MAIN GEARING INSPECTION, ALIGNMENT, AND MAINTENANCE

by

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The following notes on main gearing are intended to provide a background of facts and experience so that when main gearing is inspected, either as a routine or after damage, a broadly based appreciation may be formed with an enhanced confidence to decide the appropriate remedy, or supply the significant information in a report.

There are, very occasionally, gearing troubles arising from obscure or complex causes. These are not dealt with in this article. Instead, stress is laid on the ordinary run of experience in order that it may be fully appreciated that the life of practically all gearing is dependent on a small number of simple and well-understood factors.

These notes are limited to British-built, single-reduction, double-helical hobbed main gears produced between 1937 and 1946, and in which each pinion has two bearings only. This type of gearing gave satisfactory service and, at times, suffered serious misuse without giving trouble.

The standard notes on realignment of gearing are given as an Appendix on page 32.

FAILURES OF GEARING

The straightforward defects in gearing may be divided into five categories :—

(i) Rapid wear due to water in the lubricating oil.
(ii) Scuffing.
(iii) Tooth breakage.
(iv) Damage caused by near miss explosions.
(v) Damage caused by presence of foreign objects.

Rapid Wear due to Water in the Lubricating Oil

Wear of the whole tooth face to an even matt finish is the natural result of water in the lubricating oil. The first part of the remedy is, of course, to prevent the water (gland steam, &c.) getting into the oil and to remove, as far as possible, all water, rust, &c., from the oil and from the lubricating oil system. In many cases the pinion journals will be seriously corroded and worn and will require skimming or regrinding. Concentricity with the original journal is vital. The bearings, also, will require remetalling and reboring.

The effect of the water will have been to give the gears a drastic " run in." The teeth should now bear evenly the whole length of each tooth. In realigning the pinion, the overriding requirement is to reproduce this even bearing. If the water is, in future, kept out of the oil and this even bearing provided, no further wear should take place. The provision of the precise original centre distances at each end is of secondary importance.

Where wear is due to water in oil, a very good practice is to bore both sets of bearing brasses to, say, half the required oil clearance. The clearance on the unloaded side should be taken up with paper ; and then the meshing checked. Adjustment should be by scraping, or packing, the pinion bearing further from the turbine to give the best meshing. Both pinion bearings are then bored out to give the full oil clearance and at the same time preserve the meshing. Before describing other defects it will be as well to examine load distribution.

Distribution of Load

The distribution of load along the length of the pinion face is determined by :—

(i) The elastic deflections of the body of the pinion, and of the teeth themselves, under load.

(ii) Errors in helix angles and parallelism.

(iii) Irregularities in the surface due to daily bands.

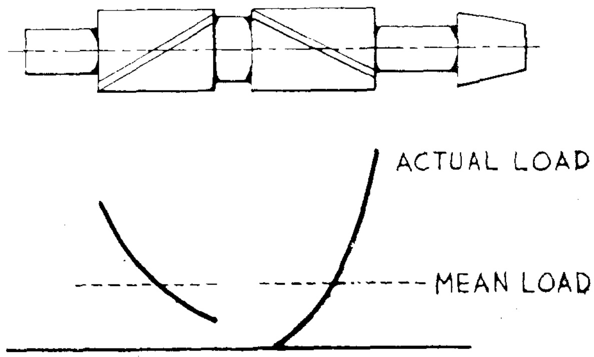

FIG. 1

Elastic Deflection.—Under load, the pinion body, acting as a beam, deflects away from contact with the wheel. At the same time, the body of the pinion twists under torque and the after end of each helix lies back relative to the forward end. The teeth of the pinion and wheel each have a certain elasticity. The amount by which the teeth themselves are deflected is, of course, proportional to the local loading.

In a perfectly accurate gear the result of this deflection is to produce a loading distribution along the pinion as shown in Fig. 1.

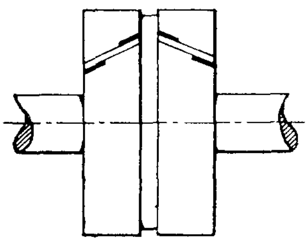

FIG. 2

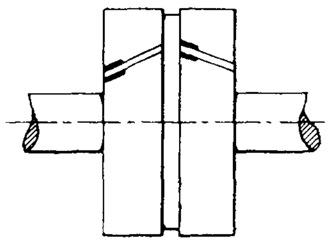

FIG. 3

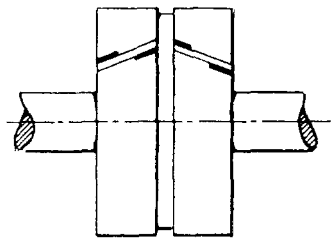

FIG. 4

Errors of Helix Angle and Parallelism.—Differences in helix angle of the pinion and wheel will lead to a concentration of loading at both outer ends of either the ahead or the astern face, and at both inner ends on the other face. (Fig. 2.)

Errors in parallelism of the pinion and wheel axes, in the plane containing these axes, will result in a concentration of loading on both helices and on both ahead and astern flanks at the same end (*i.e.* forward or aft) of each helix. (Fig. 3.)

Errors in parallelism in the plane perpendicular to the plane containing both axes will result in a concentration of loading on both helices at the same end (*i.e.* forward or aft) for the ahead flanks, and at the opposite end for the astern flanks. (Fig. 4.)

There may have been a slight wobble between axis of gearwheel journals and the axis of the hobbing machine table. This will lead to an error in helix angle on one side of the wheel and an opposite error on the other side : this effect will blur the other markings.

Daily Bands.—The finishing cut on a small wheel (*e.g. Hunt* Class destroyer 6 ft. diameter) requires some $3\frac{1}{2}$ days ; on larger wheels more time is required. Where the wheel has been cut in a machine whose temperature is not controlled within fine limits, then the effect of the daily variations in temperature will appear as alternate high and low circumferential bands along the surface of the

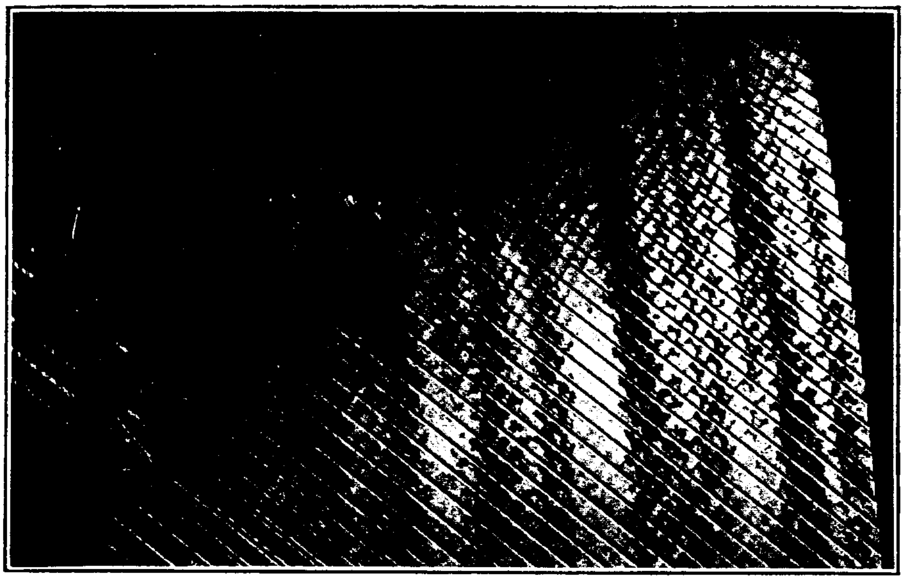

FIG. 5.—DAILY BANDS REVEALED BY HARD MARKING

teeth. The daily range is much greater in cloud-free weather than in overcast, and as the various types of weather occur haphazardly so these bands vary very greatly in severity.

On the high part of the band the additional loading caused by it will be proportional to the extra amount this part of the tooth has to be bent back to bring it into line with the rest. A load of 1,000 lb. per inch of face width will bend a 7/12 in. pitch standard AA tooth back roughly $3\frac{1}{2}$/10,000 in. out of contact. The mean load (in pounds per inch) of various classes at full power is :—

Sloop	700
Destroyer	1,450
Aircraft Carrier	1,260
Duke of York	1,230

At the outer end of the pinions this load is increased due to the effect of deflection as explained above to, roughly :—

Sloop	1,800
Destroyer	4,400
Aircraft Carrier	6,600
Duke of York	6,200

Bands on large wheels have been experienced up to a depth of 3/1,000 in. In an aircraft carrier the total load on such a band coming near the outer end of the pinion would be 11,000 lb./in.

Fig. (5) shows such daily bands revealed by hard marking.

Fig. (6) shows how the teeth on a pinion have broken where they mesh with the daily bands on the wheel. The teeth have been machined away but the fractures at the roots along the daily bands can be clearly seen.

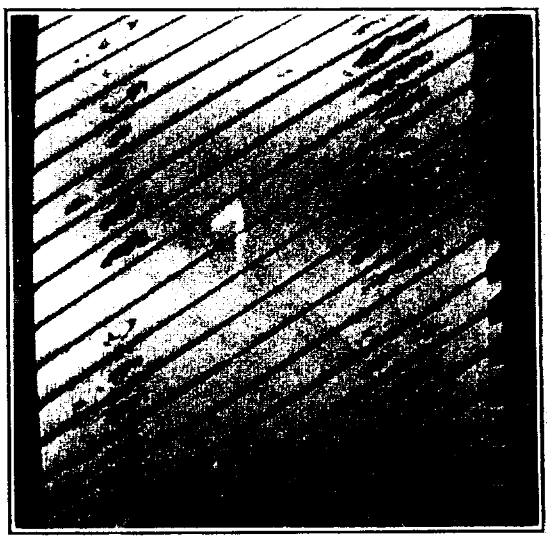

FIG. 6

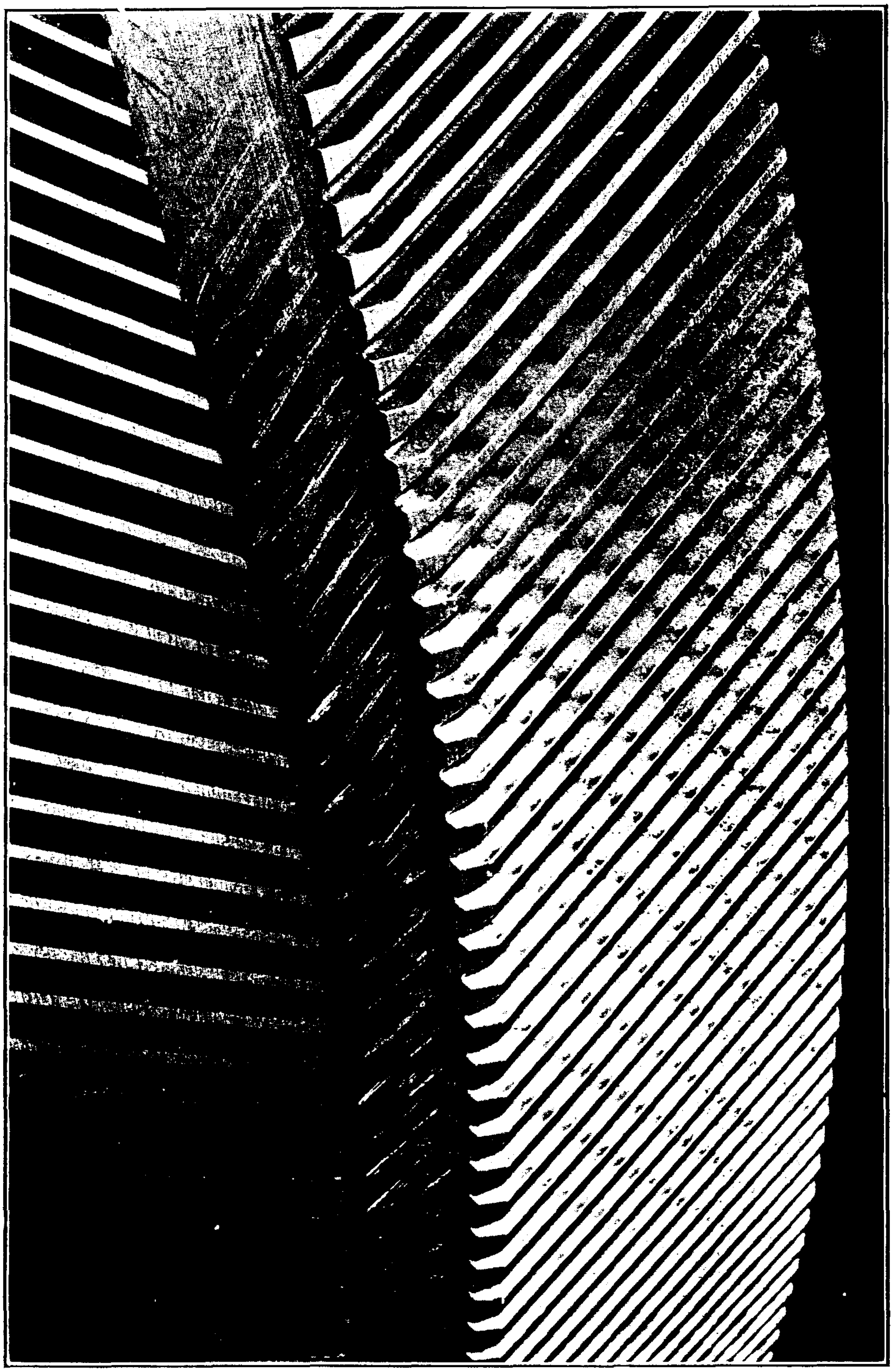

FIG. 7.—NON-CREEP ERROR

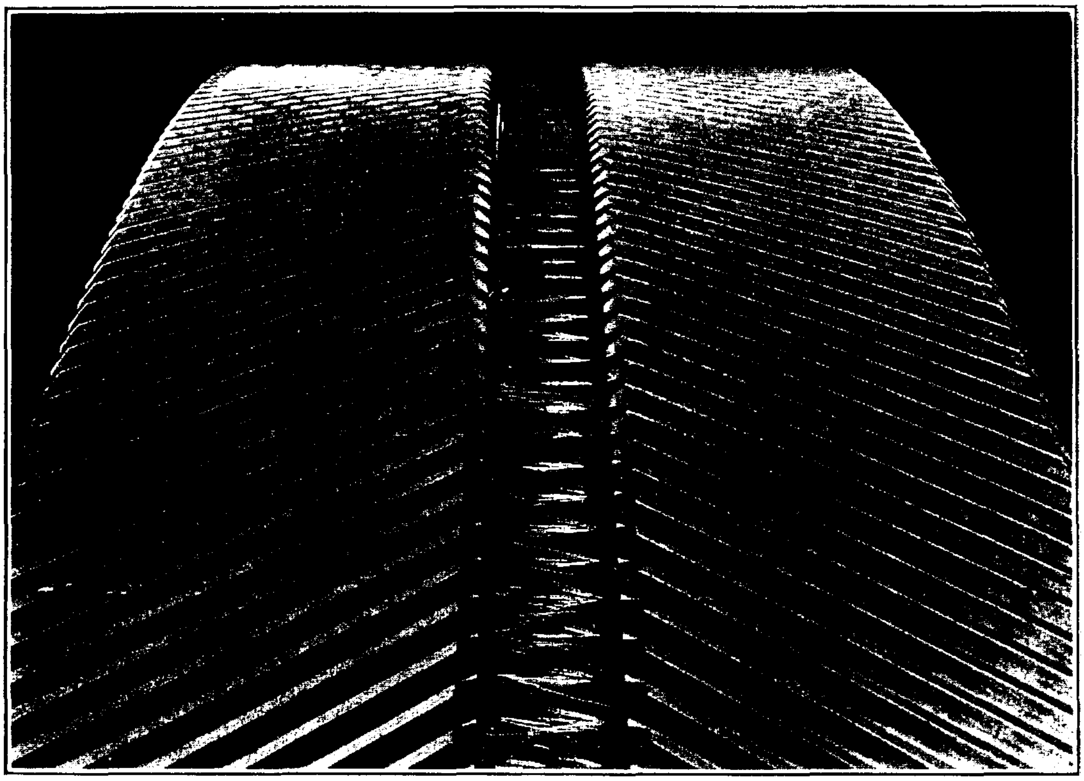

FIG. 7A.—NON-CREEP ERROR

To compensate for these three effects of deflection, errors in helix angle, parallelism, and daily bands, it will be found that certain manufacturers have employed considerable handwork on the pinions, or on the wheels, or have adjusted the bearings, to give the best meshing. Others will have taken none of these measures.

When the work of making good defects has entailed remetalling bearings, the criterion of final alignment is the meshing of the gears to demonstrate the best distribution of tooth contact along the length of the gears. The bridge gauge readings may be recorded, but are not of such direct importance since they do not afford so discriminating a check as the meshing. Reports should always include the result of the meshing test. (Appendix, para. 2.)

A further source of load concentration in surface loading is the undulations on the tooth surface arising from errors in the hobbing machine. These may frequently be seen in a good light on a new gear and appear as a patterned marking after running.

Hobbing Machine Errors

The most common errors in the hobbing machine are now well known and are being eliminated. They are non-creep, creep, and feed screw errors.

Non-creep Error.—For gears cut on hobbing machines not fitted with creep gear an error may arise from the non-uniform motion of the worm-table drive, from the drive to the worm or from the worm abutments. This error, if at all marked, gives rise to a very severe noise concentrated in a single frequency (the frequency naturally varying with the speed of rotation). This single-frequency noise is more disturbing than a noise of similar volume composed of a number of frequencies. This error appears as a pattern of *axial* bands of marking on the wheel teeth (Figs. 7 & 7A). It may be noticed that it is also possible to draw diagonal lines through the hard spots but the important factor is the *axial* nature of the bands.

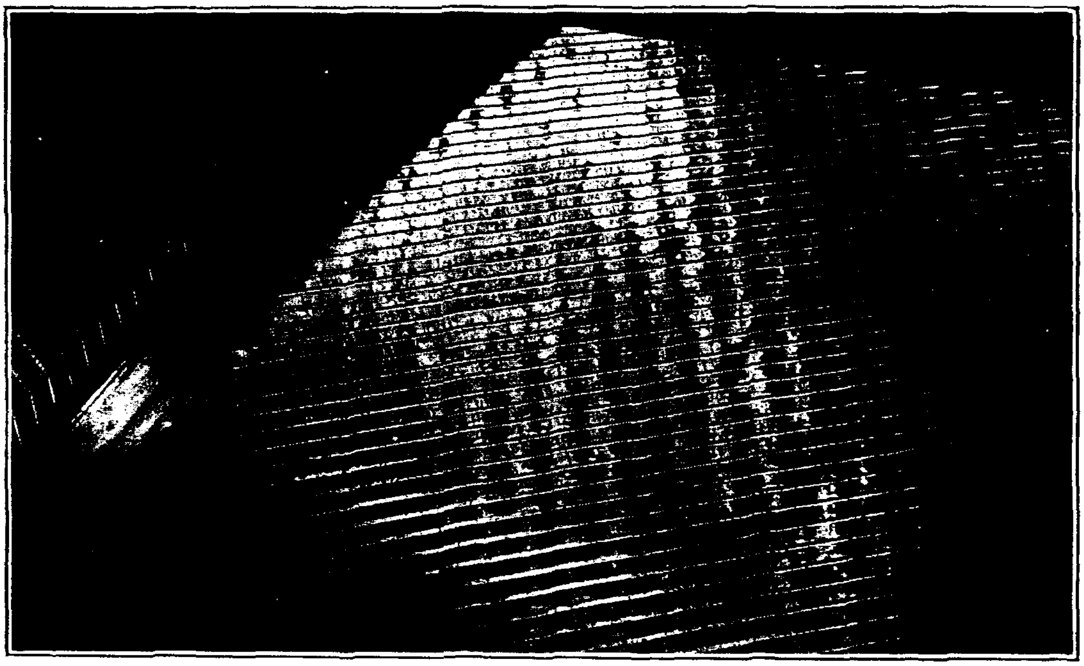

FIG. 8.—CREEP ERROR

FIG. 9.—CREEP ERROR

Only sloops, corvettes, minesweepers, and three destroyers were fitted with gears cut on such machines during the period under review.

It was to avoid the distressing noise caused by gears with this type of error that Sir Charles Parsons introduced his type of creep gear.

Creep Error.—The creep gear spaces out the periodic error associated with the worm drive to the table into a diagonal pattern around the gear wheel, but the creep gear itself has its own error which appears, also, as a spiral of fine pitch and varying amplitude. The complex result of such patterns is shown in Figs. 5, 8 and 9. The pitch and angle of these diagonals depend on the feed used. The marks cannot be linked by axial lines.

Feed Screw Errors.—On gears cut on either type of machines, occasionally

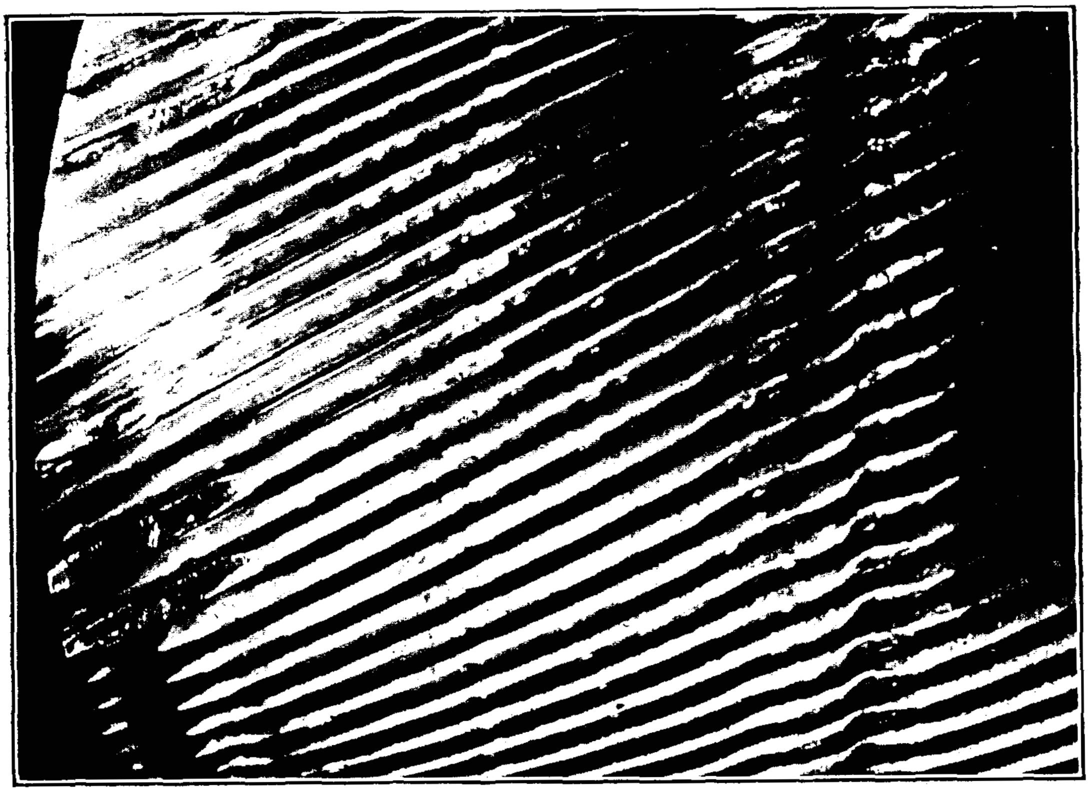

FIG. 10

there may be seen a pattern coming from an error in the hob feed screw drive, or thrust abutment. This appears as a pattern of circumferential bands spaced along the gear at the pitch of the feed screw, say $\frac{1}{2}$ in. (*see* Fig. 10, which also shows teeth broken away where the load concentration due to deflection and daily bands has been highest).

The depth of the above three types of surface markings has been in very many instances about 3/10,000 in. and has been known to be as much as 1/1,000 in. The markings do not seriously affect the value of the bending stress at the root of the tooth. They do, however, impose a severe lubrication condition. At present the lubrication conditions for gear teeth are far from being analysed or comprehended. A guess at the thickness of the oil film where the load carried is 1,000 lb. per inch and the gears of destroyer proportion would put it at between 4/100,000 in. and 1/10,000 in., and for a load of 10,000 lb./in. between 5 and 12×10^{-6} in. If the oil carries the load, as in other hydrodynamic films, then say, three-quarters of the surface will, due to these undulations, carry only one-quarter of the load and on the crest of the load of the undulations the film is very thin. At best the conditions are those of boundary lubrication and very severe.

Present Developments.—The fitting of temperature control during gear cutting is eliminating daily bands. Close checking of the feed screw and of the travel of the hob saddle will eliminate the errors from these sources.

New tests, carried out when running, reveal the periodic errors in the worm-table and hob drives. Similar tests for the " creep ring " are still under development. The errors as revealed are being reduced to very much smaller quantities by painstaking and methodical improvement of the accuracy of the machine components. In addition, the process of shaving marine gears has been developed. This process improves still further the surface finish. The shaving tool shown in Fig. 11 is rotated in mesh with the gear and pressed against it. There is a relative sideways motion of the tool and the gear, and the cutting edges of the tool remove any local proud irregularities in the gear.

In such a way short waves—except those from the non-creep error—can be

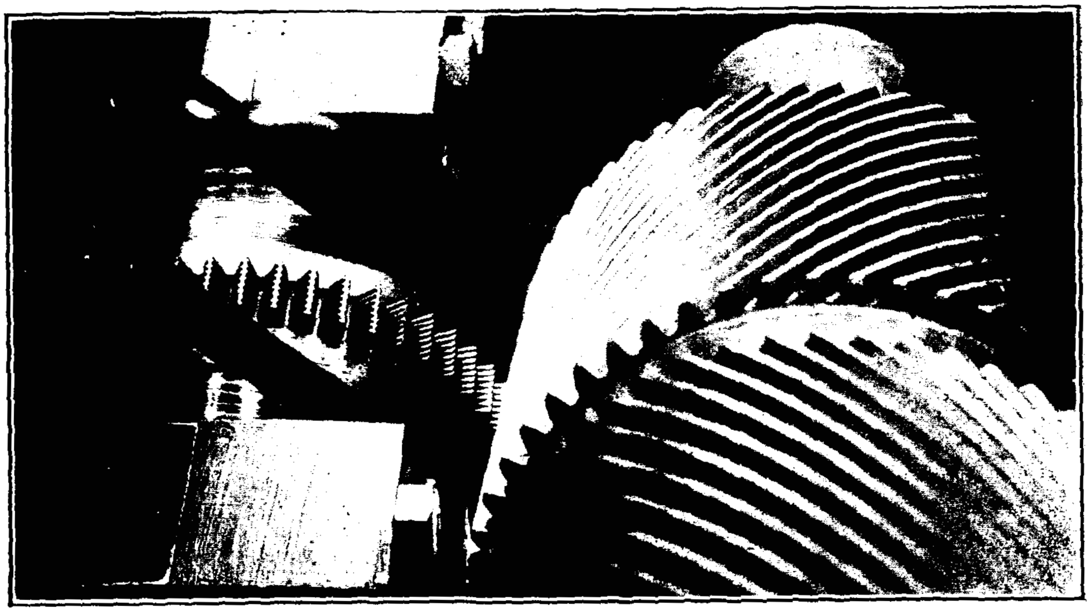

FIG. 11.—SHAVING

effectively removed. The daily bands can also be removed by selective shaving. The improvement of the surface of the gear by the latest practice now permits the accurate measurement of the helical angle, and, hence, the work of reducing the errors in it to be started.

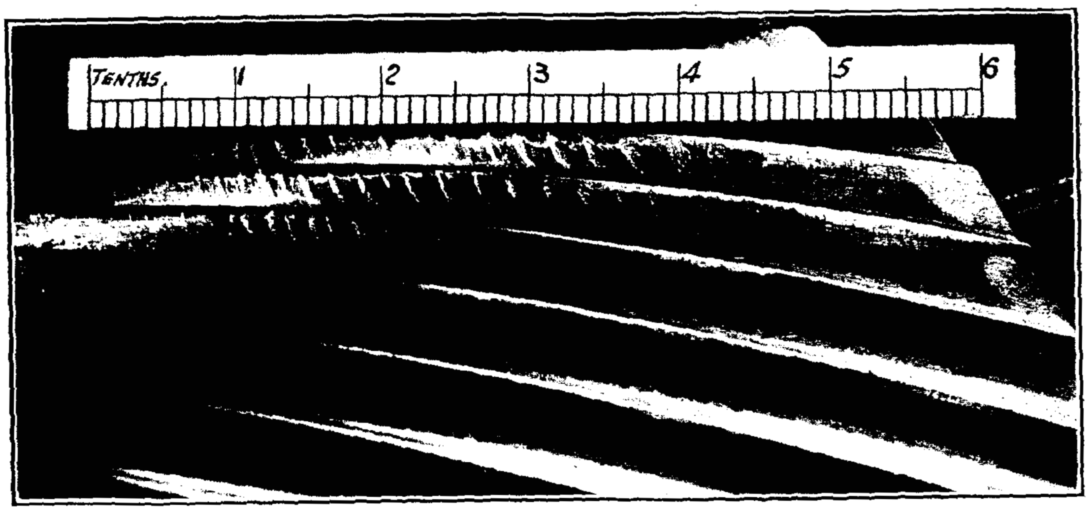

FIG. 12.—SCUFFING

Scuffing

Scuffing consists of the rapid seizure of the gear teeth faces on the ridges of the undulations and, then, of the tearing out from one tooth face of fine specks of metal (Fig. 12) ; rarely does it occur except during the trials of new ships.

It occurred most often where the surface loading and sliding speed were high and, again, where the concentration of loading due to pinion deflection, incorrect helical angles or meshing, severe daily bands or high undulations, were most marked.

It has also occurred at high torques with low speed, *e.g.* sudden full-speed astern.

The immediate wartime cure was to file the tooth surfaces smooth, improve the load distribution by adjusting the meshing where possible, clean out the forced lubrication system, and give the gears, say, two hours running at half-load. It will be realised that running-in of these gears by wearing off the crests of the undulations and spreading the load made the surface less prone to seizure.

The long-term cure is to eliminate the load concentrations.

Root Failures

Several failures have been experienced with a design of tooth form where there was a sharp radius at the base of each tooth flank. The stress concentration in this small radius was more severe than with the standard continuous semi-circular sweep from flank to flank, and so these gears were correspondingly more prone to failure.

When the severity of load concentration, caused by bad daily bands, errors in helical angle, and meshing, is cumulative the local loading can be extremely high. It will be found that this occurs most often in gears of high horse-power and low revolutions, *i.e.* those fitted in battleships and aircraft carriers.

With the large wheels used in these gears the effect of changes of temperature results in a greater depth of the daily band. The depth of the periodic errors is larger in proportion. In addition, the load-carrying capacity and wear resistance of the surface is higher. The relative curvature of the teeth being greater than that for smaller gears, the stresses in the surface of the teeth are therefore lower and there is an enhanced resistance to wear.

Breakage of main gearing teeth has occurred in a battleship and an aircraft carrier, and also in a cruiser which, having through damage lost the use of two shafts, steamed for an appreciable time with a high overload torque on the other two in order to catch the enemy. The distribution of loading in these instances was sufficiently severe to produce stresses in the tooth roots well above the fatigue strength of the material even with the standard continuous fillet.

The long-term remedy, again, is to produce gears free from this concentration of loading and with larger pitches and so, stronger teeth. The short-term remedy has been to cut away the pinion teeth in a circumferential band in way of the fractures.

Damage Caused by Near Miss Explosions

Severe damage to main gearing from near miss explosions may occur in two ways. Damage at, or near, the stern may suddenly stop a shaft. The inertia of the turbine rotors then puts very severe loads on the gearing. Alternatively, if the rotors are suddenly stopped by being damaged themselves, then the inertia of the shafts will impose a very severe load on the gearing ; sometimes the forces have broken the pinion bearing caps.

As the damage will be on an axial band along the wheel and pinion face, it is generally accompanied by a heavy bump once every revolution. Each incident must be treated on its merits. The power which the gear can still transmit " to get the ship home " can only be found by trial, while something in the way of cleaning off " rags " to improve the teeth may be possible, and the meshing should be checked to see whether improvement can be gained. The gear can only be restored to its correct condition by rehobbing or, in the more severe cases, by providing a new pinion and wheel rim.

Damage Caused by Presence of Foreign Objects

In practically all examples of damage from foreign objects, it lies on one or more circumferential bands. When the bands are cleaned away out of contact it is possible to assess how much load the gear will still take ; generally, it will still be a very high proportion of its full load. The important features to check are that the pinion is not bent, that all proud marks raised by the damage are filed back out of contact, and that the ends of the teeth left are given the same chamfer as those at the end of the helix. Once again it is prudent to check the meshing for load distribution.

CARE OF GEARING

The most important factor in the care of gearing is the supply of clean, water free, lubricating oil. On any occasion when, after repairs or alterations to the lubricating system, there is the least suspicion that dirt has got into the system, then, while the ship is in harbour, the oil should be circulated using muslins in the system. Sometimes, even when this has been done, rough weather will stir up further dirt and then muslins should be used again when next in harbour.

In many ships on service—and particularly at moderate speeds—a bumping noise comes from the flexible coupling or the gearing once per shaft revolution. The pinion may be observed to be shuttling axially to and fro.

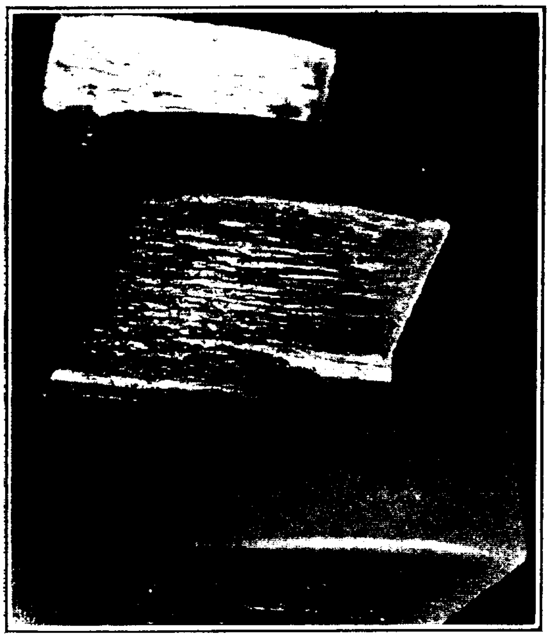

FIG. 13

It appears most often at a speed which is critical for each class of ship, *e.g.* in the *Illustrious* Class it is between 50-60 r.p.m. This speed is that at which the impulses caused by the irregular reaction of the propeller blades to the water corresponds with the natural frequency of the propeller shaft and thrust block vibrating as a torsional spring.

The frequency of the propeller impulses is, of course, the number of blades multiplied by the number of revolutions per minute. When trailing a shaft, the noise effect in the absence of the restraining ahead load is, generally, even more marked, but is not on that account the more serious.

As yet, there has been no evidence to show that such bumping has caused trouble of any kind, save the irritation of the noise itself. While it is prudent to examine the gearing and flexible coupling when such a noise is first noticed unless other symptoms are present it should occasion no undue concern.

On the other hand, if the flexible coupling surfaces become damaged as in Fig. 13, a report should be made immediately.

Even though gears are showing little signs of wear it may sometimes be observed that the load is becoming concentrated at the ends of the teeth. Where this can be remedied by remetalling and correcting the centre of the after pinion bearing, it is sound practice for this to be done when opportunity allows.

APPENDIX

NORMAL PROCEDURE FOR ALIGNMENT OF PINION ASSUMING NO SPECIAL ADJUSTMENT REQUIRED

Pinion Bearing Oil Clearances

The design position and amount of the oil clearance in the pinion bearings must be ascertained from the drawings. In some designs the clearance is arranged concentric with the bore of the bearing houses, in others eccentric and usually on the estimated load line. For an outboard pinion the direction

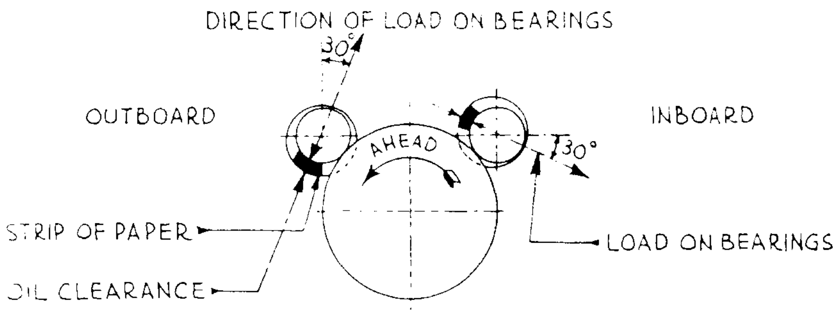

FIG. 14

of load is generally at approximately 30° to the vertical pointing inboard and upwards ; for an inboard pinion approximately 30° to the horizontal pointing inboard and downwards.

The actual oil clearance should be checked by leads. Strips of paper equal in thickness to the oil clearance should be laid in the bearing diametrically opposite to the load line to ensure that pinion will lie in position which it would normally take up when transmitting full load. (*See* Fig. 14.)

Tooth Marking

The tooth meshing should be verified by painting a few of the pinion teeth of each helix with a very thin coating of red lead or Prussian blue marking on both ahead and astern faces. The gears should then be turned ahead and astern and the marking examined. This should show contact distributed over at least 75% of the full width of each helix.

(*Note.*—Since the marking is obtained by driving the gears under only a light frictional load it may happen that the drive is transmitted by one helix only and no marking transferred to the wheel teeth on the other helix. In this event the second helix should be tested by levering the pinion along axially to make contact with the wheel and not by applying a thick coat of marking.)

Alignment of Turbines with Pinions

(i) *Coupling Alignment.*—The alignment of the pinion half coupling can be checked by mounting a dial gauge on the pinion, with its operating spindle bearing on the periphery of the turbine rotor half coupling so that when the pinion is rotated radial readings may be observed with reference to the turbine rotor couplings at 90° apart. The relevant drawing should be consulted and the alignment checked by the use of the alignment mandrels and the checking method outlined in the drawings. Centre distances may be checked by using the alignment mandrels located in the pinion bearings. An inside micrometer is then applied between the ends of the mandrel and the gear wheel journals.

It is not possible to lay down a fixed limit for misalignment which can be accepted. In most cases a lateral displacement shown by an eccentricity of 0.015 in. to 0.020 in., *i.e.* difference between dial gauge readings of 0.030 in. to 0.040 in. could be accepted to avoid adjustment of turbines.

Angular misalignment between the axes up to .0015 in. variation per inch of diameter in the distance between the end faces of the pinion and turbine rotor coupling claws, measured at points 180° apart at maximum radius available could also be accepted if time does not permit of closer adjustment. This corresponds to 0°5′ angular misalignment.

If the conditions shown by the foregoing tests are satisfactory the paper must be removed from the bearings and the gears accepted as suitable for use without further adjustment.

(ii) *Final Check.*—In all cases, however, as a final check an examination should be made as soon as possible after the gears have been run on load. The contact should then be shown by bright polished areas and these should be distributed along the helix to cover at least 75% of the full width.

Any obvious local spots of heavy marking may be eased by careful honing and the same treatment applied if local patches of tooth surface failure (" scuffing ") are apparent.

General Notes on Realignment and Improving Tooth Contact

If the conditions shown by the initial tests outlined in the preceding notes are not satisfactory the errors most likely to be encountered may be classified as follows :—

(i) Incorrect mating of pinion and gearwheel teeth.
(ii) Incorrect depth of engagement of teeth.
(iii) Differences in tooth profile and/or helical angle between pinion and gearwheel.

In most cases where trouble is encountered one or more of the possible errors predominate and each set of markings must be diagnosed in conjunction with the lead thickness measurements to detect these errors.

Misalignments

Misalignments are best considered separately in two planes at right angles, viz :—

(i) The plane containing the axes of pinion and wheel.
(ii) The plane tangential to the pitch cylinders at the zone of tooth contact.

The marking at one end of each helix and on both faces indicated in Fig. 3 shows that the teeth at the left hand are more deeply in mesh.

Alignments can be improved by scraping one of the bearing bushes or remetalling and reboring.

The marking shown in Fig. 4 represents misalignment tangentially.

Alignment may be improved by scraping out of the bearing bushes or remetalling and reboring.

(*Note.*—In making adjustments by scraping the bearing bushes it should be remembered that scraping of the surface can only be effective in correcting alignment if within about 45° on each side of the load line.)

Depth of Tooth Engagement

Provided that alignment is satisfactory, a small variation from designed depth of meshing is acceptable for involute gears as these should mesh equally well at slightly different centre distances.

The mean depth of engagement may be adjusted if required by remetalling and reboring the pinion bearings but the effect on turbine alignment must be borne in mind.

Helix Angle Errors

Appreciable errors in helical angle are unlikely with gears cut to modern standards of accuracy, and such errors can only be corrected by lightly filing or honing the pinion teeth on the hard bearing marks. This type of correction should not be resorted to until all the other methods have been tried without success in obtaining an acceptable distribution of marking. If attempted the handwork should be entrusted only to an experienced fitter, care being required to avoid destroying the involute profile of the tooth. The tooth surface must be finished by using a hone lubricated by paraffin.

The effect of a variation in the helical angles on the wheel and pinion is illustrated in Fig. 2.

The type of marking indicated shows that there is a discrepancy between the helix angles of the teeth of the pinion and wheel helices.

Improvement can be obtained only by handwork as mentioned in the preceding paragraph.

When transmitting torque, some concentration of loading at the forward end of the helices, more especially on the forward one, will arise due to torsional deflection of the pinion. The tooth loading will also cause transverse deflection of the pinion between its bearings which will entail a more severe concentration of loading towards the outer ends of the two helices. This should be borne in mind when judging the results obtained by marking obtained on light load. In general, it may be assumed that deflection will have an appreciable effect in designs which have a long face width, *e.g.* where the ratio of face length plus gap to diameter of pinion exceeds 3.

Particular attention should be given to this point during the inspection after preliminary running on load to avoid the risk of excessive loading being applied towards the extreme ends of the teeth.

Shafting Alignment

Recent experience indicates that it is desirable to check the alignment of the main gear wheel shaft coupling before aligning a new pinion. This is best done by removing the coupling bolts and checking the parallelism of the wheel shaft and intermediate shaft coupling faces. Coupling misalignment tends to cause the main gearwheel to lift in one, or other, of its bearings, thus causing unreliable tooth marking. In extreme cases the wheel has been lifted clear of one bearing. Such a state of affairs can be checked by feelers, micrometer or bridge gauge readings.

A check on the wear of the wheel bearings is also desirable prior to aligning a new pinion.