied load.

Conclusion

The vertical roller for close clearance and horizontal thrust was assumed to be sound design practice.

The flexible roller as a means of decreasing the importance of close machine tolerance was sound theory and worth a test.

The President of the Company backed Mr. Buckwalter in offering the facilities of the company to support any plan for tests of the flexible roller and flexible roller path design which the Navy Department might wish to make.

Mr. Buckwalter provided the author with copies of two unpublished reports of stresses and deformations of hollow rollers, a total of 200 pages.

March 5, 1941 - David Taylor - Model Basin, Carderock, Maryland
Commander W. P. Roop

Commander Roop expressed the opinion that a plan for testing the flexible rollers and flexible path ring should be developed with the advice and cooperation of the Timken Company.

The suggestion was offered that the author should visit the Philadelphia Navy Yard. Arrangements were made for inspecting the model turret test work in progress there and for consultation with personnel at the Yard in charge of turret design and construction.

Commander Roop suggested that a conference with a Timken representative might best be arranged for the Philadelphia Navy Yard.

March 7, 1941 - Navy Yard, Philadelphia, Pennsylvania
In the Drafting Room

A. M. Stefano expressed the opinion that the design of flexible rollers and flexible roller path was theoretically sound. He offered the suggestion that the entire path rollers and race be designed and built as a complete self-contained unit which would be seated on the stool much as any commercial bearing is seated for the load it will carry.

C. J. Lissenden said that a change from the flanged roller was desirable and that he would like to see the design tried with the 5-inch two-gun turret complete with all details. His idea was that the construction of the 5-inch turret model would be a test of the manufacturability of the full scale turret. It would tell the tolerances that could be attained and the clearances that would be necessary for the 16-inch three-gun turret.

J. L. Morehouse had suggested a flangeless roller running in a groove because he recognized the need for a change from the flanged roller. He approved the flexible roller as a means of decreasing the importance of machining tolerance.

The three men mentioned above said there was little object of a conference at their Yard with commercial bearing designers unless they first got the go-ahead signal from Washington.

Commander T. L. Schumacker suggested that the training motors might be placed normal to the center line of the ship.

Jim Smith, turret shop foreman, said the removable blocks in the present roller path would be very unsatisfactory if it was ever necessary to use them. He said the present holding-down clips were a very difficult shop job and the simpler key of the present plan would be a great improvement. His final suggestion was that a track 2.5 inches wide could be ground with greater accuracy than one 17 inches wide.

March 11, 1941 - Navy Department, Washington, D. C.
Commander C. D. Wheelock

Study of the design for flexible rollers and flexible roller path for the 16-inch three-gun turret should continue until it is in shape to be presented as a concrete plan to the designers in Ordnance.

Plans for the present design of flanged rollers should be studied and a plan should be prepared which includes the equivalent of the present training gear.

Some plan should be made for inspecting both sets of rollers and it should be possible to remove rollers from either set from inside the turret.

It was very desirable that changing of the rollers should be accomplished without lifting the turret and without making a break in the roller paths.

The suggestion was made that a plan be drawn for a bearing which showed a less radical departure from the present flanged roller design.

March 26, 1941
Dr. F. M. Walters, Metallurgical Department
Naval Research Laboratory

The machining of steels at 400 Brinell is now common shop practice. Roller paths of this hardness could be obtained by using forgings which were given this hardness as a part of their original fabrication. This would be far superior to the present 200 Brinell races.

Flame hardening is now common shop practice. If the upper and lower roller paths were fabricated from a steel satisfactory for flame hardening, it should be possible, if sufficient care were taken, to develop a technique which would give a very uniform heat treatment to the races and leave them with a hardness equal to that of the smaller commercial bearings.

Age hardening alloys are now being developed at the Naval Research Laboratory that are capable of being hardened after an initial slow cooling from the solution temperature (1800°F). These alloys after the first cooling from the solution temperature are relatively soft, about 300 Brinell, and can be machined easily. After machining, the alloys can be hardened by heating to 1000°F for about one hour. As the alloys have a very small volume change on hardening, the tendency for distortion is minimized. One of the alloys under study has the following physical properties:

Tensile strength	-	230,000 psi
% elongation	-	12.5
Reduction area	-	40.0
Rockwell "C" Hardness	-	46 $\approx$ 460 Brinell

The limitation of these alloys has not yet been determined, as more study is needed to determine the slowest cooling rate that is capable of hardening the alloys.

April 2, 1941 - Army Ordnance, Washington, D. C.
Colonel G. T. Jenks

The design is radical. It is worthy of study.

The pre-stressing of a lower flexible ring to reduce clearance is sound theory. Reduced clearances mean reduced recoil forces. The simplicity of the design makes it attractive and offers possibilities of manufacturing advantages. The Army has found that ordnance bearings must be protected from vibration while in transport.

The plan should be discussed with Mr. Dabrasky in charge of anti-aircraft design, Mr. D. A. Gurney, in Mobil Carrage Design, and with Preston Tucker of the Tucker Mfg. Co.

E. E. Honsberg

The 8-inch Army railway mounts are designed with jacks to lift the load from the bearings when the gun is in transport.

The Army is not interested in reducing weight by introducing higher strength steels in stationary mounts because weight is needed to absorb recoil.

The Army has had no trouble in obtaining either large rollers or heavy castings for the paths of the large rollers.

Compactness of the design **shown** would not be an advantage in lubrication because any lubricant used might freeze, making training of the gun more difficult.

Army guns do not need vertical rollers for horizontal forces or to reduce clearances because the largest coast defense mounts operate with a horizontal clearance of only .015 inches. This clearance is not difficult to machine and has made no trouble in operation.

Clearance of .01 inches should be easy to attain with the flexible path and should not give trouble in operation.

Independent roller races would probably be superior in strength to hardened faces on structural members. Independent races should be easy to manufacture and give no trouble in operation if they were properly seated.

Tom Conlon

Army designers make no objection to the use of rolled welded strip in roller races but Army procurement has had sad experience with manufacturers who promised that ordnance equipment could be manufactured in new and better ways and then could not deliver the goods.

The simplicity of the design shown **suggests** considerable advantage in production.

No plans could be made for the introduction of such a design until fatigue studies of hollow rollers have been completed.

The fabrication of small roller races of high quality steel has proved difficult because of warping of the race rings.

Analysis of Discussion

The problem of protecting bearings from fretting abrasion by relieving the bearing from oscillating loads was discussed with the representatives of S.K.F. and Timken companies.

Representatives of both companies cited examples of bearings which showed "fretting abrasion" after railroad shipments with very light load on the bearing. It is possible that because of sad experiences with bearings ruined by fretting abrasion these representatives have labeled as "abrasion" marks on bearing surfaces which were not abrasion but polish, and therefore harmless.

A discussion of the problem of bearing abrasion with representatives of the Bureau of Ships, the Bureau of Ordnance, and with officers acquainted with the handling of ordnance equipment on shipboard has revealed a wide divergence of opinion on the subject. Commercial designers are unanimous as to the importance of fretting abrasion, but differ widely in their suggested means for preventing it. It is very important as a problem in ordnance bearing design that more exact knowledge be obtained about fretting abrasion and means of avoiding it.

Sheet 1 shows in c a cross-section of a roller and race design in general similar to the one presented in a, but with larger, longer rollers and with the training rack attached to the stool.

It was desired to increase the width of the track and the length and diameter of the rollers, and to arrange that rollers might be changed from inside the turret. With a two-inch roller two and a half inches long, this would not be difficult with a hollow tapered roller. The taper would be only .01 inch per inch, or .025 for the roller. A high grade spring roller would not be injured by a deformation of this magnitude. On the other hand, a bearing three inches in diameter and six inches long would taper .015 per inch, or a total of .09 inch. A deformation of this magnitude might injure the roller or the track; therefore, a straight roller cut into three sections is shown in the sketch. The Precision Bearing Company have many thrust bearings of this type in service and they have given satisfaction under rather high speed service. There should be no trouble with this design in the very slow rotation of the turret support. Each roller is cut into three sections. Each section, of course, rotates at a slightly different rate and the different sections would need to be kept in line by a tapered guide, not shown. There would be about 1230 of the 2-inch wide rollers in the circle. The average gravity load for each roller would be 2,440 pounds. The Precision Company rating for the roller would be 24,000 or ten times the normal load.

The numbers and functions of all parts are the same as in the original plan a up to 11, which replaces the lower part of the original 3. 12 is a circular rolled strip 3 inches thick, welded to 5 to give added width to the lower roller track and a locking cap for the training rack 13. The lower edge of 12 is beveled to form a lock for the rack. The base support 14 for the training rack is welded to 5 with sufficient clearance to allow the different sections of 13 to be seated against 12. Key 15 is then forced into plane under high pressure, securely locking 13. After 15 is seated it is welded to 14, thus insuring a permanent lock for the training rack. b in Sheet 1 shows a cross-section of the present flanged roller with the upper and lower tracks, the rollers with their cage, the holding-down clips and training gear. This section is shown for comparison with the original plan a and c which are drawn to the same scale.

Summary

The compactness of the flexible roller and flexible roller path design gives an increase in clearance between the greatest rotating radius and the barbette of 10.75 inches. It allows an increase in the radius of the outside of the training rack from 15'5" to 16'6", a gain of 13 inches in radius and of over 7 per cent in the mechanical advantage of the training gear.

The total weight for the 2.5 inch roller path including rollers, holding-down key, the training gear and its support is approximately 30 tons. This includes ring 5, which is 24 inches wide. The 6-inch roller ring assembly would weigh about 50 tons. The total weight of the flanged roller path assembly with the upper and lower track, the holding-down clips, the rollers with their cage, the training gear and its support would be nearly three times the weight of the 6-inch roller path, or about 5 times the weight of the 2.5 inch path. The estimated weight saved by the use of the more compact path would therefore be well above 100 tons per turret, or 300 tons per ship. In spite of this decrease in weight the flexible assembly would be stronger because of its compactness and because of its flexibility. In case of damage from enemy projectiles the upper and lower rings would support each other, any forces tending to distort one ring would be resisted by the other. The plan presents a smooth exterior rotating surface with no projections which might be fouled by battle damage. This should be a distinct advantage over the present assembly shown in b. If advantage were taken of the 10.75 increase in clearance between the rotating turret and the barbette, the diameter of the barbette could be reduced by 21.5 inches, while the present clearance was maintained. This would allow the saving of a trip of barbette armor 5.5 feet in width. The decrease in diameter would increase the strength of barbette. The total saving in weight should be about 200 tons per turret.

ADDENDA

April 8, 1941 - Army Ordnance

Mr. Dabrasky: Anti-aircraft.

The cone worm drive is very satisfactory. It is more efficient, it is stronger, it gives smoother, more positive control of training than the spur gear. Tests have shown that the cone worm drive gives 100% area contact on the faces of the gear teeth. There is no theoretical objection to a cone worm drive of the dimensions shown in Plan F-411. If alignment can be maintained between the worm and the rack, cone worm drive should be superior to spur gear drive.

C. E. Burns: Railway Mounts.

Brinelling is a very important problem for the designer of turret bearings. There has been much trouble from this in the past. Super finish grinding is very effective in preventing brinelling. A polish finish is bad.

The use of hollow rollers for bearings was investigated by the Bureau of Reclamation. Their decision was against the hollow rollers. Their records are available for anyone interested.

The cone worm drive is very satisfactory if alignment can be maintained.

April 8, 1941 - Navy Department

Lieutenant J. H. Ellison

The 6" three-gun turrets for light cruisers, diameter 14'9.5" to the center of the tracks, have been used as construction models for 16" three-gun turrets to be built later.

The flexible roller and pre-stressed flexible roller path ring might be designed and constructed for the 6" turret. This would be a far more satisfactory test than that of the 5" anti-aircraft mount. If the design were tested with a 6" turret, half scale models could be given static and dynamic tests at the Philadelphia Navy Yard.

Commander C. D. Wheelock

The proposal to test the flexible roller and flexible roller path design on the 6" three-gun turret is a good plan.

This should give a satisfactory test of both the mechanical characteristics and the manufacturability of the designs.

The 6" and the 16" turrets are very similar. If the manufacturing and mechanical bugs are worked out of the flexible path design for the 6" turret it will then be a very simple matter to scale up the design for a 16" turret.

The flexible design with the use of the highest possible strength materials for both the rollers and the roller path rings offer great advantage in the saving of weight, in the saving of space, and the possible saving of manufacturing time and cost. I am very much interested in finding if the design can be made to work.

Captain E. Q. Cochrane requested an account of statement made by representatives of the Timken Roller Bearing Company of Canton, Ohio concerning cooperation between the Navy Department and the Company in the development of new designs for ordnance training mounts.

H. B. Maris: Naval Research Laboratory

On March 3, 1941 I visited the Timken Plant in Canton, Ohio to discuss the possibilities of flexible rollers and flexible roller paths with T. V. Buckwalter, Vice President of the firm. During the day Mr. Buckwalter took me to the office of the President, W. E. Umstattd, for a brief discussion of the problem of ordnance bearings.

President Umstattd's discussion may be summarized as follows:

High strength, high brinell materials are as important in military bearings as in industrial bearings. It would probably be advantageous to use those high brinell materials in all ordnance bearings if designs were adapted to their use.

In the past there has not been close cooperation between Army, Navy, and industrial designers. Closer cooperation could very greatly reduce the cost, the time, and the headaches in the manufacturing ordnance bearings. Lighter, stronger, more compact bearings with longer life could be obtained with less trouble if there were better coordination of the efforts of the different designers.

The Timken Company was ready to do anything possible to support better cooperation in the design of bearings.

The flanged roller was not a good design.

Vice President Buckwalter's discussion may be summarized as follows:

Higher brinell materials should be used in turret bearings.

Designs could be simplified. If there was anything the design section or the Research Laboratory of the Company could do to demonstrate these needs they would be glad to do it. If the Navy would outline some plan for testing the flexible bearing and flexible roller path the company would assist with the plan in any way it could.

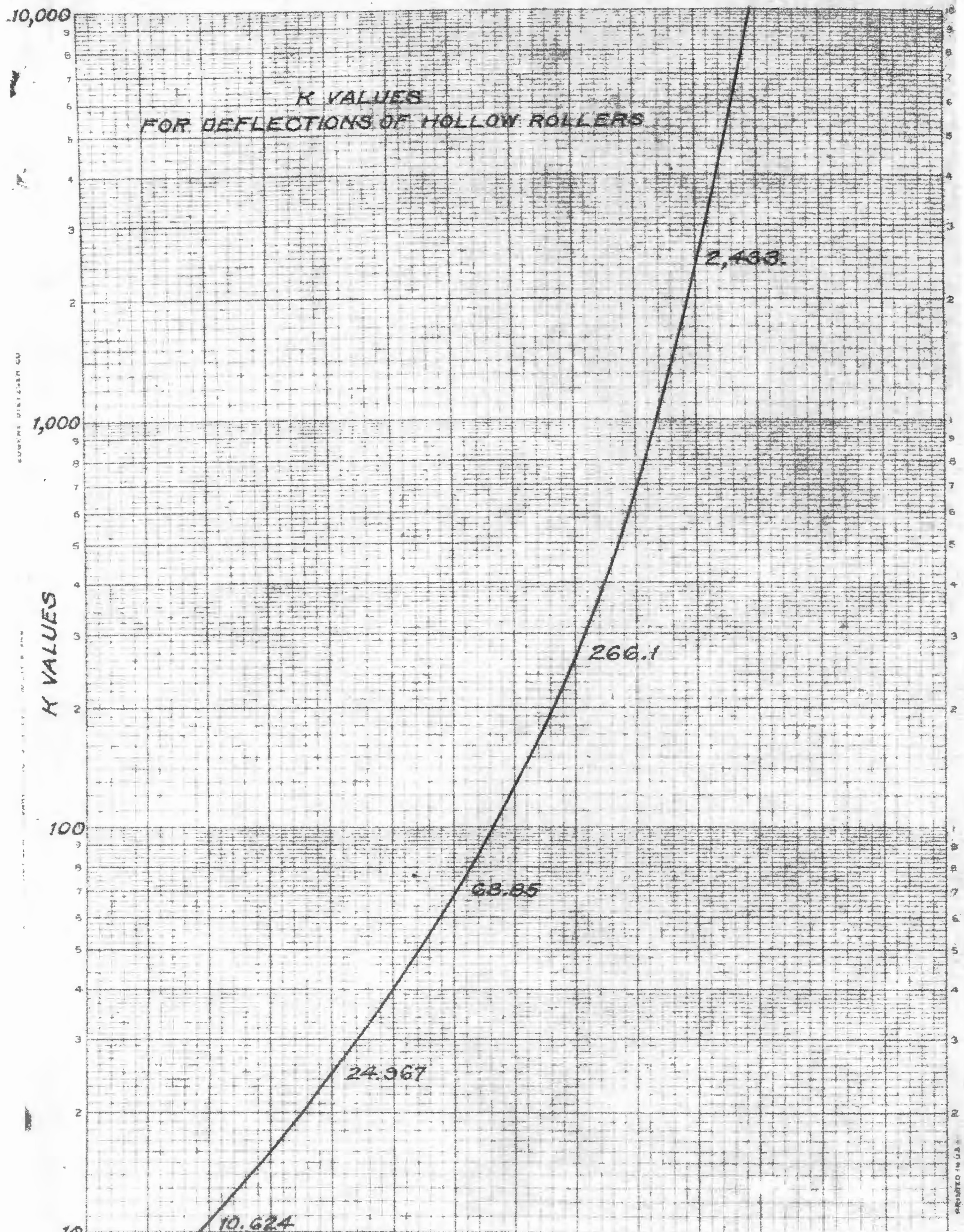
On April 5, 1941 I spent most of the day with Mr. Buckwalter studying the problem of designing and testing a new type of turret bearing. The Taylor Model Basin was visited and their equipment for testing turret bearings was inspected.

Commander W. P. Roop raised the question of design of a flexible roller, flexible roller path model turret for test with the Carderock test stand. After some discussion the conclusion was reached that the design should be drawn for a full scale 16" three-gun turret. Mr. Buckwalter agreed to take on the job. Commander Roop spoke of a $20.00 a day limit for contract design work. Mr. Beckwalter said there would be no charge. He explained that he wished to maintain his identification with the Timken Company and that such activity was in line with the established policy of the Company, offering as an example the pioneering of roller bearings for railroads, where the company built a complete locomotive and loaned it to the different railroad companies for service tests.

As a final step Mr. Buckwalter agreed to come to Washington at his own expense whenever it would be possible to arrange a conference for deciding on details of a cooperative program of design for either models or full scale turret roller paths.

At the Naval Research Laboratory Mr. Buckwalter, after a study of Sheet 1 of Plan F-411 said the dimensions of design a were as large as was needed. He asked that a copy of the letter accompanying F-411 be sent him for preliminary study before the conference to arrange details of the cooperative design program.

Mr. Buckwalter is expecting a letter from the proper authorities in the Navy Department asking him to come to Washington for a conference.



K VALUES
FOR DEFLECTIONS OF HOLLOW ROLLERS
K VALUES
10,000
1,000
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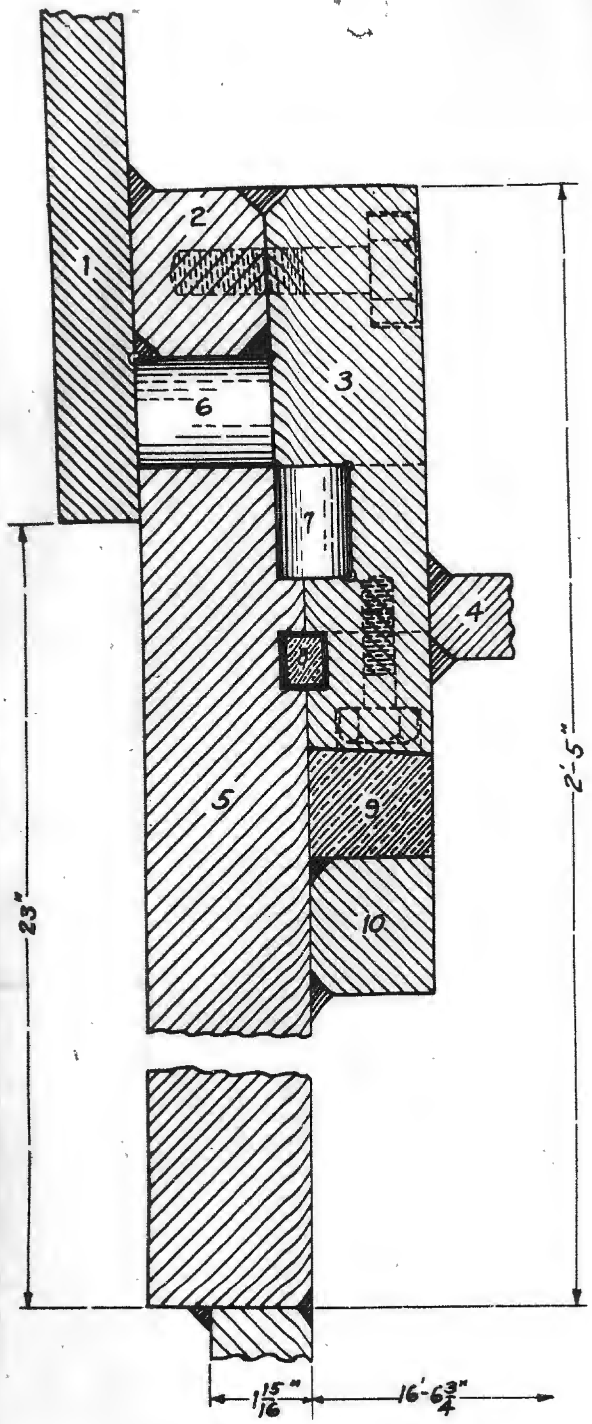
2,433.
266.1
68.85
24.967
10.624



SCALE- 6"=1 FT.

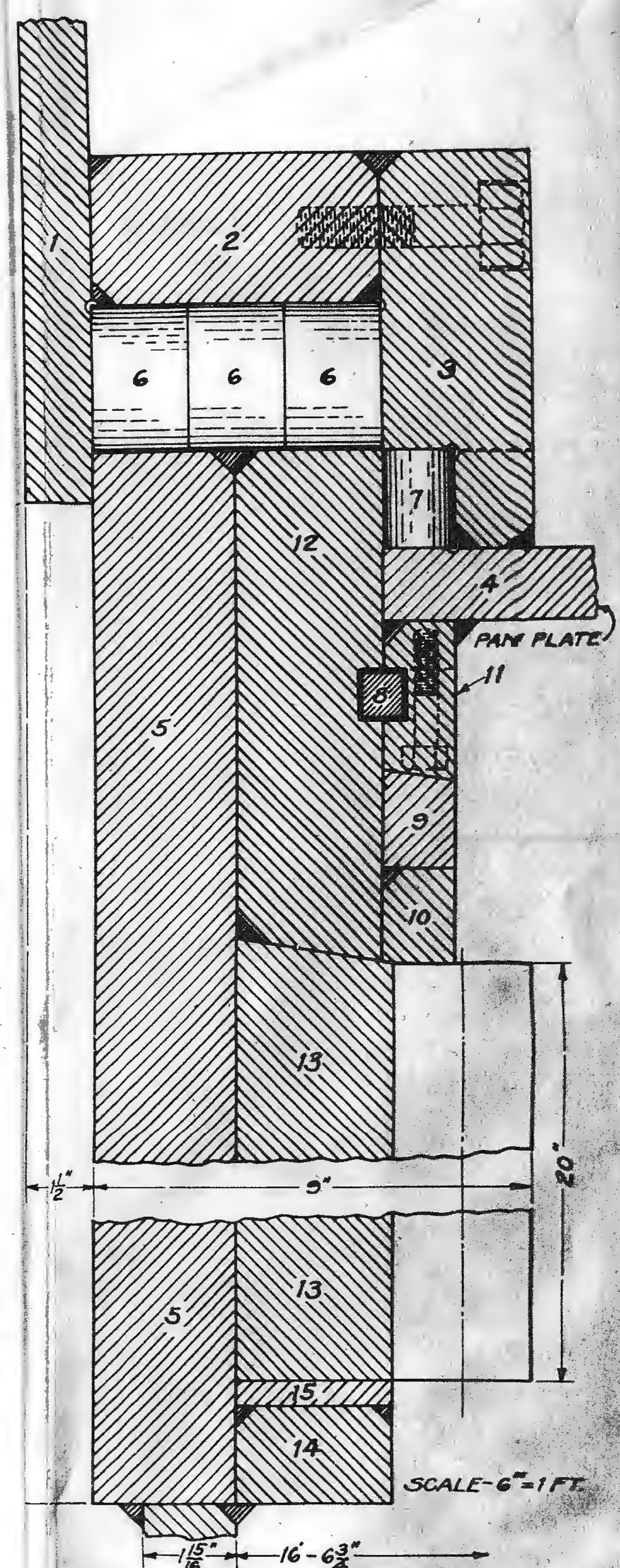
FLEXIBLE ROLLS AND ROLLER PATH FOR 3 GUN TURRET

-HORIZONTAL PATH-

CIRCUMFERENCE	1,250."
NUMBER OF ROLLS	620
DIAMETER	2"
LENGTH	2.5"
WALL THICKNESS	.20"
STRENGTH (PRECISION)	20,000.#
LOAD (NORMAL)	4,800.
FLATTENING	.011"

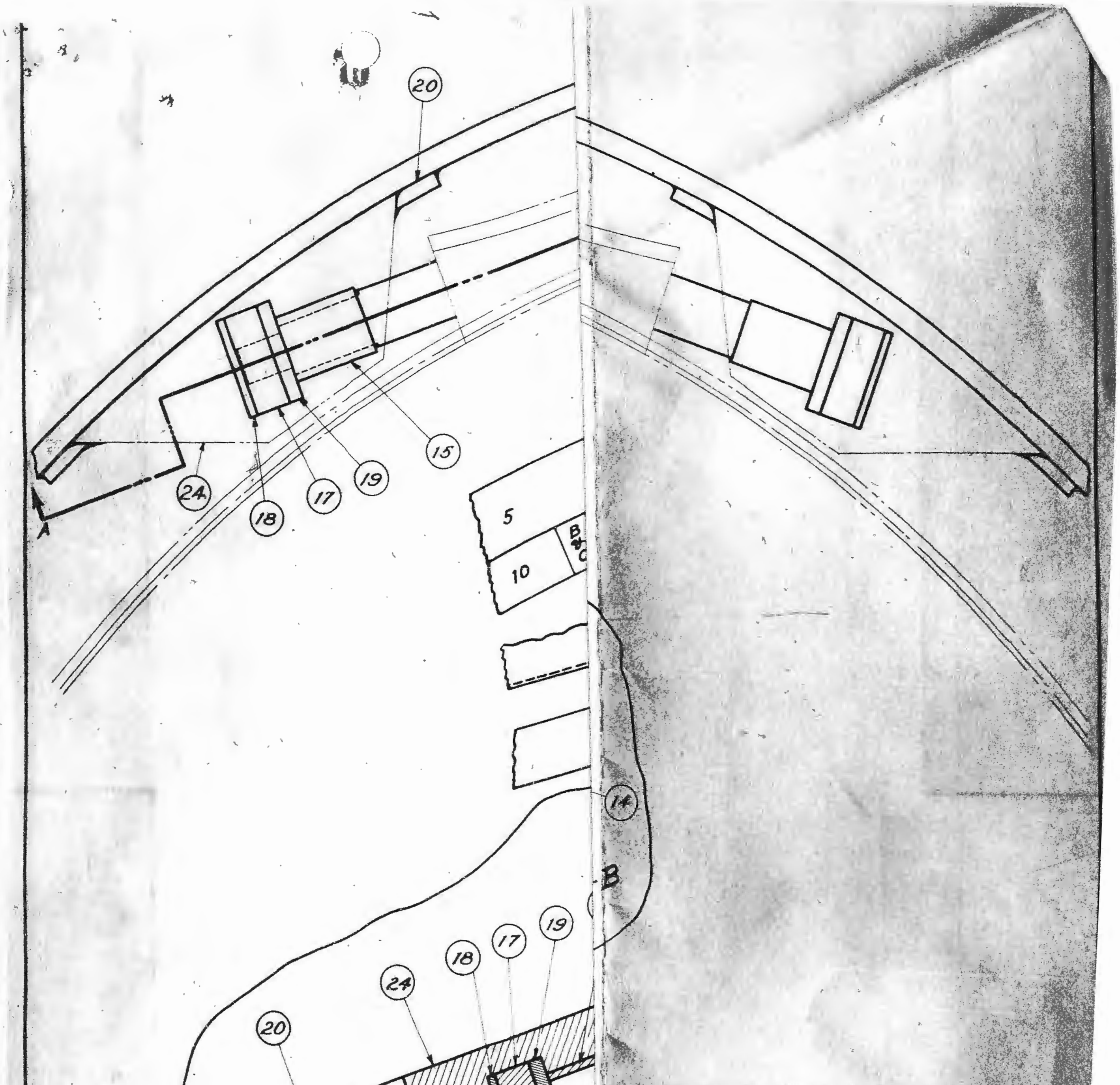
-VERTICAL PATH-

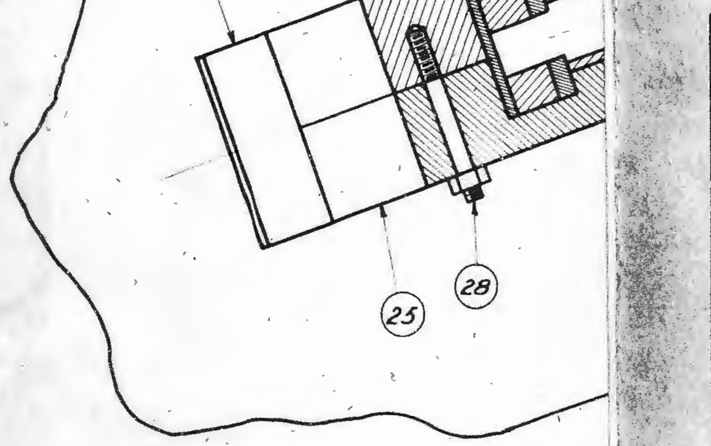
CIRCUMFERENCE	1,238."
NUMBER OF ROLLS	825.
DIAMETER	1.5"
LENGTH	2.0"
WALL THICKNESS	.15"
STRENGTH (PRECISION)	12,000.#
FOR .05 INCREASE IN PATH DIAMETER	
LOAD ON ROLLS	1,600#
FLATTENING OF ROLLS	.005"



SCALE-6"=1 FT.

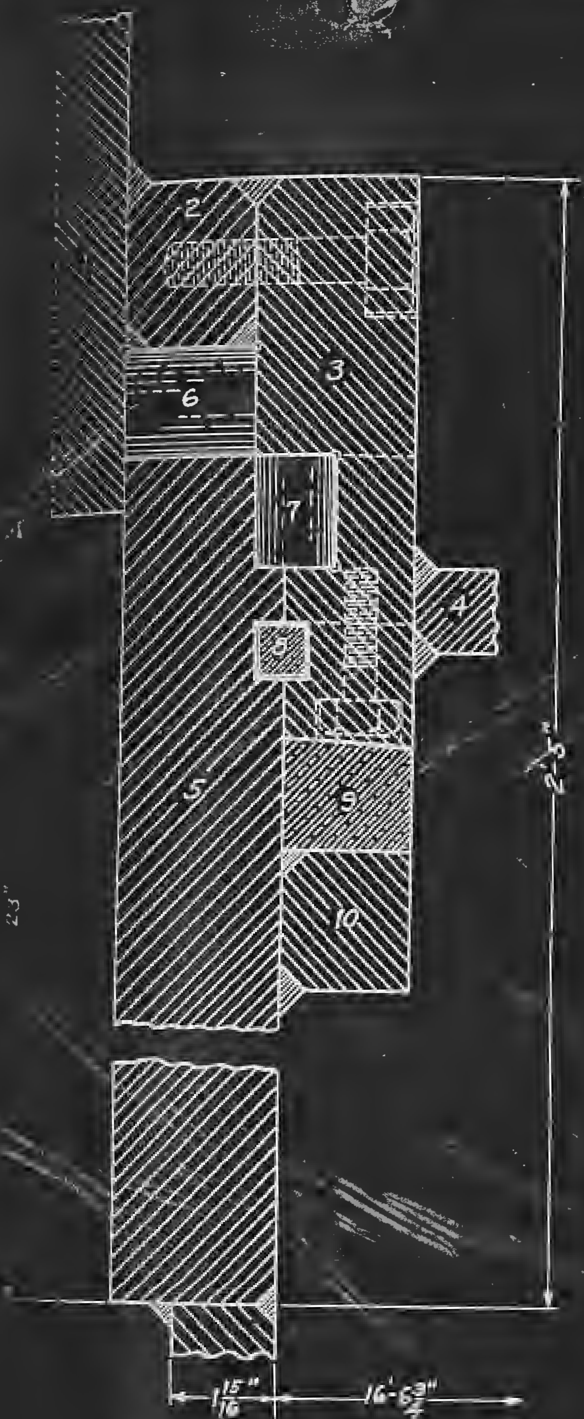
DRAWN	U. S. NAVAL RESEARCH LABORATORY BELLEVUE, D. C.
E.L.H	
CHECKED	TRAINING DRIVE FOR 16" TURRET
APPROVED	SCALE 3" AND 6"=1 FT. - DATE 4-7-1941
	REFERENCE / NUMBER F411



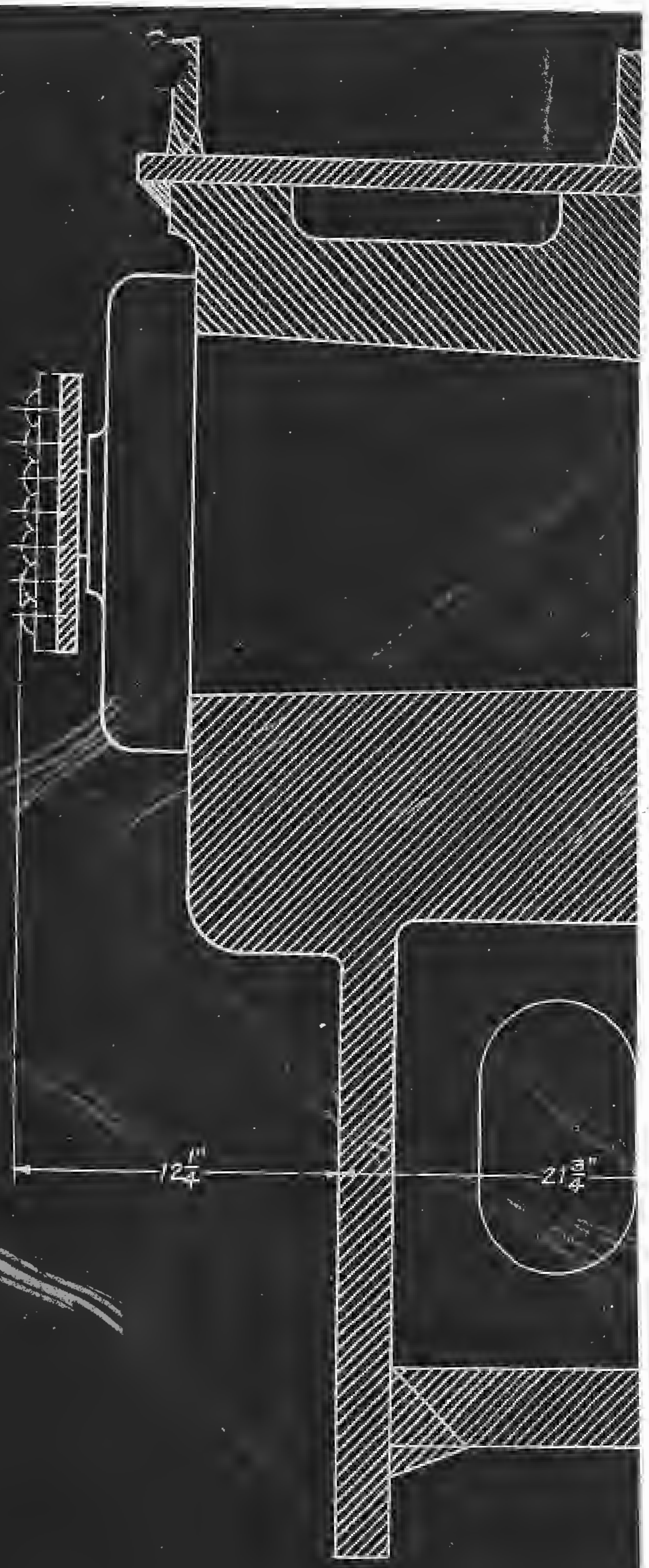


13	RACK
14	WORM GEAR
15	RADIAL BEARING
16	GEAR
17	THRUST BEARING
18	THRUST PLATE
19	WASHER
20	STRIPS
21	GEAR
22	SHAFT
23	GEAR
24	END CASTING
25	" " (LOWER)
26	CENTER CASTING
27	" " (LOWER
28	STUD BOLT
29	BRACE RING

DRAWN J.B.W.	U. S. NAVAL RESEARCH LABORATORY BELLEVUE, D. C.
CHECKED	TRAINING DRIVE FOR 16" TURRET
	SCALE 1" = 1'-0" DATE FEB. 18,
APPROVED REFERENCE	NUMBER 5411



SCALE- 6" = 1 FT.



SCALE-3"

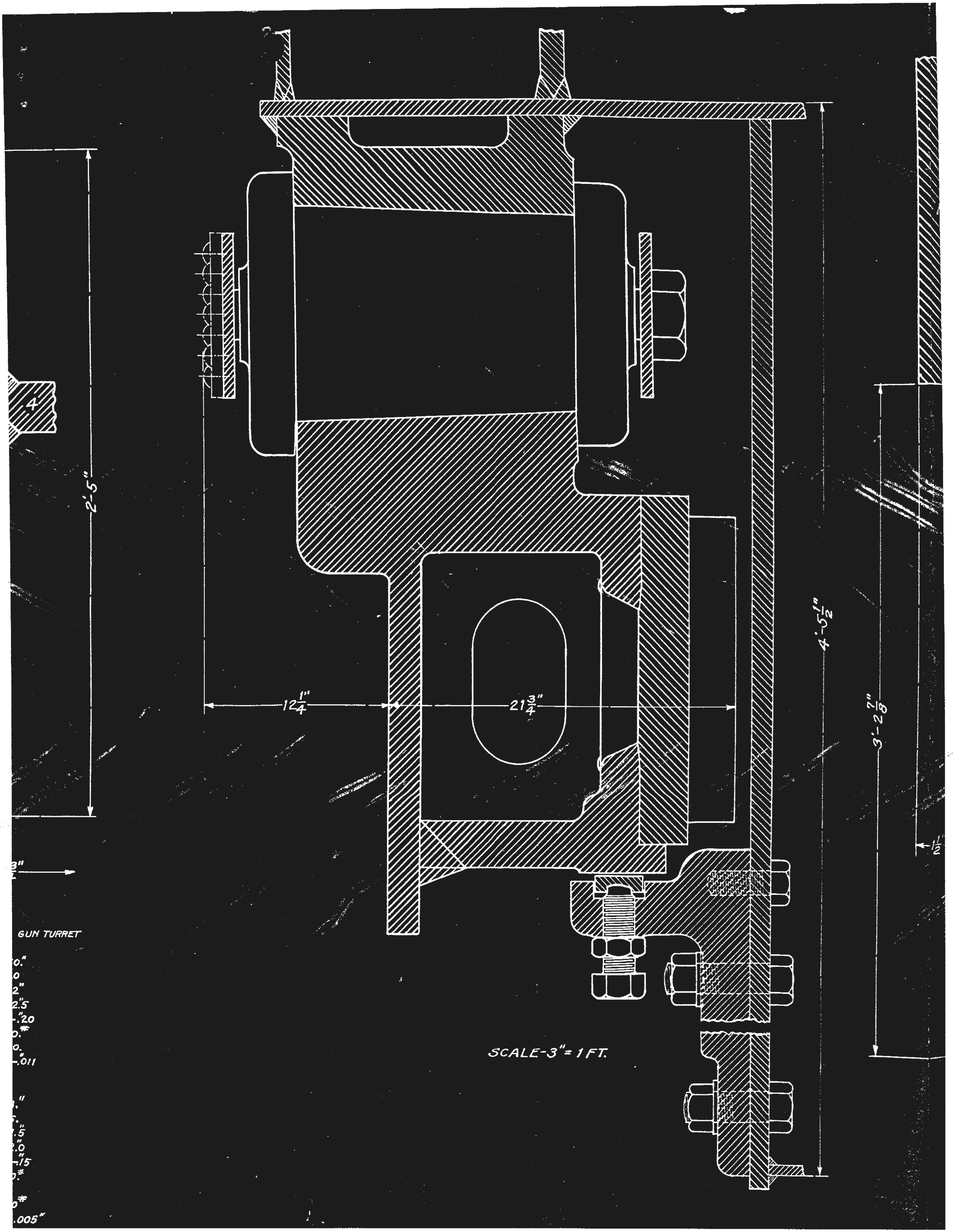
FLEXIBLE ROLLS AND ROLLER PATH FOR 3 GUN TURRET

-HORIZONTAL PATH-

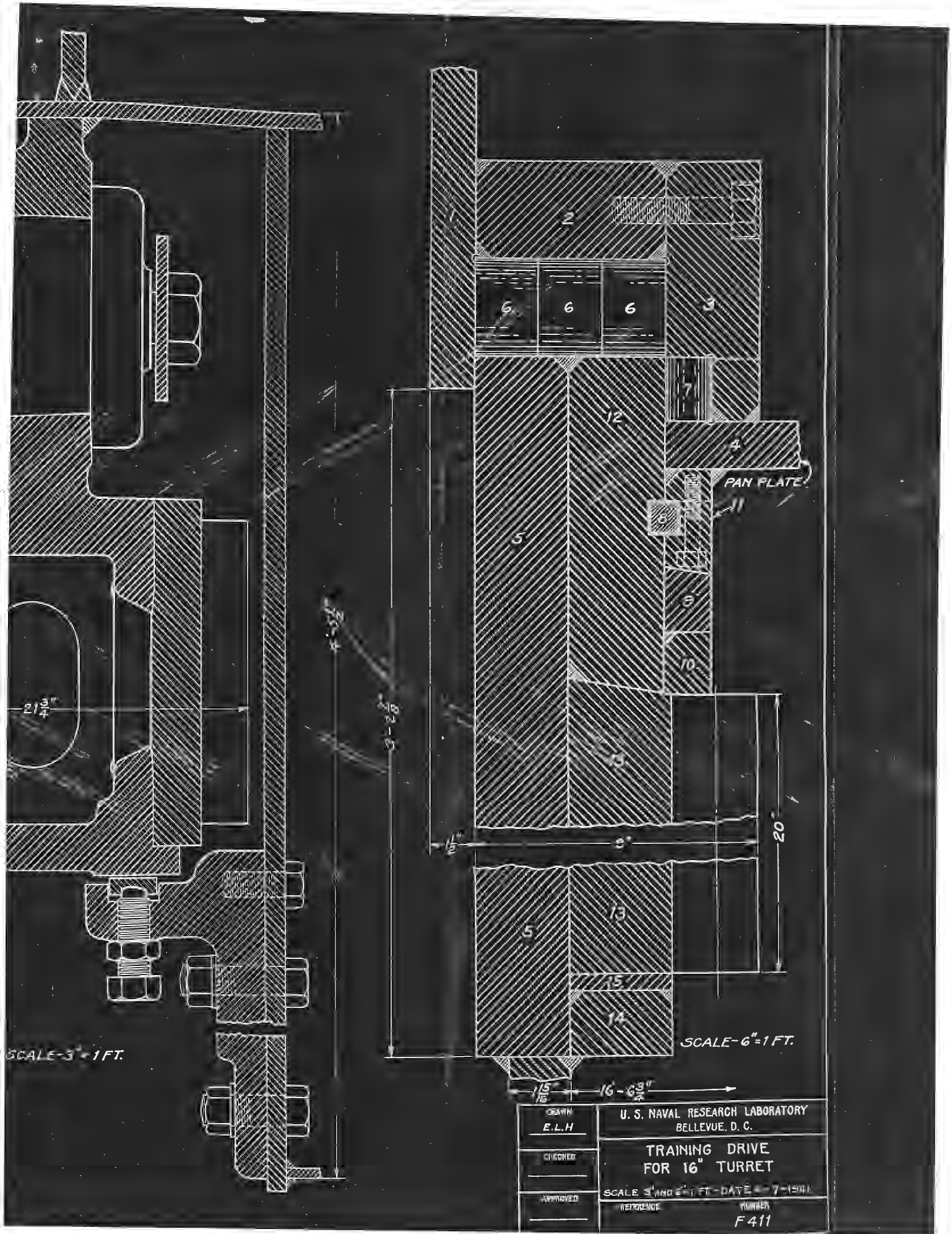
CIRCUMFERENCE	1,250."
NUMBER OF ROLLS	620
DIAMETER	2"
LENGTH	2."5
WALL THICKNESS	".20
STRENGTH (PRECISION)	20,000.#
LOAD (NORMAL)	4,800.
FLATTENING	".011

-VERTICAL PATH-

CIRCUMFERENCE	1,238."
NUMBER OF ROLLS	825.
DIAMETER	1.5
LENGTH	2.0
WALL THICKNESS	.15
STRENGTH (PRECISION)	12,000#
FOR .05 INCREASE IN PATH DIAMETER	
LOAD ON ROLLS	1,800#
FLATTENING OF ROLLS	.005"



2'-5"
12 1/4"
21 3/4"
4'-5 1/2"
3'-2 7/8"
1/2"
GUN TURRET
SCALE-3" = 1 FT.


1
2
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12
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PAN PLATE
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5
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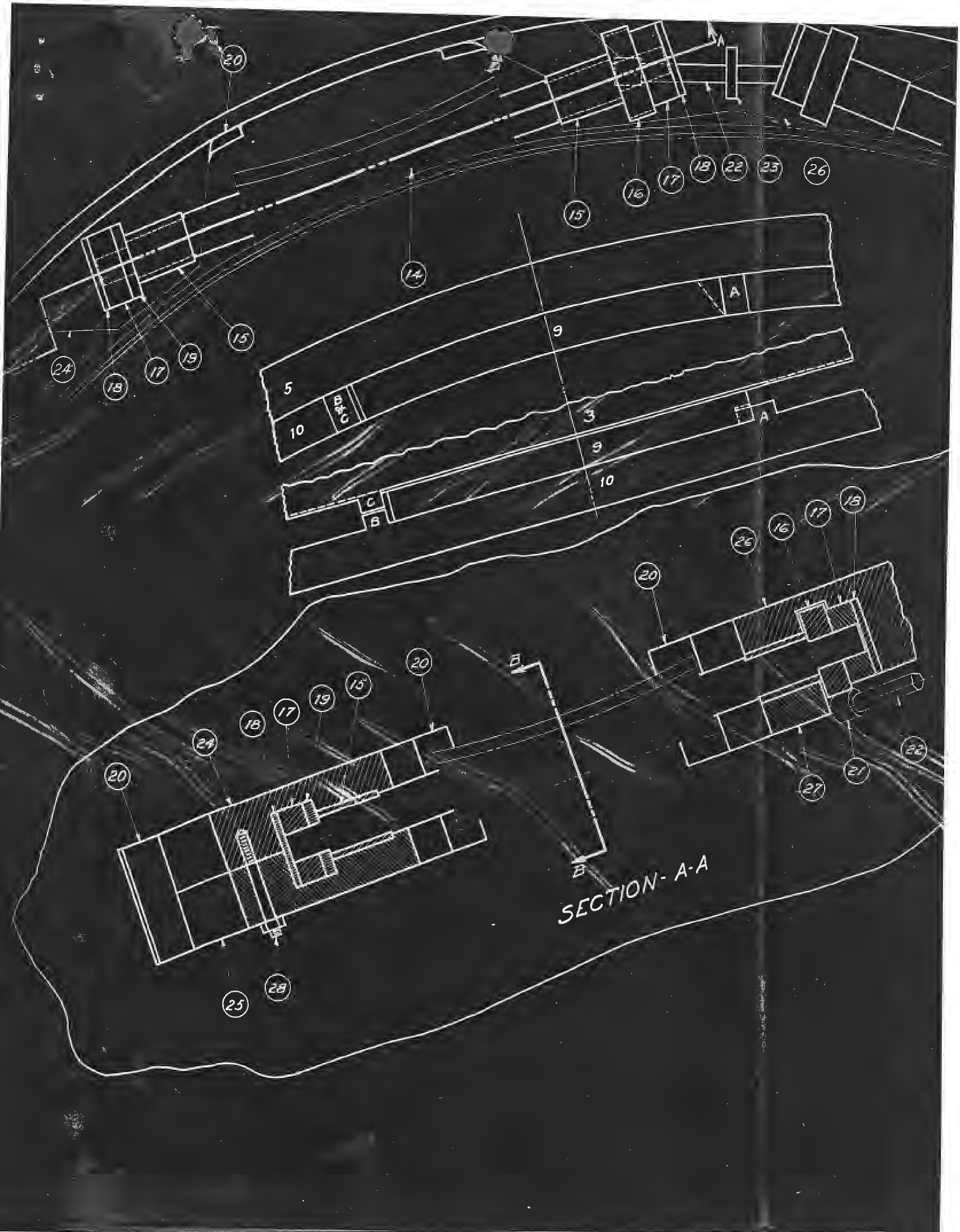
15
14
$21\frac{3}{4}"$
$1\frac{1}{2}"$
8"
20"
SCALE-3"=1 FT.
SCALE-6"=1 FT.
$1\frac{15}{16}$
$16'-6\frac{3}{4}"$
DRAWN
E.L.H
CHECKED
APPROVED
U. S. NAVAL RESEARCH LABORATORY
BELLEVUE, D. C.
TRAINING DRIVE
FOR 16" TURRET
SCALE
REFERENCE
NUMBER
F 411



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24
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A
9
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B & C
10
3
C
B
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B
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28
27
21
SECTION- A-A

13	RACK
14	WORM GEAR
15	RADIAL BEARING
16	GEAR
17	THRUST BEARING
18	THRUST PLATE
19	WASHER
20	STRIPS
21	GEAR
22	SHAFT
23	GEAR
24	END CASTING
25	" " (LOWER)
26	CENTER CASTING
27	" " (LOWER
28	STUD BOLT
29	BRACE RING

DRAWN J.B.W.	U. S. NAVAL RESEARCH LABORATORY BELLEVUE, D. C.
CHECKED	TRAINING DRIVE FOR 16" TURRET
	SCALE - 1" = 1'-0" DATE - FEB. 13, 1941
APPROVED	REFERENCE / NUMBER F 411

13	RACK
14	WORM GEAR
15	RADIAL BEARING
16	GEAR
17	THRUST BEARING
18	THRUST PLATE
19	WASHER
20	STRIPS
21	GEAR
22	SHAFT
23	GEAR
24	END CASTING
25	" (LOWER)
26	CENTER CASTING
27	" (LOWER
28	STUD BOLT
29	[illegible]

DRAWN [illegible]	U S NAVAL RESEARCH LABORATORY [illegible] D C
CHECKED	TRAINING DRIVE FOR [illegible]
[illegible]	[illegible]

SCALE- 6" = 1 FT.

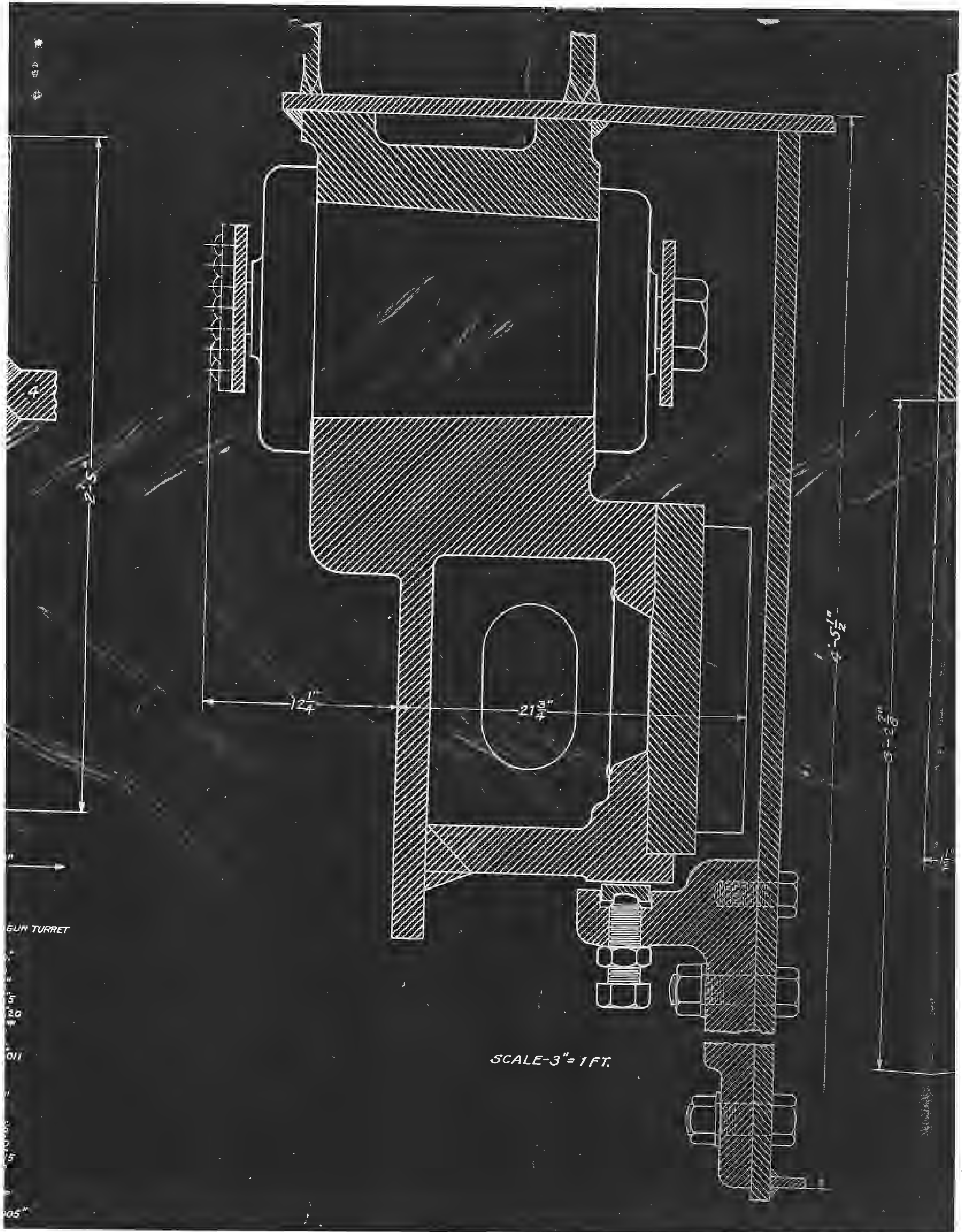
FLEXIBLE ROLLS AND ROLLER PATH FOR 3 GUN TURRET

-HORIZONTAL PATH-

CIRCUMFERENCE ------------- 1,250."
NUMBER OF ROLLS ------------- 620
DIAMETER ------------------- 2"
LENGTH --------------------- 2.5"
WALL THICKNESS --------------- .20"
STRENGTH (PRECISION) ---------- 20,000.#
LOAD (NORMAL) --------------- 4,800.
FLATTENING ------------------ .011"

-VERTICAL PATH-

CIRCUMFERENCE ------------- 1,238."
NUMBER OF ROLLS ------------ 825.
DIAMETER ------------------ 1.5"
LENGTH -------------------- 2.0"
WALL THICKNESS -------------- .15"
STRENGTH (PRECISION) ---------- 12,000.#
FOR .05 INCREASE IN PATH DIAMETER
... ON ROLLS --------------- 1,600#
...NG OF ROLLS ------------- .005"


2'-5"
12 1/4"
21 3/4"
4'-5 1/2"
GUN TURRET
SCALE-3"=1 FT.