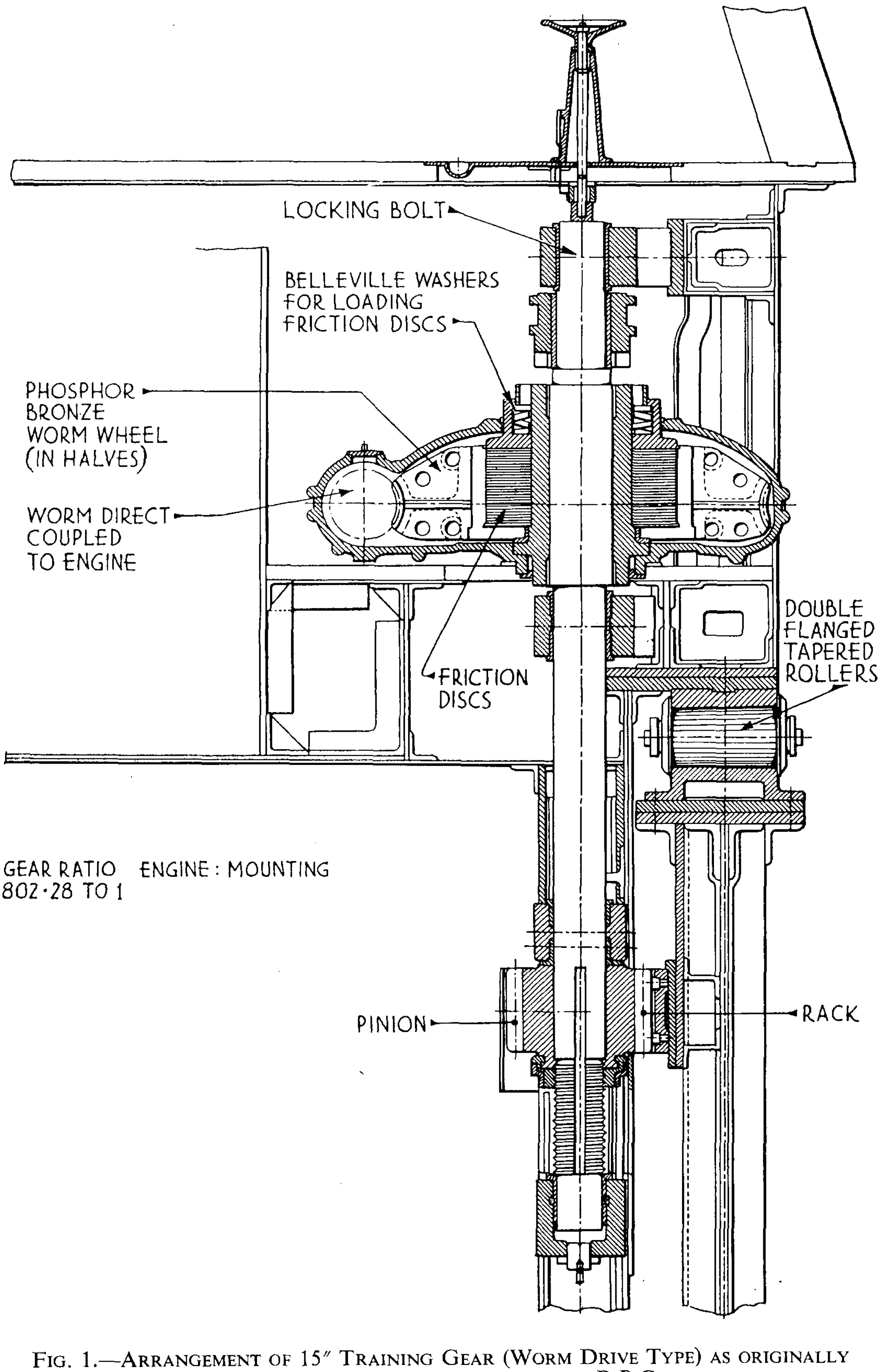

FIG. 1.—ARRANGEMENT OF 15″ TRAINING GEAR (WORM DRIVE TYPE) AS ORIGINALLY FITTED BEFORE THE INTRODUCTION OF R.P.C.

PART I

GUN TRAINING DRIVES

SOME FACTORS INFLUENCING DESIGN

by

COMMANDER (E) J. O. H. GAIRDNER, O.B.E., R.N.
Naval Ordnance Department

When a gun is fired the energy which is released is partly converted into heat in the recoil cylinder and partly dissipated in accelerating the turret to the rear and deflecting the resilient gunmounting structure.

If a wing gun of a turret is fired the energy released gives rise to a torque about the training axis which may be sufficient to override the training engine unless special precautions are taken to oppose this tendency. This torque follows inevitably from the law of conservation of energy and vitally affects and determines the design of turret training gears. It is the purpose of this article to examine some of the features of gunmounting training designs in the light of the exacting engineering and gunnery requirements and of the inflexible laws of nature.

" Throw-off "

In hand-operated gunmountings a non-reversing drive was always fitted between the hand wheels and the gun. This was necessary to prevent the wheels from being back-driven by the gun on recoil. To limit the stresses, surplus energy of recoil was arranged to be dissipated, if necessary, through a suitable safety valve in the form of a slipping clutch between the gun and the non-reversing feature. Non-reversing worms, however, were retained when power-operation was introduced since auxiliary hand drives were in these cases always provided.

In those days the amount the gun overhauled or " threw off " was of little importance as the rates of fire were low and allowed sufficient time for the gun to be brought back on to the target. In more modern times this is no longer true. Two or more semi-automatic guns may be mounted in a turret and fired rapidly and at random, in which case the displacement of the turntable by one gun will affect the bearing at which its neighbour is fired. The same is equally true of the elevating motion of multiple guns fitted in a common cradle.

" Throw off " may therefore become a major error in gunpointing if the reversibility of the drive or its ability to " render " is not carefully controlled.

Irreversible Drives

The simplest form of irreversible drive for gunmountings is the low lead-angle or " inefficient " worm. A worm drive is, moreover, a convenient feature in gunmounting drives since a high gear ratio between engine (or hand) and gun is normally required. Unfortunately, for a drive to be irreversible at a given speed its forward efficiency at that speed must be 50% or less. This is a heavy price to pay, especially in hand-worked mountings.

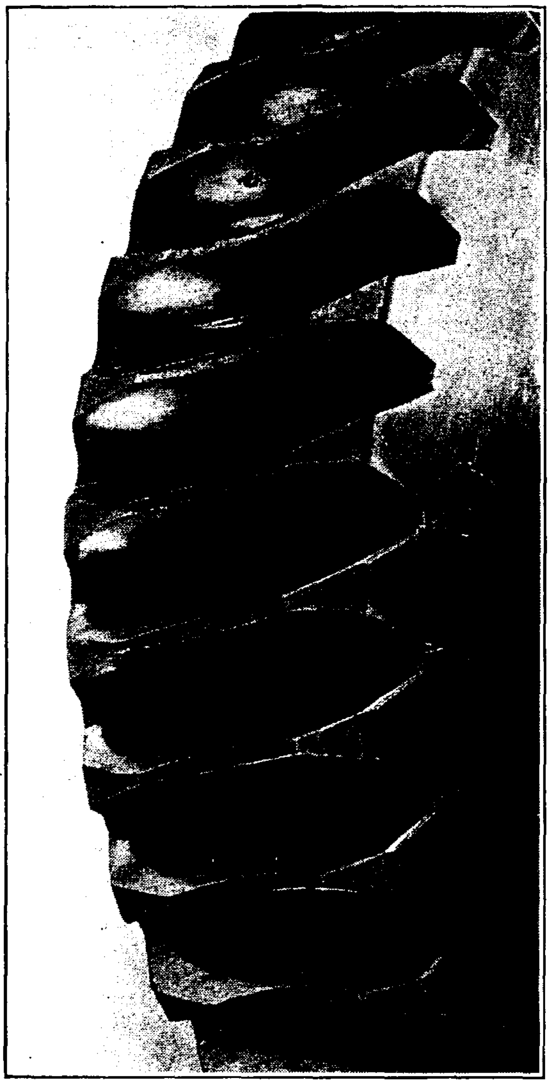

FIG. 2.—SEVERELY WORN WORM-WHEEL

Low efficiency worms may, however, lead to heavy stresses in worm wheels, pinions, and racks. Consider an engine capable of exerting a maximum torque (limited by relief valves) of T lb.ft. If the forward efficiency of the worm gear is E_f, the maximum torque obtainable at the rack is $T.E_f.R.$ lb.ft. where R is the gear ratio. With over-riding conditions, the torque acting upon the worm wheel shaft and sufficient to lift the relief valves at the engine is $\frac{T.R.}{E_r}$ where E_r is the reverse efficiency.

Thus the ratio of the maximum torques borne by the worm wheel shaft when over-riding to that to which it can be exposed when being driven by the engine is $\frac{1}{E_f \times E_r}$. As the reverse efficency of a drive having $E_f = .6$ may be only .3, in such a case the pinion and rack would need over five (5.2) times the strength necessary to carry the driving torque. Reverse efficiencies increase rapidly, however, with increasing values of E_f and a drive of 80% forward efficiency may be expected to have E_r 75%, whence $\frac{1}{E_f \times E_r}$ is only 1.7.

Some Disadvantages of Worm Drives

Worm drives have the great disadvantage of poor starting efficiency due to the lack of a fluid film between worm and wheel at low rubbing speeds and to the high proportion of sliding friction present in this form of drive. Apart from the excess of power thus required at creep speeds the degree of reversibility of a worm will increase with speed. Hence, to ensure that a worm drive is irreversible at high speeds, the efficiency at creep speeds may have to be undesirably low.

There is little that can be done about this since the evils are inherent in the kinematics of worm gears though various novel worm profiles and designs have been developed from time to time to reduce the rubbing friction or to lower the speed at which fluid film lubrication is generated. Poor starting efficiency can be somewhat improved by the selection of materials and lubricants but little significant advance can be expected so long as the gear is subject at low speeds to the laws of mixed friction.

Despite these objections the low-efficiency worm-gear serves its main purpose for both hand and power-operated mountings, but, applied to servo-controlled (R.P.C.) mountings, a new disadvantage comes to light.

The smooth and accurate deceleration control of an inertia load through a low-efficiency drive is a difficult proposition, as may be imagined were it necessary to control a motor car with such a back-axle drive. The difficulty is greatly increased if the torques on the worm are subject to rapid fluctuation.

These conditions, do, in fact, prevail in R.P.C. mountings in which heavy retarding torques may be developed with great rapidity and frequency. To aggravate matters the stresses set up under the over-riding conditions, to which reference has already been made, cannot now be limited by slipping clutches, for positional control of the gun is customarily effected by controlling the position of the driving motor, and a solid connection between the motor and load is thus essential.

The conditions necessary for controlled deceleration may be seen by examination of the expression :*

$$\text{(i)}\quad D = T\left\{\frac{n}{I_w n^2 - I_1\left\{\dfrac{\tan(\phi - \lambda)}{\tan\lambda}\right\}}\right\}$$

where D = deceleration of the load

T = reverse torque applied by engine

n = gear ratio of the worm pair

I_1 = Moment of Inertia of everything on the wheel side of the worm gear

I_w = Moment of Inertia of everything on the worm side of the worm gear

ϕ = Friction angle ($\tan^{-1}$) at the worm contact

λ = lead angle of worm

If the R.H.S. of this expression is to be positive then

$$\text{(ii)}\quad \frac{I_w n^2}{I_1} > \frac{\tan(\phi - \lambda)}{\tan\lambda}$$

which is the essential requirement if D is to be a function of T, *i.e.*, if we are to have controlled deceleration.

If the difference between the two sides of the expression (ii) is small the rate at which deceleration increases with increments in torque T is large, until, in the limit, when the two sides are equal the smallest increment in retarding torque T will give an infinite value of D, *i.e.*, the gear will lock.

In gunmounting drives fitted with low lead-angle worms the rate of increase of deceleration with increments in retarding torque is high and if servo-control is to be fitted it becomes necessary to limit the rate at which the retarding torque can vary. This can be achieved by a process of "watering down" the servo performance, which carries with it certain disadvantages, but if remedial action of some sort is not taken sporadic angular departures between the initiating transmission and the controlled load, as are inevitable in R.P.C., give rise to engine torque fluctuations which cause quite disproportionate decelerations and "bumpy" running results.

If, then, the left-hand side of expression (ii) is to be markedly greater than

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* The writer is grateful to Mr. J. M. Ford of the Admiralty Gunnery Establishment, Teddington, for this presentation of the problem.

FIG. 3.—HEAVILY SCORED WORM-WHEEL

the right-hand side the only alternatives that can be applied are :—

(*a*) Increase I_w by the addition of a flywheel

(*b*) Increase n

(*c*) Increase λ

(*d*) Reduce ϕ

Clearly (*a*) and (*b*) will have serious repercussions on the engine size and very little significant reduction in ϕ is possible without departure from conventional worm technique, so that the increases in λ remains as the only practicable method of achieving the results desired. In plain terms the worm must be made more efficient.

Change to All-spur Drives

Heavy scoring of low-lead-angle worm-wheels (see Fig. 3) and some costly failures of pinion shafts amply confirmed the logic of this reasoning but the importance attached to " throw-off " gave to low-efficiency drives in R.P.C. turrets a longer life than they otherwise deserved, and for many years a compromise was effected by the use of worms of about 20° angle having an efficiency at creep speeds of about 68%. In more recent times this compromise has been abandoned and all-spur drives have been introduced. Examples are to be found in *Vanguard's* 15 in. and 5.25 in. mountings. (Fig. 4).

This departure has far-reaching implications. The drive, being now reversible, the engine under " throw off " conditions acts as a pump, and stresses must be limited by relief valves in the pressure pipes. The " torque-ability " of the engine, as governed by its size and permissible peak pressure, now governs the throw-off angle that occurs.

The amount and control of throw-off is influenced by whether the power drive is electric or hydraulic. The hydraulic drive has the advantage that, given no internal leakage, it will develop a large opposing torque for a very small angular displacement of the shaft independent of the correcting influence of the servo. The electric drive, on the other hand, generates no such torque of itself though its high rotational inertia (frequently as great as that of the mounting itself when the square of the gear ratio is introduced) helps to limit the throw-off angle.

To sum up, therefore, the modern fashion is to use all-spur drives and to limit the throw-off by relief valves. Further reductions are then only obtainable by raising the hydraulic and mechanical stiffness of the transmission.

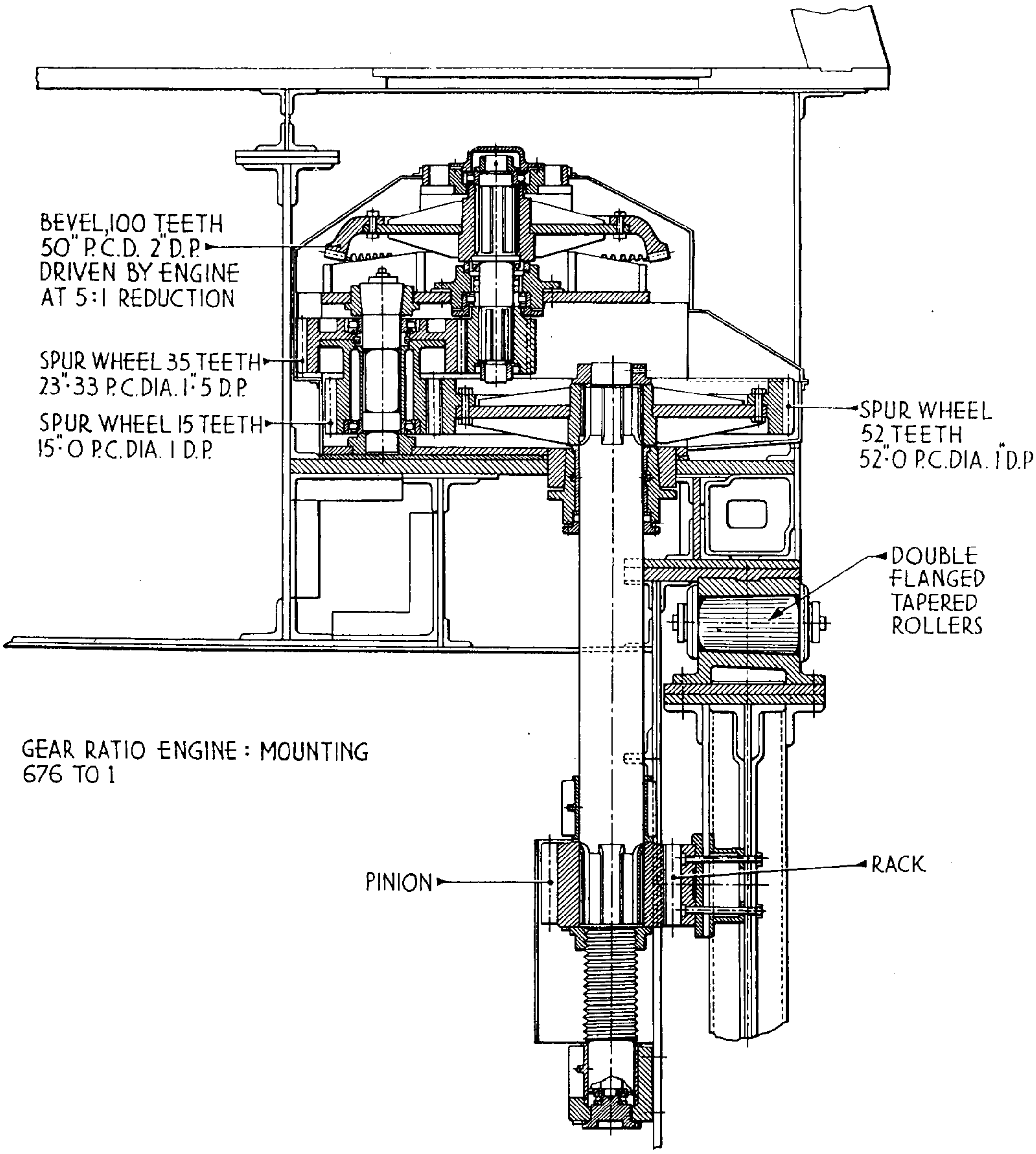

FIG. 4.—ARRANGEMENT OF 15″ TRAINING GEAR (ALL SPUR TYPE) AS FITTED IN H.M.S. "VANGUARD"

Mechanical and Hydraulic Resilience. The Servo problem

Whilst the actual throw-off angle of a gun may, to a small extent, be due to the twisting of the structure or the compression of oil in the hydraulic system, these two factors have a more important bearing upon the behaviour of the gun-driving servo, or R.P.C. system. Without introducing superfluous technicalities, it is true to say that the more highly a servo is tuned (*i.e.*, the greater the output torque developed by a unit "error" signal) the more is it influenced by disturbing phenomena of which resilience and backlash are the two more commonly encountered. In the face of these a servo is liable to hunt, unless the sensitivity is reduced below a certain critical figure. The price of any such reduction is a loss of accuracy.

The omission of the irreversible drive raises the resilience of the transmission in two ways, firstly since the abutment of the drive, hitherto at the worm wheel, is now carried back to the engine ; and secondly because the motor is then subject to an angular wrap-up prescribed by the compressibility of the oil in the system.

Oil, particularly if slightly aerated, is not incompressible and, with pressures up to 2,000 lb/sq. in. or more and pipe system often of necessity of some length, the influence of hydraulic resilience upon the performance of the servo may be considerable.

In brief, the transmission resiliences, be they hydraulic or mechanical, behave as spring couplings whose natural frequencies are readily calculable. To avoid resonance troubles this spring system must have a frequency well above that of the servo itself, as exhibited by the expression

$$F_N = \frac{1}{2\pi}\sqrt{\frac{\text{Output torque for unit misalignment signal}}{\text{Inertia of driven load.}}}$$

Transmission stiffness must therefore be kept as high as possible by the designer. Given a free hand he would site the engine as close to the final rack as possible, build the hydraulic pump and engine into a common casing with no interconnecting pipes and keep oil pressures low. Unfortunately such latitude in design is well nigh impossible to achieve. Hoists, loading gear, etc., all clearly related to the position of the gun, rarely allow the mounting designer to put the elevating and training pumps and engines where he well knows they ought to be. Nor can he tolerate the added bulk of low pressure hydraulic units.

Much can, however, be done by careful design of individual components of the mounting.

The Design of a Stiff Transmission System

The following are some of the lines of profitable approach :—

(i) *Fabrication of structures and gear boxes.* Fabrication offers possibilities of raising the stiffness of a structure without corresponding increase in weight. The fabrication of structures as large as the 4.5 in. Mk. VI turntable (weight 6 tons, diameter 11 feet) is now common practice for this reason.

Structural rigidity must be considered for the turntable as a whole, down to the individual and highly stressed details such as trunnion brackets, training rack supports, gear boxes and support brackets.

Little is known of the behaviour of structures or components under firing conditions and the flow of stress is being studied with the aid of strain gauges from firing trials conducted on the prototype 4.5 in. turret at Eastney.

Static tests on components, or on Xylonite models loaded to represent gun firing loads, are also being run, from which information should be obtained whether these details are of sufficient rigidity or whether weight could be saved.

(ii) *Use of hollow shafts.* Careful choice of shaft dimensions enables the designer, by using a hollow shaft, to achieve increased stiffness without increase in total weight. For example, if a solid shaft of unit length and diameter having unit torsional stiffness be replaced by a hollow shaft one and a half times the original diameter and $\frac{d}{D} = .75$ then a torsional stiffness of 3.57 is achieved without increase in weight.

Many cases arise in which a 50% increase in shaft outside diameter can readily be accommodated.

(iii) *Stiffness of turret supporting structure.* The torque reactions of the training engine and gear boxes have to be taken by the turret support. Torsional resilience of this structure may be regarded as another resilience in series with that of the mounting itself, and its own natural frequency must be as high as possible and at least as high as that of the transmission itself.

(iv) *Reduction of hydraulic resilience.* It has been shown that it is desirable to reduce the volume of oil under compression in the system to a minimum, or to lower its pressure, or both. Lowering the pressure leads to less specific internal leakage or " hydraulic backlash " (an important point in servo systems) but space restrictions clearly limit the extent to which pressures can be reduced in ordinary types of hydraulic machinery. However, if composite pumps and engines have to be ruled out for reasons of bulk, some saving in oil volume can be achieved by a reduction in the bore of the connecting pipes.

Generally speaking, high rates of flow are not objectionable since maximum flow occurs at maximum turret speed when torques are low and the pressure drops in the pipes can be accepted.

While sensational increases in hydraulic resilient frequencies are not to be expected by these means they are always helpful. For example, in a system involving two V.S.G's (A and B ends respectively) joined by 10 feet of 2 in. pipe the natural frequency is 4.3 c.p.s. If the pipe diameters are halved, everything else remaining equal, the figure is raised to 5 c.p.s. The latter figure takes no account of the increased damping due to pipe frictional losses.

The palliatives (i)-(iv) are those that can be applied by the gunmounting designer. One other must be mentioned which lies in the province of the designer of the servo equipment.

(v) " *Divided reset.*" The unstabilising effect of backlash in R.P.C. mountings has been referred to already (see page 205). A technique of " Divided Reset " was devised by Messrs. Metropolitan-Vickers Ltd., to combat backlash, and it is fortunate that it was afterwards found useful also for dealing with resilience.

In a simple positional servo means must be provided for the detection of any departure of the position of the driven load (in this case, the gun) from that of the transmission it is called upon to follow. In practice, small rotational A.C. electrical machines or " magslips " are fitted at transmitter and gun and so connected that an A.C. voltage is generated of amplitude and phase proportional to their relative angular position. It is this " error signal " that is amplified up to a power level sufficient to drive the gun into alignment.

The magslip mounted at the gun is termed the " resetter " and is normally fitted adjacent to the driving motor or engine. This establishes the position of the motor but not of the driven load (the gun) should the motor and load be connected by a shaft having resilience or backlash or both. Now a servo consisting of an engine rigidly coupled to an inertia load can be tuned to a certain maximum sensitivity for a given amount of damping that is a function of the aggregate of the driving and driven inertias. Above this sensitivity figure, hunting will develop.

If, now, backlash be introduced, the driven inertia when the engine is in the backlash zone is momentarily nil. In this zone, therefore, conditions for hunting may be set up to the detriment of the gears and the only immediate cure is to lower the sensitivity and sacrifice performance.

Twin (or " divided ") resetters can, however, be sited on either side of the backlash and so connected that their signals are additive. Now when the engine is in the backlash zone the first resetter only is moved and the output voltage (and hence the engine torque) is less than when the backlash is taken up and both resetters revolve. In practice much of the evil of backlash can be avoided by these means.

Again, with a resilient shaft, torsional oscillations can be set up, the effect of which is picked up by a single resetter unless placed at a node, *i.e.*, that central point on either side of which the system oscillates in opposite phase.

Unfortunately the position normally occupied by a resetter is an anti-nodal point and the combination of load inertia and resilience will ultimately limit the permissible sensitivity of the servo.

Most fortunately it was found that by fitting the " divided resetters " and varying the proportionality of the voltages to that of the driving and driven inertias to which they are coupled, then it is possible largely to synthesize the condition of a shaft of infinite stiffness.

These techniques have been used with great effect on the Mk. VI Director and Mk. VI Bofors mounting. Divided reset does not, however, overcome the bogey of hydraulic resilience which seems unlikely to yield to simple indirect treatment.

The Lay-out of Training Drives

The introduction of R.P.C. has caused many accepted conceptions of design and lay-out to become outmoded. We have seen that the days of long springy drives with worms and slipping clutches are over, and short, stiff, high-efficiency drives and highly stressed hydraulic pipes have taken their place. A number of alternative arrangements of drive are however still available, the choice of which must best suit the often conflicting requirements of servo and mounting designer.

(i) *Should training gear be on or off the mounting?* If pump, engine, and ancillary equipment are off the mounting then space, mounting weight, and inertia are saved and powers can in consequence be somewhat reduced. There will also be less noise in the gunhouse and pipe lengths about the revolving structure. Access to bulky units will be greatly improved.

With this arrangement, however, there will always be a zone of bearings in which the effect of recoil will be to reduce the working clearance between the rack and the pinion. This does not occur in a single pinion " on-mounting " drive but it can be avoided by a system of lateral rollers to limit the turret movement. In a large design the radial accuracy of such an assembly calls for very high class workmanship and allows of little distortion of the roller path support.

Whereas in on-mounting gear it is possible to control the valve or pump swash by direct hand-operated mechanisms, the " local " control of off-mounting equipment must employ electrical links.

(ii) *Should one or more than one pinion be employed?* For several years the normal practice was to use two on-mounting worm gears and pinions, about 180° apart, driven through a common cross shaft. This, in theory, satisfied the requirement for a pure torque, but a serious mistake was made in not separating the worms by a differential. Correct in theory, the torque could never be correctly shared in practice and the firing of a wing gun of a twin mounting, if accompanied by the rendering of the appropriate friction clutch, was liable to leave the cross shaft connecting the worm in a state of constrained

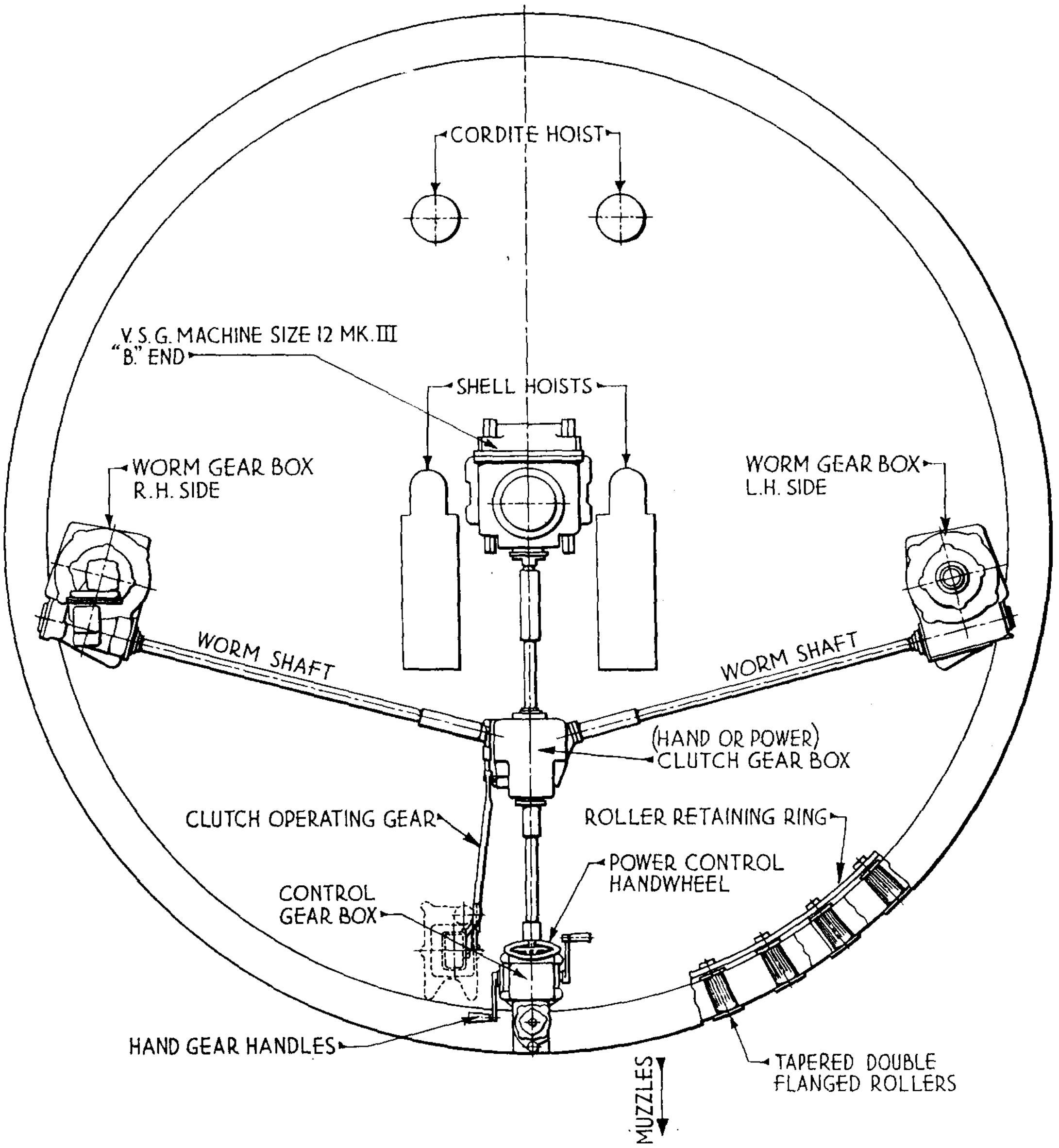

FIG. 5.—LAYOUT OF TRAINING GEAR FOR 5.25″ TURRETS AS DESIGNED BEFORE THE INTRODUCTION OF R.P.C.

NOTE : (i) LENGTH OF RESILIENT SHAFTS BETWEEN ENGINE AND RACK.
(ii) TWIN WORMS WITHOUT DIFFERENTIAL
(iii) TAPERED DOUBLE-FLANGE ROLLERS

twist which forced the worm wheels in opposite directions and jammed the turret (Fig. 5).

One way to overcome this tendency is to employ a cross shaft with a right-handed worm at one end and a left-handed worm at the other. Such an arrangement needs neither differential nor thrust bearings since the shaft will float axially until the opposing torques on the two ends are equal. Alternatively, the shaft may be arranged slightly obliquely across the structure one worm engaging the forward and the other the rear of the associated wheels, the two worms being then to the same " hand " (Fig. 7 and 8). This arrangement was to be found in certain German naval mountings.

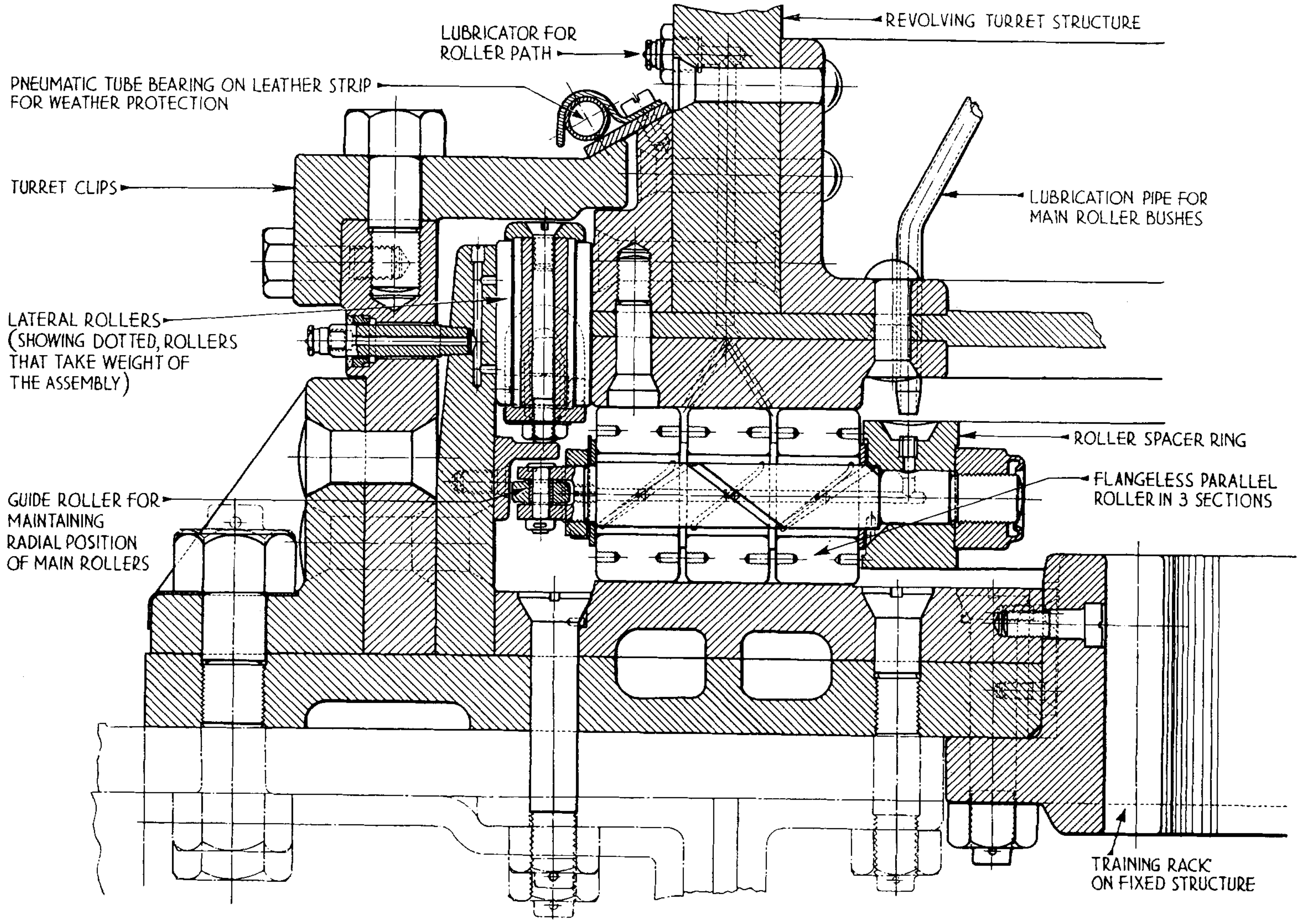

FIG. 6.—5.25″ MARK I*. H.M.S. "VANGUARD." TURRET ROLLER PATH SYSTEM.

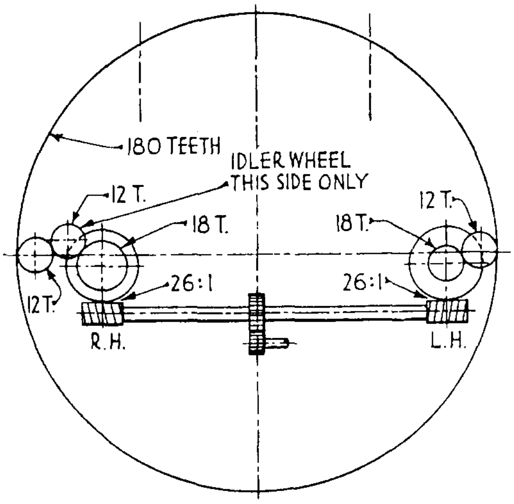

FIG. 7.—ARRANGEMENT OF TRAINING DRIVE EMBODYING OPPOSED PINIONS AND BALANCED CROSS-COUPLED WORM DRIVES, AS FITTED IN CERTAIN GERMAN MOUNTINGS

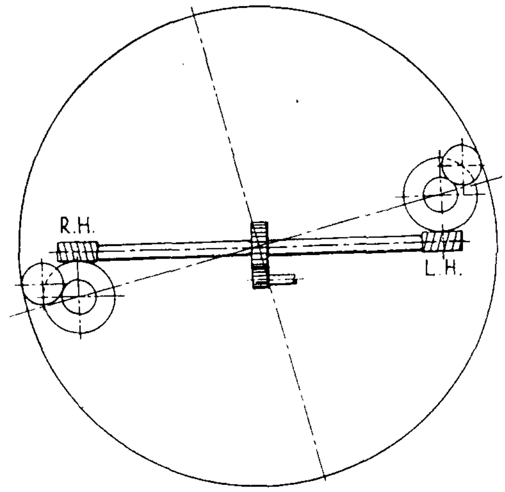

FIG. 8.—ALTERNATIVE VERSION TO FIG. 7, TO AVOID NECESSITY FOR FITTING IDLER WHEEL

The two-pinion drives cannot be employed safely if there is any danger of the lower roller path being distorted due to firing blows or ship working. Such distortions, though at first sight improbable, did actually occur during the Second World War leading in extreme cases to the turret jamming when the pinions lay across the minor diameter.

The conclusions reached may be summarised that a two—or more—pinion design is a better engineering proposition provided it is differentially coupled and provided the turret support is made adequately rigid and divorced from the constraint of the upper deck. A hydraulic off-mounting multi-pinion design, however, raises problems of its own, since means must be contrived for the load to be shared among the pinions without the use of a galaxy of large bore equalising pipes between the power units.

The above difficulty, happily, does not arise in electrically-driven turrets since the armatures of any number of electric motors may be connected in series ; true sharing of torque then follows.

Roller Path Design

(i) *Parallel* v. *conical rollers.* Roller paths have, traditionally, been fitted with conical rollers. These give true rolling without skidding but flanges are needed on the inner edges to prevent the rollers from working outwards. These flanges add significantly to the friction of the mounting and may even lead to scoring and scuffing.

The alternative is to fit parallel rollers which are at once easy to make to the accuracy necessary and easy to retain in position. Such rollers have a skidding component which may add somewhat to the training efforts, the magnitude of which can be reduced by sub-dividing into two or three or more parts along the axis of rotation. These sections should be of unequal lengths and threaded on in different patterns so that ridges are not worn on the paths.

Arguments as to the feasibility of a scheme which is demonstrably " in-correct " geometrically still rage, though the 5.25 in. *Vanguard* mountings have

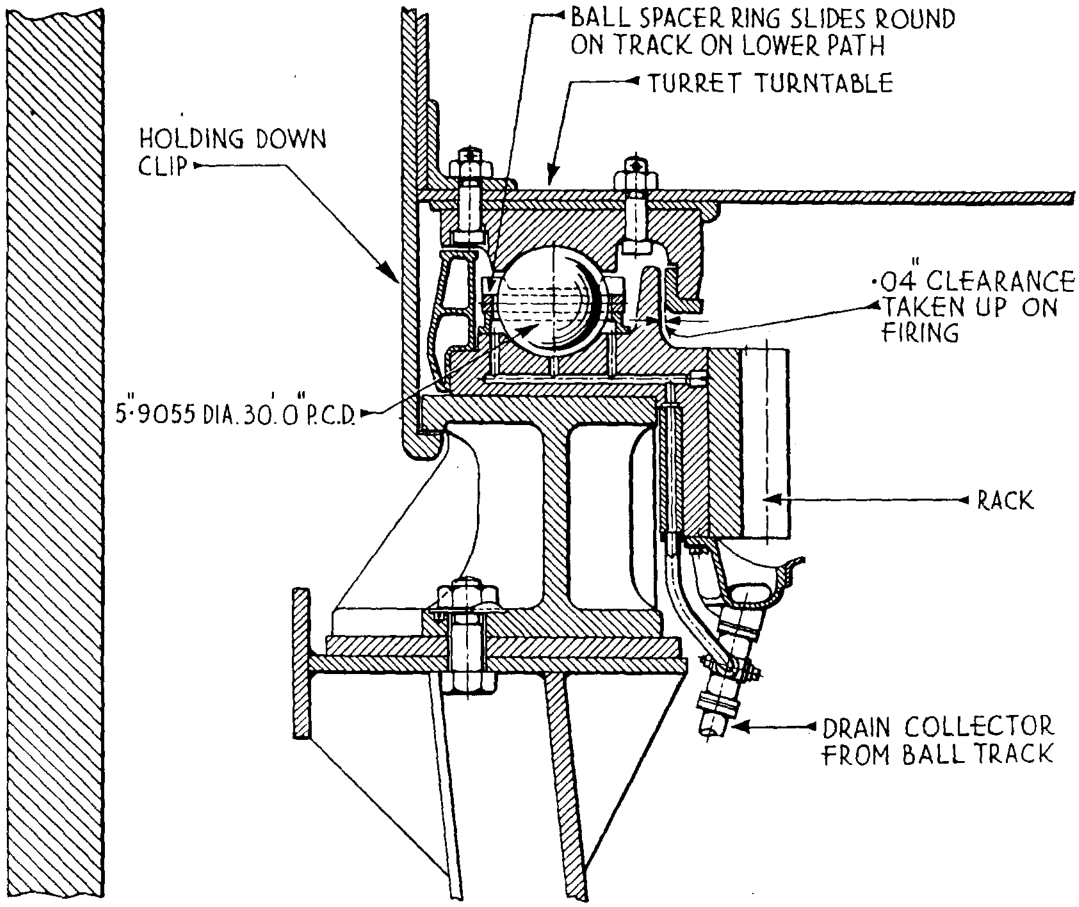

FIG. 9.—BALL RACE ARRANGEMENT IN GERMAN GUNMOUNTINGS

so far proved satisfactory with parallel rollers. More exact data is being sought from a special rig to determine the behaviour in larger sizes.

(ii) *Hardening of roller paths.* Roller paths are notoriously subject to " brinelling " due to vibration or vertical shock. This first came to light in 1940 when ships that had steamed at high speeds for prolonged periods often found it impossible to train directors or turrets smoothly due to the dents that the rollers simultaneously encountered. Staggering the pitch of the rollers was the obvious step though this did not prevent the appearance of the nuisance at the housing position.

Attention is therefore being given to induction-hardened paths and to the use of resilient rollers, these being likely also to soften the blow caused by an under-water explosion.

(iii) *Lateral rollers.* Rollers with flanges on the outer edges to prevent lateral movement of the turret under a list are no longer used since the friction set up (together with the out-of-balance moment of the turret itself) was on occasions enough to stall the training gear. They can be avoided by the use of lateral rollers (Fig. 6).

(iv) *Ball races for turret supports.* The support of gun turrets up to the largest sizes on balls instead of rollers has been German practice through two wars and the Russians are believed to have used 8 in. diameter balls for coast defence mountings. Such an arrangement goes far to avoid the necessity for lateral thrust races and at first sight a somewhat simpler design follows.

The amount of lateral movement of the turret then possible is a function of the difference between the radii of the balls and of the toroidal track in which

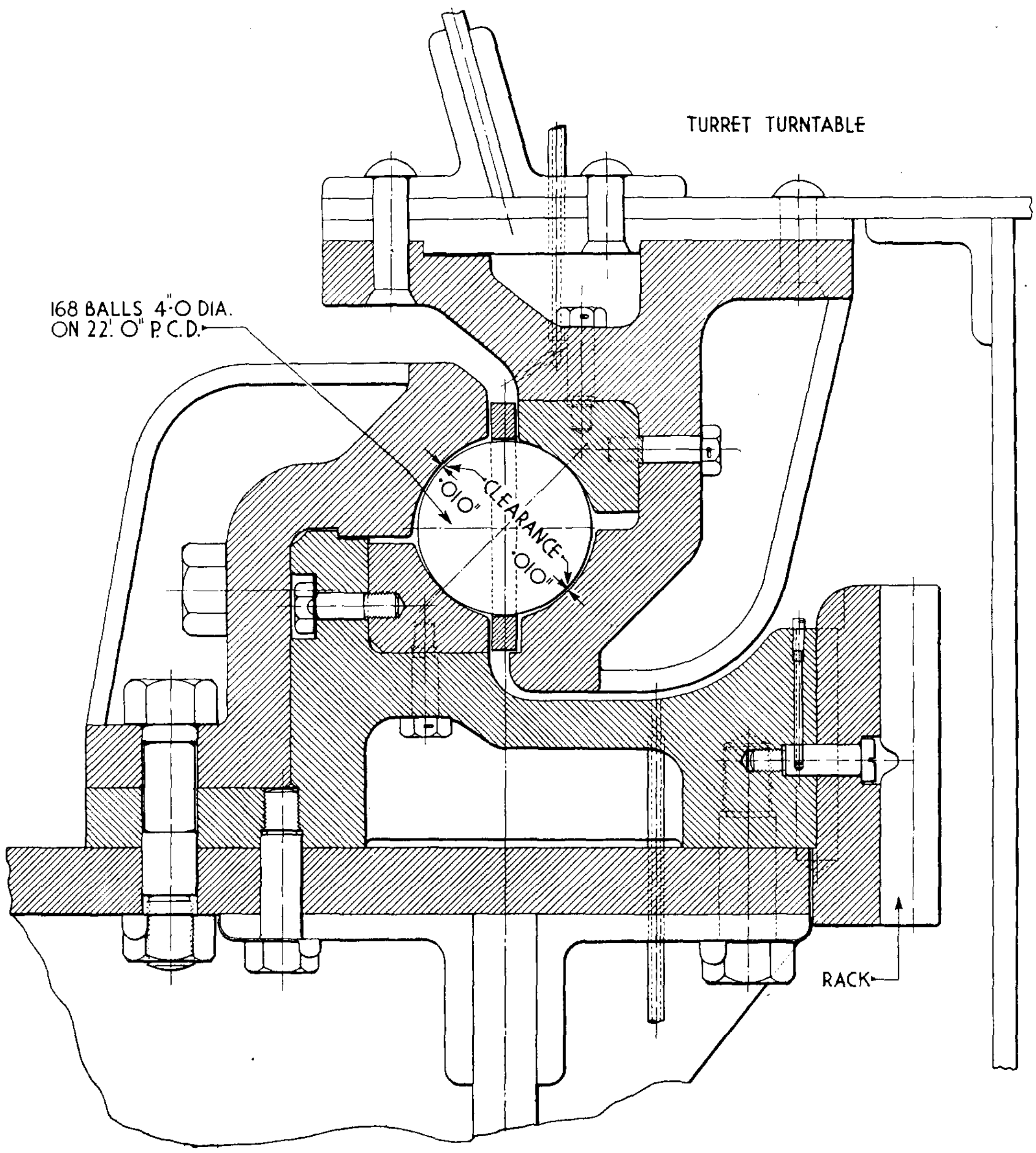

FIG. 10.—ARRANGEMENT TO AVOID RUBBING STRIP

they run. The Germans limited lateral movement by a rubbing strip which, in a 30 ft. diameter path and using 5 in. balls, was set to a clearance of .04 in. (Fig. 9). The balls took the lateral thrust due to the ship's roll but the rubbing strip took the firing blow. The clearance thus provided in the ball track allowed for slight possible distortion in the track due to working of the ship or other causes.

The rubbing strip can be avoided by the use of an angular contact bearing (Fig. 10). While this provides more positive location of the turret a premium is put on the truth of the track since distortion might lead to excessive load on, or even jamming of, the balls.

Neither design appears to compete with normal British practice on the score of weight, and it is certain that much of the successful employment of balls depends upon the machining and heat treatment technique applied to the track if brinelling troubles are to be avoided. These designs are not, therefore, at present being pursued though certain tests on the behaviour of balls loaded to conditions representative of firing blows are being run.

(v) *Roller path strength.* Whatever type of anti-friction bearing is used it is certain that paths must have the maximum possible hoop strength. Path distortion (due either to ship working or to firing loads) may lead to too great or too small pinion tooth clearance or even to turret jamming.

Whenever possible, training racks should be integral with the roller paths and the latter should be part of a circular box structure of maximum possible strength/weight ratio.

Gearing

The present trend in turret machinery being towards high rotational speeds, the development of high-performance and high-efficiency gearing is of major importance.

The requirements of turret training gears are minimum bulk, minimum noise and ability to run for long periods without developing backlash. Assuming the tooth to be correctly cut to the optimum profile shape, the load that it will stand without scuffing, and its life are largely dependent upon the hardening, grinding, lapping, or shaving processes during manufacture.

Induction-hardening provides a quicker, cheaper, and more readily controllable process than the more usual heat-treatment, and this technique is being actively developed by Admiralty as well as by industry generally.

Where gear reductions of 3 : 1 and over are necessary, epicyclic gears probably provide the neatest and least bulky solution. With careful design they need be no more noisy than normal gear-boxes but epicyclic gear designers have rarely, if ever, been confronted by the consideration of backlash. The great majority of commercial gear-boxes are either uni-directional or, if reversible as in automobiles or motor boats, backlash then is of no significance.

Much new ground has therefore to be broken for the gunmounting designer's specification will not be easy to meet. It is not, for example, everyday practice to hob internal helices.

Gunmounting designers are often faced with the need for angled drives. Worm gear being, at present, discredited, bevels must then be introduced. Power drive requirements are not likely to be satisfactorily met by straight bevels, and spiral bevels, already established in fire control work, are now required in the larger sizes. Once again, the backlash specification is something new to those manufacturers who normally specialise in back-axle components, for gears must not only be precisely made but contained in boxes of sufficient rigidity to preserve the tooth clearance under peak or impact loads. The experimental gears now being made will show whether this type is suitable for this new application. If introduced, " hypoid " or " extreme pressure " lubricants may have to be specially carried in ships.

Conclusion

To sum up, therefore, the whole question of training gear design is one of satisfying a large number of important and diverse requirements. Apart from the vital question of reliability, training gear must be as light as possible yet stiff enough to combat excessive wrap-up or servo instability. It must be capable of many hours of heavy duty running without the onset of backlash, and it must offer the minimum frictional drag.

Finally, its efficiency must be as high as possible. With power requirements for modern high-duty turrets running into hundreds of horse-power the demands made by the armament upon the electricity generating capacity of ships are becoming a serious problem affecting hull and machinery design in a significant fashion. Moreover, turret-driving machinery losses ultimately appear as heat with discomfort to personnel and raising new ventilation problems.

Turret training gear, formerly akin to the slewing gear of a dockyard crane, now emerges as high-performance precision engineering. Indeed precision design and manufacture must be the first requirement of gear that will keep a 100-ton turret within 5 minutes of arc of an arbitrary transmission signal while at the same time subjected to violent recoil torques. Also, if the gear is, in fact, well designed and engineered, reliability should automatically follow, together with constancy of servo performances.