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# Developments in Marine Reduction Gearing

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***Synopsis***

*The factors that influence the load capacity of large high-speed helical gears are reviewed, and consideration is given to the directions in which improvements may most reasonably be expected.*

*It is shown that the involute helicoid tooth form, which has great practical advantages over any other, can give greater load capacity only if the depth/pitch ratio is increased, and there are practical limitations in that respect.*

*Harder materials offer considerable scope for improvement but also tend to introduce difficulties in cutting. Profile grinding of large gears, expensive though it is, may therefore need to be considered in the future.*

*Important characteristics of gear cutting machines are discussed, and the value of post-hobbing processes is examined.*

*Comments are made on mounting, lubrication and load-testing of high-speed helical gears.*

## Introduction

Since the first successful use of toothed gearing for marine main propulsion, general development has been in the direction of reducing the tendency to noise production and increasing the load capacity or the life expectation for given overall dimensions. Recent developments have all had the same main aims, and the possible means of accomplishing them may be considered under the following headings:—

(1) Variations in tooth-form.
(2) Higher standards of accuracy in manufacture.
(3) Use of materials of higher load capacity.
(4) Refinement in mounting.
(5) Use of lubricants of higher load capacity.

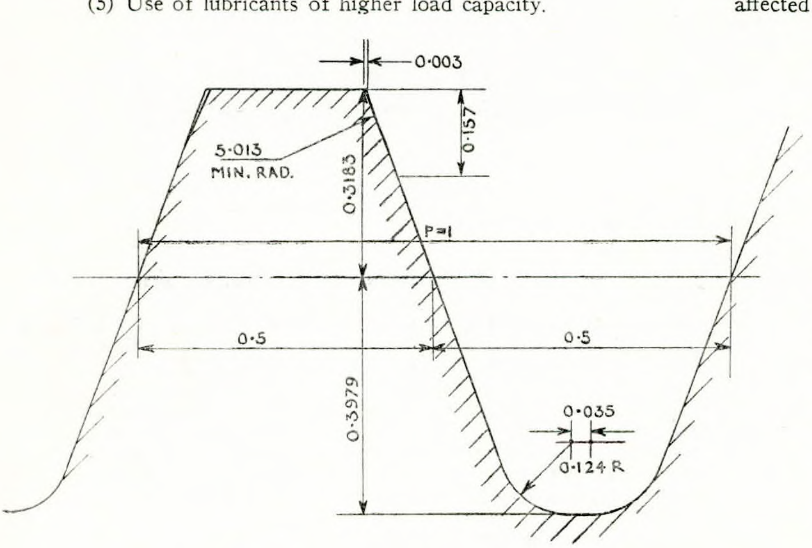



FIG. 1.—*British Standard tooth-form for basic rack*

## Tooth-form

Despite repeated trials of other tooth-forms, the involute helicoid is at present in no danger of losing its pre-eminence in the wide field of power-transmission gearing. It owes its position to the practical advantages associated with the elemental simplicity of its basic rack-form which has plane flanks (Fig. 1) and to the wide range of gears that can be cut with a single generating cutter. There is scope, however, for considerable variation in dimensional details of involute helicoidal gears, and a number of different systems have been used in the past, but for general purposes the present British Standard system cannot be surpassed. Within the relatively restricted field of marine main propulsion gearing, it is, however, possible that different tooth-proportions may be preferable and it is of interest to examine the nature of the problems involved.

The factors to be considered are:

(1) Ratio of working depth of tooth to normal pitch.
(2) Normal pressure angle.
(3) Helix angle.
(4) Ratios of addendum and dedendum to normal pitch.
(5) Form of root fillet.
(6) Form of profile modification to reduce shock on tooth engagement.

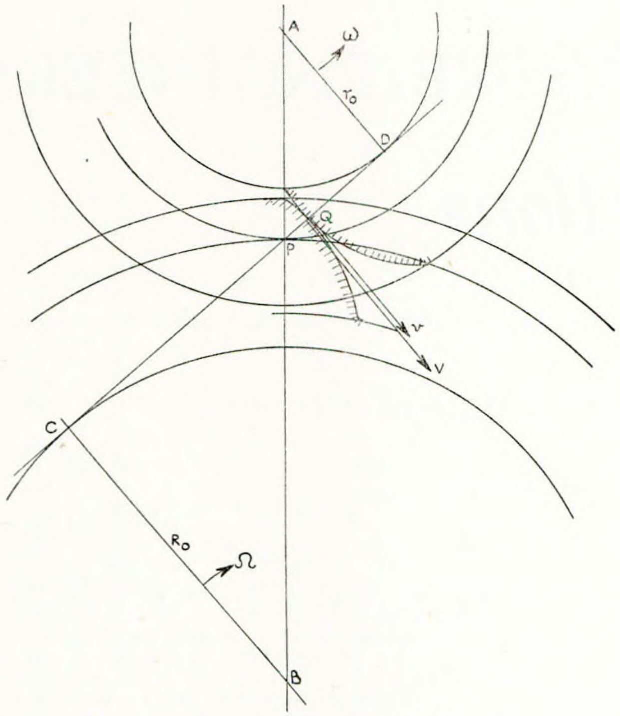
Factors (1) to (4) are all concerned in the question as to what particular form of the involute helicoid tooth affords the greatest load capacity for a given velocity ratio, centre distance and facewidth.

It is shown in Appendix I that, given centre distance, ratio, facewidth and materials, the load capacity of the teeth as determined by surface stress is limited by the ratio of tooth-depth to pitch and is affected but little by changes in normal pressure angle or helix angle.

The geometrical limitations to this ratio are the tendency to produce sharp-pointed teeth if the pressure angle is large or under-cut teeth if the pressure angle is small. In either case, and particularly in the latter, the relative slenderness of the teeth and of the teeth of the cutters that produce them tends to introduce difficulties in manufacture. Shaving cutters are specially affected by this consideration.

The British Standard ratio of working depth to normal pitch is 0·636. A ratio of nearly unity has been tried and found to introduce difficulty in cutting accurately by reason of the tendency to excessive deflexion of the teeth of the cutter and of the work. A ratio of 0·8 is suggested as a compromise applicable to the vast majority of marine main propulsion gears.

There is a need to determine a standard basis of tooth design for large high-speed gears and in doing so to remember the desirability of keeping down the number of different designs of hobs and shaving cutters. The line of least resistance for the gear designer is to assume that special tools will be made for each job, and the tool-designer is often reluctant, for one reason or another, to question the need for so much variety as is sometimes demanded. True skill in gear design shows itself by the ability to use standard cutting tools of simple form to produce gears to meet any normal requirement. At one time this would have been regarded as axiomatic, but tendencies in some directions




$$V = \Omega CQ = \Omega CP + \Omega PQ$$
$$v = \omega DQ = \omega DP - \omega PQ$$
$$\Omega CP = \omega DP$$

Hence sliding velocity at $Q = V - v = (\Omega + \omega)\ PQ$.

FIG. 2.—*Sliding velocity between gear teeth*

seem to suggest that the art of gear design, like so many others, is dying out.

**Sliding Velocity**

The maximum sliding velocity between gear teeth is equal to the product of the sum of the angular velocities of the gears and the greater of the two distances from tip circle to pitch-point measured along the line of action as shown in Fig. 2. This distance increases with the depth of tooth and so deep teeth have relatively high maximum sliding velocities.

If the ratio of depth to pitch of tooth is fixed, low sliding velocity demands a fine pitch, but a limit in this direction is set by consideration of bending stress.

To equalize the maximum sliding velocities at the tips of the teeth of wheel and pinion the addenda must be different to an extent that depends on the numbers of teeth in the gears but they need never differ by more than about 10 per cent. from half the working depth of the tooth.

For helical gear teeth of British Standard proportions the maximum sliding velocity in feet per minute is approximately equal to the product of half the normal pitch in inches and the sum of the speeds of the gears in r.p.m.

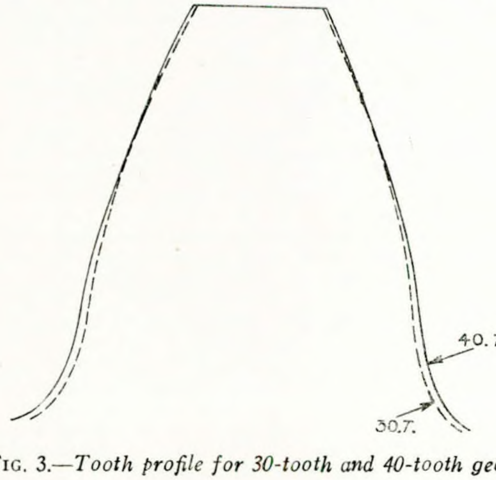



FIG. 3.—*Tooth profile for 30-tooth and 40-tooth gears*

**Bending Stress**

For teeth of geometrically similar section similarly loaded, the bending moment about the neutral axis of the weakest section is proportional to the pitch and the modulus of the section is proportional to the square of the pitch.

Hence the bending stress set up by given loading on a helical gear of any given diameter, width and helix angle would be inversely proportional to the pitch if the pitch could be changed without change in geometrical form of the tooth. Actually the change in form for a small proportional change in pitch (i.e. for a small proportional change in the number of teeth) is negligible unless the number of teeth in the gear is less than about 30 and so it is very nearly true to say that bending stress is inversely proportional to pitch (see Fig. 3).

**Form of Root Fillet**

As gear teeth are subjected to load repeatedly applied, any failure by bending is caused by fatigue and so stress-raising influences must be carefully avoided. For this reason, the root fillet should have the largest practicable radius and the British Standard basic rack form has this characteristic. It means that the clearance below the inner limit of working depth is greater than would be required with a fillet of smaller radius, but the effect of increased root bending moment on the root-stress is more than offset by the effect of larger fillet-radius. An even greater clearance may be useful in facilitating the application of shaving cutters.

**Tooth Profile Modification**

In straight tooth spur gears, deflexion of the teeth that are loaded at any instant causes unloaded teeth in one gear to be "out of phase" with unloaded teeth in the mating gear. Consequently the first contact of a pair of teeth must bring them into phase as defined by the loaded teeth. If the profiles are geometrically correct for unloaded meshing this initial adjustment of relative position takes place with an impact. It is normal practice to give the tips of spur gear tooth profiles a tapering "relief" in order to permit contact to commence much more smoothly than would otherwise be possible. This "tip relief" is a particular case of what is basically "approach relief".

In helical gears, contact does not commence over the whole width of the tooth-tip as in the case of spur gears but at tip or root of the leading end-section of the tooth. The desirable approach-relief is therefore not tip relief but "end relief". It is preferable in general that contact lines on gear teeth should not extend quite to the tip as otherwise there may be a slight tendency to penetration of the oil film on a driven tooth by the sharp tip of a mating tooth. Helical gear teeth may therefore be provided with some small tip relief but it is not required for the purpose that makes it imperative in fast-running spur gears.

The amount of end relief in helical gears for marine main pro-

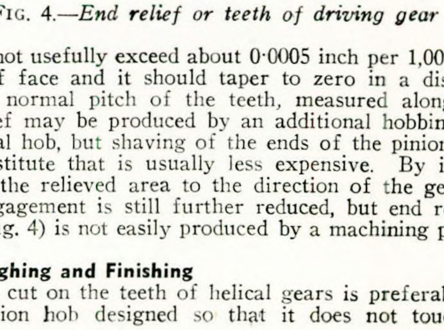



FIG. 4.—*End relief or teeth of driving gear*

pulsion does not usefully exceed about 0·0005 inch per 1,000lb. load per inch width of face and it should taper to zero in a distance about equal to the normal pitch of the teeth, measured along the helix. The end relief may be produced by an additional hobbing operation, using a special hob, but shaving of the ends of the pinion teeth is an adequate substitute that is usually less expensive. By inclining the boundary of the relieved area to the direction of the generator, the impact at engagement is still further reduced, but end relief of this nature (see Fig. 4) is not easily produced by a machining process.

**Hobs for Roughing and Finishing**

The final cut on the teeth of helical gears is preferably made by a high-precision hob designed so that it does not touch the root

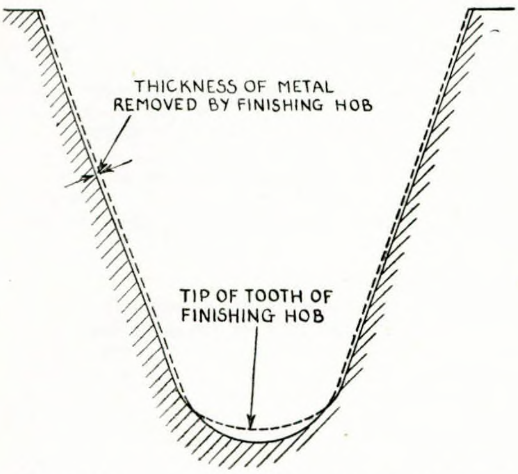



FIG. 5.—*Tooth-forms produced in large gears by roughing and finishing hobs*

fillet. This has the advantage that the life of the hob is much longer than if it treated the whole tooth profile because the first hob-wear that necessitates sharpening is confined to the part of the tooth-tip that cuts the fillet (see Fig. 5).

The hob for the preceding cuts should therefore be designed to cut to full depth whilst leaving on the flanks a few thousandths of an inch of metal for the finishing hob (and the shaving cutter) to remove. There might seem to be a case for providing the roughing hob with "tip raised" teeth so that the root form of the semi-finished gear tooth allows the final cuts to be made without leaving a step, but chamfering of the tips of the teeth of the finishing cutters is a preferable alternative.

**Accuracy of Gear Teeth**

The first reason why a high degree of accuracy is required in the teeth of high-speed gears is that objectionable noise may be produced by tooth-form errors of less than 0·0005in. The second reason—second only because its existence is not so immediately obvious as in the case of noise—is that errors of less than 0·0005in. may cause appreciable local stress concentrations and thus compel the nominal tooth loading to be less than might otherwise be permitted. Excessive noise is objectionable only because of its effect on human beings subjected to it and does not necessarily imply mechanical inefficiency or overloading. The other main effect of inaccuracy has been countered by adopting sufficiently low nominal stresses but in some fields it is now becoming necessary to consider heavier loadings in order to minimize size and weight of gears.

The flank of a gear tooth hobbed with perfect accuracy is not quite smooth as it is formed by separate cuts by the flanks of the hob teeth moving in circular paths. The shape cut by a single flank is a shallow depression of elliptical form but the resulting surface is that produced by the overlapping of such depressions regularly spaced in a manner determined by the form of the hob and the rate of the feed-motion. In ideal circumstances the height of the ridges between the facets in a hobbed helical gear would be of the order of one fifty-thousandth of an inch.

It is clear that quite small errors in the mechanism of the hobbing machine can cause tooth-surface errors greater than this. In general, cyclic irregularity of motion causes the depths of cuts by successive hob teeth to vary between a "low-cut" and a "high-cut" but the resulting variation in tooth profile is often less than the difference in depth of such cuts simply because the overlapping of low-cut depressions (if they are sufficiently close together on the gear tooth) may remove all trace of the intervening high-cut and of many of the intermediate cuts. In general, the high-spots of the resulting surface are at the same positions as the high-cuts.

**Distribution of High Spots**

A cyclic error that passes through an exact whole number of cycles during one revolution of the table of the hobbing machine causes the gear tooth profiles to have high-spots disposed on lines parallel to the line of contact of the tooth with mating teeth. This is the most undesirable distribution of high-spots because it means that during rotation of the mating gears, lines of contact encounter high-spots in groups that they strike simultaneously over their whole length and so the resulting shock and noise are the maxima that high-spots of any given height can produce.

No variation in frequency of the error (provided that it completes a whole number of cycles in one revolution of the table) can alter this pessimum distribution, although a high frequency, by placing the low-cuts close together, emphasizes their overlapping effect and reduces the height of the high-spots.

The most serious error of this type arises from the worm that meshes with the "main worm wheel" attached to the hobbing machine table. During a revolution of the table it completes as many cycles as there are teeth in the table gear. Increase in that number of teeth reduces the height of the high-spots for a given worm error, but still leaves them in the most objectionable distribution.

Reduction of the axial feed of the hob per revolution of the table reduces the variations in height along lines running in the direction of the line of contact with the mating gear but does not affect the distribution of high-spots or their height above the intervening low-spots.

**Creep Machines**

A hobbing machine constructed in such a way that during one revolution of the table no other moving part in it completes an exact whole number of revolutions is called a "creep machine". That effect is in fact secured for the table gear itself if its pitch-point of engagement with its mating gear moves circumferentially in relation to the pitch-point of engagement of hob and work as the cutting operation proceeds (see Fig. 6). Few hobbing machines have that characteristic and so few "creep" machines are creep machines in every respect.

The first advantage of a creep machine is that the undesirable

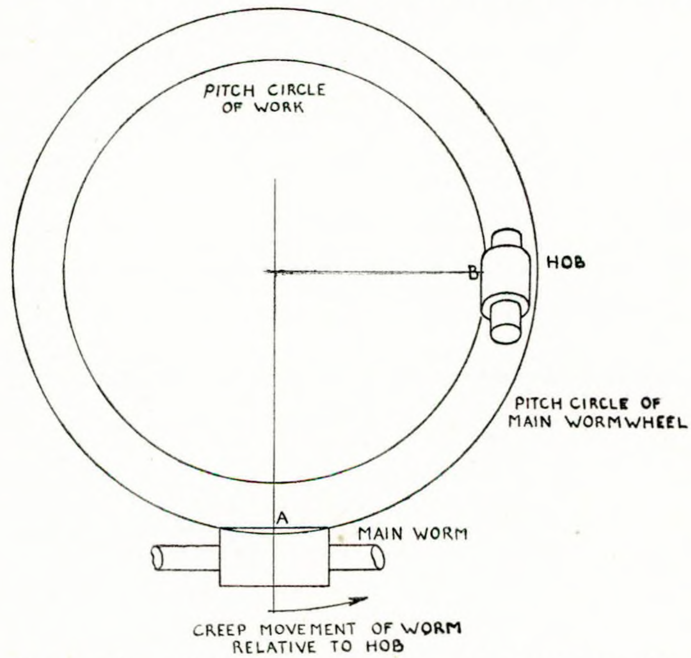



FIG. 6.—*Essential kinematic arrangement of true creep machine*

low-cut distribution associated with the products of "non-creep" machines is avoided.

The second advantage is that by suitable choice of creep effect, low-cuts can be situated so as to overlap each other more effectively than in the case of a gear cut on a "non-creep" machine with the same feed rate.

The third advantage is that with a suitable choice of creep effect and use of a sufficiently fine feed in hobbing, it is possible to cut teeth with a smoother finish than on a "non-creep" machine, however fine the feed.

If the creep machine has serious errors at low frequency the rate of feed necessary to obtain a smooth finish on gears cut by it may be prohibitively fine and in fact gears cut on early creep machines, often showed a rough finish although achieving the quiet running that is the main object of creep. Although the creep principle would be superfluous in a perfectly accurate machine, that is an impossibility and it is important to achieve the minimum practicable errors in constructing even creep machines.

**Creep Fraction**

The fractional part of the number of revolutions made by any part of a hobbing machine during one revolution of the work-table is called the "creep fraction". It follows therefore that every hobbing machine may have as many creep fractions as it has rotating parts. As the most important parts from this point of view are the main wormwheel and its mating worm, it is common to apply the term "creep machine" to one in which the creep fractions of these parts differ from zero even though the creep fractions for other parts may be zero. Conversely a machine with zero creep fraction for main wormwheel and worm is regarded as a non-creep machine even though other parts of it may not have zero creep fractions.

The number of revolutions made by any part during one revolution of the table may be conveniently called its "velocity ratio". The habit of describing the velocity ratio of the main wormwheel as the "creep ratio" is at once unneccessary and misleading.

Although any degree of creep shows some advantage over none at all, it is useful to select creep fractions with due regard to the distribution of cuts on the teeth of the work. In the case of main worm error, for example, low-cuts with zero creep are located on lines parallel to the involute helicoid generator and intervening ridges are left. A creep fraction of ½ causes alternate low-cuts to be staggered by half the pitch of the non-creep ridges with the result that what were originally ridges and troughs become lines of low-cuts at double the original spacing. The maximum height of irregularities is thus reduced and the ridging effect is almost eliminated. By use of a creep fraction slightly different from ½, the high-spot lines become inclined to the generator as is desirable (see Fig. 14, Appendix III).

#### Accuracy of Hobbing Machines

In order to attain the degree of accuracy necessary for quiet running of large high-speed gears, the hobbing machine used for cutting them needs to be made with exceptionally small limits of error in relation to its size. This may be appreciated by reference, for example, to the dividing wormwheel which may have some 700 teeth on a diameter of 10ft. The positional error of any point on any tooth flank in relation to any other tooth is less than 0·002in. in a modern machine of the highest class. Similarly the screw that imparts the feed motion to the hob saddle must be so accurate that the positional error of any point on the thread flank in relation to any other point does not exceed 0·0003in. plus 0·0001in. per foot of axial distance between the points. The hob saddle guides must not depart from straightness by more than 0·0005in. at any point in their length, which may be as much as 10ft.

An obviously important requirement is that the work-table shall rotate truly about a fixed axis and shall not be free to float laterally by more than a fraction of a thousandth of an inch. Its weight holds it in contact with the surface that supports it and so prohibits any vertical freedom, and the same principle may be used to provide lateral restaint. The table may be supported in double conical slideways in which case the weight ensures both vertical and horizontal location or alternatively and preferably the weight may be taken by flat surfaces and a plain cylindrical journal bearing used for lateral location. In the second case lateral pressure is exerted by springs or otherwise either on the edge of the table through rollers or on the journal through an insert in the bearing shell and the bearing shell is cut away for an angle of about 15 deg. on each side of the lateral load line. By this means the journal is caused to "bed" on two separate arcs and is completely located in all horizontal directions (see Fig. 7). The principle of this method of location is not new, but it does not seem to have been used in machine tool practice so frequently as might have been done with advantage.

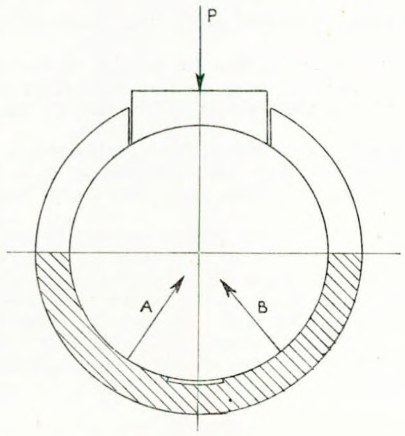



FIG. 7.—*Divided bearing with pre-loading*

Any attempt to locate journals of (say) 10in. diameter within 0·001in. simply by sufficiently reducing bearing clearance is fraught with considerable risk of seizure, as any rise of temperature tends to diminish the clearance and thus has a cumulative effect. The principle of spring loading avoids that danger and also avoids any need for exceptionally fine tolerances on the diameters of journal and bearing. It is applicable to hob spindle bearings and is in fact even more valuable there because of the higher surface speed of the journals.

#### Temperature Control

Unless special precautions are taken, atmospheric conditions can produce temperature differences between different parts of a hobbing machine and differential thermal expansion may cause distortions that result in unacceptable errors in the work. For that reason it is now the practice to house high-precision hobbing machines in temperature-controlled rooms and this provision is equally useful for both erection of the machine and its subsequent operation.

As the coefficients of thermal expansion of the materials (all either steel or cast iron) of work and machine are equal it is not necessary to insist on the maintenance of one particular temperature but it is important that a uniform temperature be maintained in all parts throughout the gear-cutting operation. It is nevertheless usual to employ apparatus for thermostatic control of the atmosphere round specially housed hobbing machines.

#### Post-hobbing Processes

In the past, the errors in hobbed gears were sometimes large enough to produce perceptible departures from uniformity of load distribution on the teeth, and the finish of the teeth of gears cut on creep machines often obviously left something to be desired.

As an alternative to the post-hobbing process of hand-filing the teeth, which was apt to occupy highly skilled men for long periods on one pair of gears, use was sometimes made of lapping, either by running the gears together with abrasive whilst mounted in their gear case or by running one or both gears in mesh with a narrower cast-iron gear coated with abrasive paste and traversed across the facewidth. This process was apt to be slower than filing until use was made of tooth forms that had an abnormal amount of sliding, which accelerated lapping but also increased the tendency to scuffing in service.

#### Crossed-axis Shaving

The most useful post-hobbing process at present in regular use is that of crossed-axis shaving. The gear is rotated in mesh with the shaving-cutter, which is a helical gear of the same normal base pitch, with teeth deeply serrated in planes perpendicular to its axis. The helix angle of the cutter is such that it meshes with the gear with a shaft angle of 10 deg. to 30 deg.; the result is that rotation of gear and cutter causes sliding of the teeth in a direction that has a component along the tooth helix and this causes the edges of the serrations to take long narrow shavings from the flanks of the teeth. The shaving cutter is slowly traversed across the facewidth of the gear.

Gear and cutter may be rotated rapidly without setting up an excessively high cutting speed and the immediately obvious result of shaving is that, in a much shorter time than is required for a finishing cut in the hobbing machine, it improves the finish even of teeth hobbed to the highest degree of smoothness attainable in commercial practice.

Another effect is that the tooth-to-tooth pitch errors are made comparable with those of the shaving cutter itself, which is a high precision tool. The exact value of this effect clearly depends on the magnitude of local pitch errors before shaving commences. The accuracy of high-class hobbing is such as to leave little scope for improvement in this respect by shaving.

The tooth load in shaving may be secured either by

(a) forcing cutter and gear tightly into mesh ("crowding" or "crush-shaving"), or

(b) rotating the pair by driving torque applied to one member and resisting rotation by braking torque applied to the other ("brake-loading").

The first method saves time as both sets of flanks are shaved simultaneously but the tooth thickness of the cutter has to be properly related to that of the gear if the two are to be tight in mesh with contact extending to the full working depth and yet without interference of tip and root.

The second method allows one cutter to be used for a wider range of gears than is possible by the first method.

By either method it is possible to vary the amount of metal removed at different parts of the facewidth, by variation of tooth loading as the operation proceeds or by repeated shaving over certain parts of the width. (If the crowding method is used, "selective shaving" requires one set of shaver tooth flanks to be without serrations). This "selective shaving" may be used to correct slight errors of helix angle or small departures from uniformity of lead of helix revealed by trial contact marking of the mating gears.

Small departures from the fundamental involute helicoid form of the gear teeth (e.g. easing of tip and root) may be secured by appropriate modification of the cutter tooth-profile from the truly involute helicoid form. Such modifications tend, however, to limit the application of any one cutter to gears with numbers of teeth within a certain range. The "brake-loading" method of shaving has an advantage here because it permits some variation in depth of engagement of cutter and work (if the cutter-teeth be sufficiently thin) and, with the same proviso, a shaving cutter may be set to shave a gear of any number of teeth and any amount of addendum-modification, if they have the same normal base pitch.

**Running-in**

As a means of correcting any slight mismatch of the teeth after the gears are mounted in their bearings, the use of a running-in oil is suggested as a preferable alternative to the corrective lapping that has sometimes been employed at this stage.

A running-in oil permits smooth abrasion to take place where local pressures are *high* in conjunction with relative sliding of the tooth surfaces. When satisfactory bedding has been achieved, the normal lubricating oil is to be substituted. Unfortunately some running-in oils are difficult to remove completely from metal surfaces on which they have been placed and their effect is apt to continue for some time after it has ceased to be useful.

**Grinding of Gear Teeth**

The profile-grinding of the teeth of gears less than about 15in. diameter has been commonplace for 25 years or so but application of the process to gears of say 6ft. diameter is only just coming seriously under review.

Gear teeth can be satisfactorily cut in steels with tensile strengths up to 75 tons per sq. in. but anything much harder than that needs to be finished by grinding, unless distortion in the hardening operation can be so closely controlled as not sensibly to impair the accuracy of the hobbed gear. This latter result is achieved in quantity production of gears less than about 10in. diameter but there is no evidence at present that it will become practicable for large case-hardened steel gears.

So profile-grinding needs consideration for case-hardened steel gears at least, and existing equipment is capable of dealing with marine gear pinions. Such a pinion in conjunction with a 75-ton steel wheel, which need not be ground, would have a load capacity nearly double that of gears of similar size made in the materials normally used at present and therefore offers the possibility of a considerable advance in power/weight ratio without exceeding the limits of present-day production technique.

Even when the teeth are afterwards to be profile-ground, the distortion of a large wheel in the carburizing and quenching operations demands special methods for its adequate control. Grinding cannot be used to correct accumulated pitch errors of more than about 0·020in. as it would remove too much of the case and would, moreover, be slow and expensive.

**Gear Materials**

The fundamental difficulty in the way of developing high-duty materials for gears is that, broadly speaking, high resistance to wear and surface fatigue is accompanied by difficulty in machining with precision. At present marine main propulsion pinions are made in steel of about 60 tons per sq. in. tensile strength and the mating wheels in 45-50 tons steel.

The performance of materials considerably harder than these can be fairly safely predicted from their load capacity in gears of small size, but actual experience of their use in large gears is desirable.

Precision-cutting of gear teeth is possible in steel with a tensile strength of about 70 tons per sq. in. and such material has a surface stress capacity about twice as high as that of 45-ton steel. Use of 70-ton steel for the wheel would require the material of the pinion to have a tensile strength of 90 to 100 tons per sq. in. but this is beyond the limit at which precision-cutting is possible with high-speed steel hobs although it may be found possible to deal with it by means of tungsten-carbide tipped hobs. Profile-grinding, if found practicable in large sizes, would be a possible, but expensive method of finishing the profiles of teeth made in such material. Once such a necessity is accepted it is natural to consider the possibility of making the pinion from case-hardened steel. The surface stress capacity of such steel is very much higher even than that of 100-ton direct-hardened steel and so a case-hardened steel pinion has a load capacity greater than that of any wheel that can be meshed with it unless the wheel also is made from case-hardened steel.

Flame-hardening of teeth is successfully carried out on gears of medium and small sizes but does not seem to have been applied to large high-speed marine gears. It has the advantage of providing tooth flanks of harder material than can easily be cut with precision without any need for profile-grinding.

Induction hardening offers the same possibility in the future, but except for small gears, it is at present far less advanced than is flame-hardening.

A difficulty that may arise with any method of local hardening is the tendency to surface cracking. Avoidance of this demands careful selection of material and of heat treatment; air-quenching is preferable for this reason to the water-jet quenching originally used in connexion with flame-hardening.

**Mounting of Gears**

If the load on gears produced no distortion in them or in their supporting elements it would be satisfactory to carry out the cutting and mounting of them on purely geometrical principles. It the tooth loading is moderate and the general proportions such as to afford the necessary rigidity, such procedure is in practice adequate. The chief stumbling block is the rigidity of the pinion in gears of ratio higher than about 6 to 1 in one stage. In such cases, the need for restricting wheel diameter, from consideration of space occupied or of peripheral speed, may suggest the use of a pinion of which the width is large compared with its diameter. Increase in this ratio causes increase in the ratio of maximum local surface stress to mean surface stress on the tooth flanks. No conveniently useful procedure appears to exist for calculating the second of these ratios from the first, but practical results indicate that values of the ratio (total working facewidth/root diameter) up to about 2·5 are satisfactory with nominal stresses in accordance with British Standard practice. Values over 3 are likely to produce excessive stresses in the teeth at the end nearer to the driving extension of the pinion shaft.

If a central bearing is applied between the right-hand and left-hand helices, facewidth/diameter ratios up to 4 are permissible.

It is clear, however that if the maximum load is to be transmitted within given limitations of facewidth and diameter, departures from the fundamental geometrical form must be made in order to offset the effects of load distortion. Any method of predeterming such modifications will need confirming by very careful testing; even then, its application in any particular case may be nullified by minute errors of setting of the bearings or by the development of bearing wear. The use of a running-in oil for the first stages in the life of the gear would seem to be a practical means of correcting any small errors that produce appreciable departures from uniformity of load distribution.

If this is not done, examination of the distribution of the contact areas on the gear teeth after running for a short time under full load may suggest that a re-adjustment of the bearing alignment could improve matters.

**Form of Gear Wheels**

Where, as in much land practice, weight reduction is not an important consideration, the usual form of a high-speed gear wheel is that of a steel rim shrunk on to a substantial cast-iron centre, although for peripheral speeds over about 8,000 f.p.m. cast steel is preferred because of the centrifugal tensile stresses.

For many marine applications the cast steel centre is suitable, but where the utmost weight reduction is necessary, wheel centres made from steel plate are favoured because smaller thicknesses of metal may be used than are practicable in large castings. Such constructions tend to be expensive firstly because the parts are secured together by many fitted bolts and secondly because they require the wheelshaft to be forged with large-diameter collars. The wheel rim is similarly expensive because of its internal flanges.

Welding of rim to side-plates offers the possibility of considerable economy in material, but doubts about the distribution of the driving load around the length of the weld retards extensive application of this method of building large wheels.

Granted confidence in welding, even greater economies are possible in the case of wide wheels by eliminating the internal shaft altogether. (See Fig. 8). Here the wheel consists of two rims uniting three plates of which the outer ones are stiff enough to transmit bending moment from the steel-shafts to the rims. This scheme has been used for turbine rotors but has not as yet been applied to large gear wheels.

A principle that seems to be worth consideration is to design

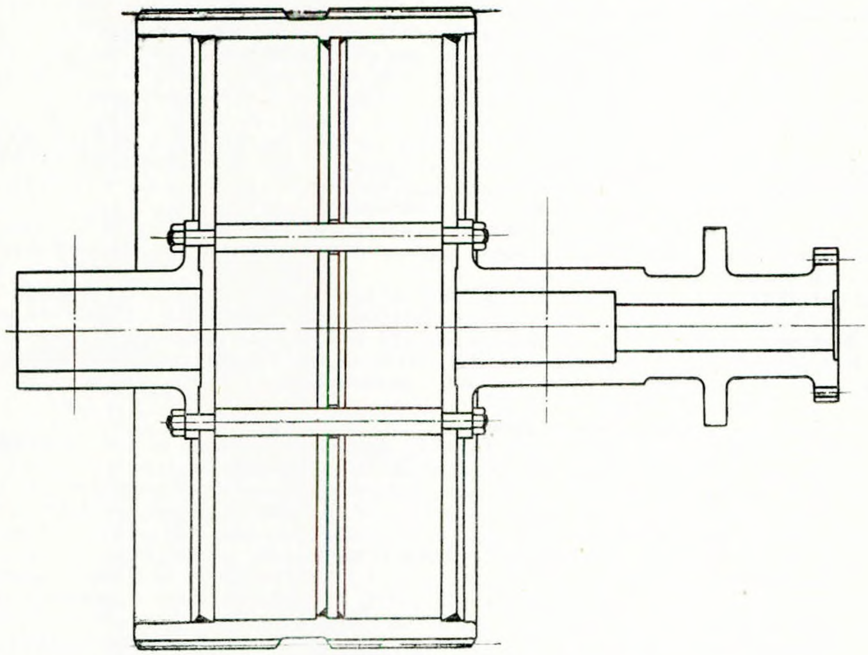


FIG. 8.—*Welded steel wheel with stub shafts*

the wheel so that distortion of the rim under load tends to match the distortion of the pinion and thus to make the distribution of the tooth load across the width of the teeth more nearly uniform than would be the case with a rigid wheel. A method of working this out in detail would be first of all to determine the displacement of the pinion teeth from their unloaded positions by the application of a uniformly distributed load and then to design the wheel rim and centre so that accurately cut teeth are correspondingly displaced by the same loading. Such a scheme, if found practicable would have the advantage of giving uniform load distribution at all loads.

The alternative procedure of cutting the pinion teeth on non-uniform helices calculated to become uniform under full load uniformly spread, can be fully effective only at one particular load.

**Lubrication of Teeth of High-speed Gears**

The purpose of any gear lubricant is to prevent metallic contact of the teeth. The lubricant of high-speed gears performs the additional duty of carrying away some of the heat generated at the surface of the teeth by lubricated sliding contact with mating teeth. The aim is to supply the gears with oil at such a rate and in such a way that the teeth are prevented from reaching undesirably high temperatures whilst excessive power loss in oil turbulence is avoided.

The last mentioned consideration rules out the oil-bath for high-speed gears and it suggests that oil should be sprayed on to the teeth immediately after they come out of mesh, reliance being placed on the natural adherence of oil and metal to hold the oil on the teeth, despite centrifugal force, while they pass round to re-enter the engagement zone.

It might be regarded as doubtful whether oil would ever penetrate into the depth of the teeth in view of the centrifugal acceleration but actual trial indicates that such penetration does take place; probably oil picked up near the tips of the teeth is spread by the tooth action over the whole flank.

A special test made to measure the power loss by applying oil sprays to the teeth firstly as they approach the point of mesh and secondly as they leave it, revealed no appreciable difference. It is obvious however that if oil is supplied at an excessively high rate the power loss is likely to be greater in the first case than in the second, as oil thrown off one gear has a good chance of being caught by the other and urged by it towards the point of mesh. On the other hand, if the feed rate is low, the first method is safer as it tends to make better use of the available oil.

A complicating circumstance is that air is ejected from the tooth spaces as the teeth enter into mesh, and is drawn inwards as the teeth disengage themselves. This tends to offset the difference between kinematic conditions affecting access of oil to the gears in approach and in recession.

**Lubricants**

Partly because the tooth loading of high-speed gears has in the past been very low compared with what is allowable at low speeds, it has not been difficult to obtain an adequate lubricating oil where choice has been unrestricted. A difficulty is, however, that turbine gears have usually to be fed with oil that is also suitable for meeting the conditions in the steam turbine, and the best compromise is rarely the most suitable oil for either duty.

The aim should be to use an oil of minimum viscosity consistent with adequate load capacity and the trend in lubricating oil development has long been in that direction. There is an increasing tendency to use oils with additives and such lubricants may become imperative for high-speed gears.

Where nominal tooth stresses are high, "extreme pressure" lubricants may be useful during the running-in period; some modern oils of this class are sufficiently stable to be used as the regular lubricant, but usually it is preferable to change over to "straight" mineral oil when satisfactory contact marking has been secured.

**Load-testing of Gears**

If two identical gear units are available, a convenient and economical method of load-testing them is by "power circulation". The wheelshafts are coupled directly together and the pinion shafts are similarly coupled whilst equal and opposite external torques are applied to them. The external torques are then removed, but the coupling prevents the stressed members from recovering their original dimensions and so the stresses remain. In particular, the tooth loading is that corresponding to the applied external torques, and rotation of the gears at any speed produces the same conditions in each gear unit as if it were transmitting a power equal to the product of speed and applied torque. The power required to rotate the gears in such conditions is simply that necessary to overcome frictional resistances, i.e. about 2 per cent. of the transmitted power for each gear unit, and if it be measured fairly accurately, it enables the mean efficiency of the two gear units to be calculated with some precision.

Actually the efficiencies of the two units are not quite the same as in one the pinion is driving the wheel and in the other the wheel is driving the pinion. On elementary theory the difference in efficiency for ratios up to (say) 10 to 1 is very small, but to achieve this in practice, the amounts of addendum-modification in the two pinions should be different and the depth of tooth should be small compared with the pitch diameter of the pinion. The first of these conditions is rarely observed and, probably for this reason, it is usual to detect some difference in performance between the "driving" and "driven" members in a power circulation system.

A very simple method of setting up power circulation within a gear unit fitted with double helical gears is to apply an end thrust to one of the shafts, the reaction being taken on the other shaft, either by a thrust bearing in the gear unit itself or by an equal and opposite external end thrust. This scheme has the disadvantage that the tooth loads on the two halves of either gear have considerable components in opposite directions and the bearing loading therefore differs from that which applies in normal operation of the gear unit.

**Appendix 1**
**Tooth Form**

In Fig. 9 the pitch circle diameters have been determined from centre distance and velocity ratio and the transverse pressure angle $\psi_t$ has been assumed, thus fixing the position of the plane of contact, whose end view is AB. The view of the plane regarded in the direction perpendicular to itself is ABCD.

The length AB is the distance intercepted on the tangent plane to the base circles by the tip circles of the two gears. If the depth of tooth is small compared with the radii of the pitch circles the arcs EA and FB depart but little from straight lines and so

$$AB = EF \operatorname{cosec} \psi_t \text{ very nearly.}$$

On the plane of contact the lines PQ and ST inclined at $\sigma_o$ (the base helix angle) to BC are lines of contact between adjacent pairs of teeth. The perpendicular distance between PQ and ST is the normal base pitch $p_{no}$; the distance PR, measured along AB is the base pitch $p_o$.

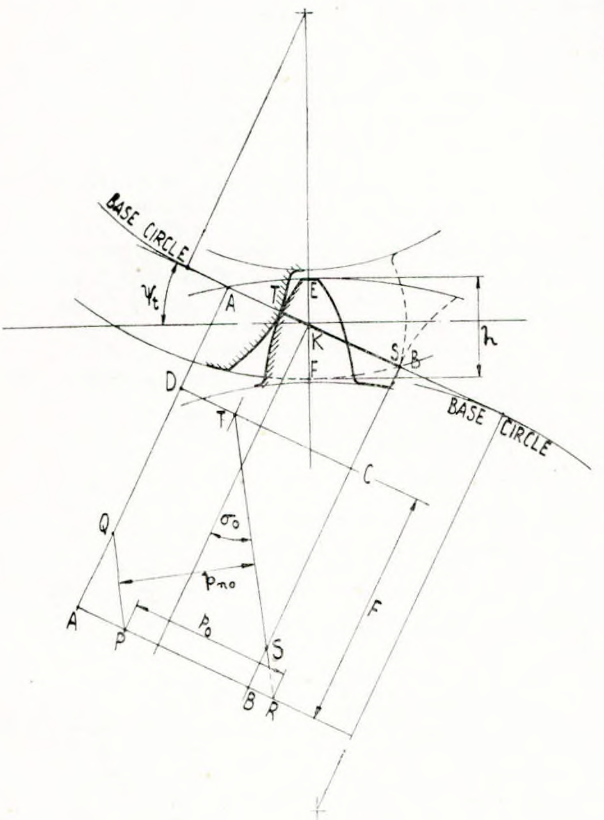



FIG. 9.—*Contact lines on helical gear teeth*

The total length of the lines of contact is PQ + ST

$PQ = PA \operatorname{cosec} \sigma_o$

$ST = CS \sec \sigma_o = (BC - BS) \sec \sigma_o = F \sec \sigma_o - BS \sec \sigma_o$

where $F$ = facewidth

$BS = BR \cot \sigma_o = (PA + PR - AB) \cot \sigma_o$

Hence

$PQ + ST$

$= PA \operatorname{cosec} \sigma_o + F \sec \sigma_o - (PA + PR - AB) \cot \sigma_o \sec \sigma_o$

$= (AB - p_o) \operatorname{cosec} \sigma_o + F \sec \sigma_o$

If $F = p_o \cot \sigma_o$, then $F \sec \sigma_o = p_o \operatorname{cosec} \sigma_o$

and $PQ + ST = AB \operatorname{cosec} \sigma_o$ ..................................................(1)

and this is independent of the phase of engagement. For values of F other than multiples of $p_o \cot \sigma_o$, the total length of contact-line fluctuates with change in phase of engagement. As a general average, the total length of contact line is proportional to the facewidth and so for any value of F it may be taken as

$$AB \operatorname{cosec} \sigma_o \frac{F}{p_o \cot \sigma_o}$$

$$= \frac{F\ AB}{p_o \cos \sigma_o}$$

$$= F \frac{EF \operatorname{cosec} \psi_t}{p_o \cos \sigma_o}$$

But $p_o \cos \sigma_o = p_{no} = p_n \cos\psi_n$ ........................................ (2)

Hence total length of line of contact $= \dfrac{Fh}{p_n} \dfrac{\operatorname{cosec} \psi_t}{\cos \psi_n}$ .........(3)

where h = working depth of tooth.

The radius of curvature of the surface of an involute helicoid at any point P (Fig. 10) is the length of the perpendicular PS to the surface at P. This perpendicular lies in a plane that touches the base cylinder in QS and the rolling of this plane on the base cylinder generates the involute helicoid surface with the straight line PQ

From the triangles OPR and PRS it may be seen that

$$PS = r \sin \psi_t \sec \sigma_o$$

where r is the distance of the point P from the axis of the gear. The radius of curvature thus varies between different points on the line of contact with the mating gear. For purposes of comparison, the radius of the pitch circle may be used as the mean value for all points on the line and the corresponding mean value for the mating gear-tooth is then $\sin \psi_t \sec \sigma_o$ multiplied by the pitch radius.

On the assumption that for a given surface stress, the allowable line loading is proportional to the relative radius of curvature of the contacting surfaces, it is in this case proportional to $\sin \psi_t \sec\sigma_o$ since the pitch radii are fixed.

Hence the allowable total load on the line of contact is proportional to $\dfrac{Fh}{p_n} \dfrac{\sec \sigma_o}{\cos \psi_n}$ ..........................................................(4

Since this load acts in the direction perpendicular to the line of contact, and in the tangent plane to the base circle, the distance of its line of action from the axis of the gear is the base radius $r_o$ and its component perpendicular to the axis is its magnitude multiplied by $\cos \sigma_o$.

Hence the allowable torque is proportional to

$$\frac{Fh \sec \sigma_o}{p_n \cos \psi_n} r_o \cos \sigma_o \quad \text{..................................................(5)}$$

Now $r_o = r \cos \psi_t$ and r is fixed

Hence allowable torque $\alpha \dfrac{Fh}{p_n} \dfrac{\cos \psi_t}{\cos \psi_n}$

$$\alpha \frac{Fh}{p_n} \frac{\sec \psi_n}{\sqrt{(1 + \tan^2 \psi_n \sec^2 \sigma)}} \quad \text{.........(6)}$$

The variation of the last factor over the range of angles likely to be employed is indicated in the following table

| | $\sigma$ | |
|---|---|---|
| $\psi_n$ | 20° | 30° |
| 14½° | 0·996 | 0·990 |
| 20° | 0·992 | 0·982 |

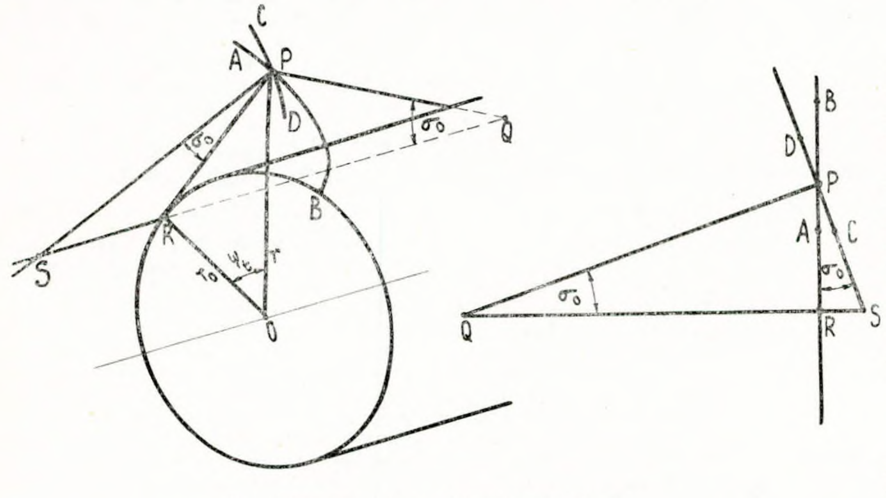



FIG. 10.—*Curvature of involute helicoid*

This variation is seen to be negligible and so the load capacity for given centre distance, facewidth and velocity ratio is proportional to the ratio of working depth of tooth to normal pitch and is hardly affected by changes in helix angle to normal pressure angle.

For any selected ratio of tooth-depth to pitch there are, however, upper limits of normal pressure angle and helix angle dictated by the need for avoiding excessively narrow tooth-tips. The normal pressure angle has a lower limit because of the need to avoid undercutting at the roots of the teeth of the pinion.

To minimize the sliding velocities at the tips of the teeth, the depth of tooth should be small and therefore, if the pitch/depth ratio has been fixed, the pitch should be small. Consideration of bending stress sets a lower limit.

## Appendix II
### Addendum Modification

The primary need for inequality between the addendum and dedendum of an involute gear tooth is associated with avoidance of the unsatisfactory tooth shape that is produced when the root circle lies too far inside the base circle. In helical gears of about 20 deg. pressure angle and with not fewer than about 30 teeth in the pinion this "interference" condition is not approached, and so there is no need on this account to adopt "addendum modification".

In high-speed gears a more important consideration is the avoidance of unnecessarily high rubbing speeds between the teeth. The rubbing speed at any point of contact of helical gears is equal to the product of the sum of the angular velocities of the gears and the distance of the point of contact from the common tangent to the pitch cylinders. The rubbing speed is greatest for contact at the tips of the two gears and increase in the addendum of either gear increases the maximum rubbing speed at its tooth-tips and (if the working depth is kept constant) reduces the maximum rubbing speed at the tooth-tips of the other gear.

It would seem desirable to keep down the maximum rubbing speed as much as possible. This is achieved by equalizing the tip rubbing speeds of wheel and pinion. Even in the extreme case of a wheel of infinite diameter meshing with a 30-tooth pinion, equality of maximum rubbing speeds requires the pinion to have

$$\frac{\text{addendum}}{\text{dedendum}} = \frac{0{\cdot}53}{0{\cdot}47}$$

and this means that there is no object, from this point of view in using such extreme addendum-modification as is found in the "all-addendum" type of tooth.

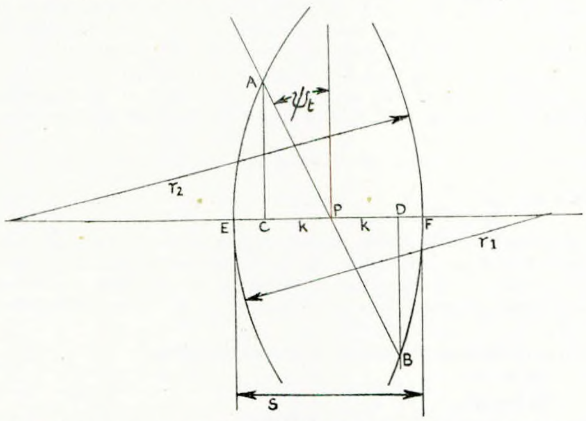



FIG. 11.—*Determination of condition for equality of rubbing speeds at tips of pinion and wheel teeth*

In Fig. 11, P is the pitch-point of engagement of two gears with tip radii $r_1$ and $r_2$ and AB is the path of contact. Equality of rubbing speeds at A and B requires that AP = PB. Let PC = PD = k

Then $AC = k \cot \psi_t$

and $$EC = \frac{AC^2}{2r_1} = \frac{k^2 \cot^2 \psi_t}{2r_1} \text{ (very nearly)}$$

Similarly $$DF = \frac{k^2 \cot^2 \psi_t}{2r_2}$$

The working depth EF is denoted by s and so

$$s = EC + CD + DF = 2k + \frac{k^2 \cot^2 \psi_t}{2}\left(\frac{1}{r_1} + \frac{1}{r_2}\right)$$

whence

$$k = \frac{-2 \pm \sqrt{\left[4 + 2s \cot^2 \psi_t \left(\frac{1}{r_1} + \frac{1}{r_2}\right)\right]}}{\cot^2 \psi_t\left(\frac{1}{r_1} + \frac{1}{r_2}\right)} \quad \text{.........(7)}$$

Assuming unit module in the transverse section, for 30 teeth $r_1 = 16$ (approximately) and for a mating gear with a large number of teeth, $r_2 = \infty$. Also $s = 2$.

Taking these in conjunction with $\psi_t = 20$ deg.

$$k = \frac{-2 \pm \sqrt{\left[4 + 30{\cdot}28\left(\frac{1}{32} + 0\right)\right]}}{7{\cdot}57\left(\frac{1}{32} + 0\right)}$$

$$= 0{\cdot}94$$

$$EC = \frac{0{\cdot}94^2 \times 7{\cdot}57}{2 \times 32} = 0{\cdot}105$$

$$DF = 0$$

Addendum of pinion = PE = PC + EC = 1·045

Addendum of wheel = PF = PD + DF = PC + 0 = 0·94

$$\frac{\text{Addendum of pinion}}{\text{Working depth of tooth}} = \frac{1{\cdot}045}{1{\cdot}985} = 0{\cdot}528 \quad \text{..............................(8)}$$

$$\frac{\text{Addendum of wheel}}{\text{Working depth of tooth}} = 1 - 0{\cdot}528 = 0{\cdot}472$$

There is evidence that failure of a gear lubricant occurs when the product of surface stress and rubbing speed reaches a particular value. If this be accepted, the addendum modification in gears of a particular ratio should be adjusted so as to equalize this product in the two limiting places, i.e. the tip of the pinion tooth and the tip of the wheel tooth.

Assuming that the local surface stress is inversely proportional to the radius of curvature of the profile of the pinion tooth (that of the wheel tooth being much larger), it can be shown that for equality of product of surface stress and rubbing speed at tip and root

$$\frac{\text{Addendum of pinion}}{\text{Working depth of tooth}} = \tfrac{1}{2} + \sqrt{\left[\left(\frac{r \sin^2 \psi_t}{2s}\right)^2 + \tfrac{1}{4}\right]} - \frac{r \sin_1 \psi}{2s}$$

where $r$ = pitch radius of pinion

and $\psi_t$ = transverse pressure angle

or $$\frac{\text{addendum of pinion}}{\text{working depth of tooth}} = \tfrac{1}{2} + \sqrt{[(0{\cdot}0218t)^2 + 0{\cdot}25]} - 0{\cdot}0218t$$

where $t$ = number of teeth in pinion

If $s = 0{\cdot}636\ p_n$

$\psi_n = 20°$

$\sigma = 30°$

and $t = 30$,

$$\frac{\text{addendum of pinion}}{\text{working depth of tooth}} = 0.67$$

For larger values of t this ratio becomes nearer to 0·5. From this point of view the highest ratio of addendum to dedendum that may be expected to be useful in practice is therefore $\frac{0{\cdot}67}{0{\cdot}33}$

## Appendix III

### Overlapping Effect of Hob Cuts

In cutting a helical gear tooth, each cutting edge of the hob produces a large number of surfaces each approximating to a portion of the surface of an ellipsoid and the final surface of each tooth flank is the result of the intersection of such ellipsoidal surfaces, whose centres lie on a surface of the correct theoretical form and

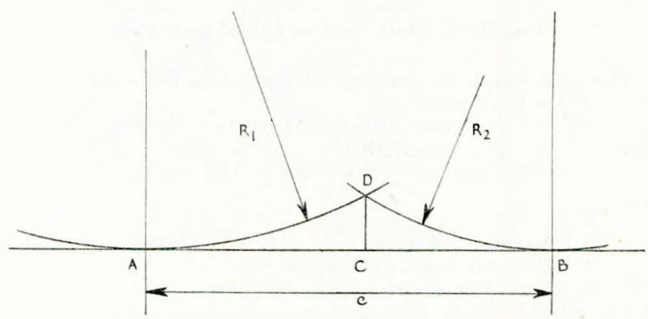



FIG. 12.—*Determination of height of wave between adjacent hob cuts*

are the "low-spots" of the tooth. The heights of the intervening high-spots are determined by the radii of curvature of the ellipsoidal surfaces in planes containing the low-spots and lying perpendicular to the nominal surface, and the distances between adjacent low-spots.

Thus Fig. 12 represents the section of a tooth profile on a plane perpendicular to the nominal surface and containing two low-spots A and B at which the radii of curvature are $R_1$ and $R_2$.

Then $$CD = \frac{AC^2}{2R_1} \text{ and also } CD = \frac{BC^2}{2R_2}$$

Hence $$AB = AC + BC = (\sqrt{R_1} + \sqrt{R_2})\sqrt{2CD}$$

and so $$CD = \frac{c^2}{2} \cdot \frac{1}{(\sqrt{R_1} + \sqrt{R_2})^2} \quad \text{..............................(9)}$$

The radius of curvature of a cut-surface in the plane containing the cutting edge is infinite because the edge is straight but, relatively to the nominal surface it has the radius of curvature of that surface. This is of the order of 0·4 times the pitch radius of the gear, assuming it to have a normal pressure angle of 20 deg. and helix angle of 30 deg.

The radius of curvature of the cut surface in the perpendicular plane is the pitch radius of the hob multiplied by the cosecant of the normal pressure angle. In practice this is about 10 inches.

For a gear 100 inches diameter, the two radii of curvature are therefore 20 inches and 10 inches and in any plane containing the low-spots the radii of curvature of the two cuts concerned must lie between 10 and 20 inches.

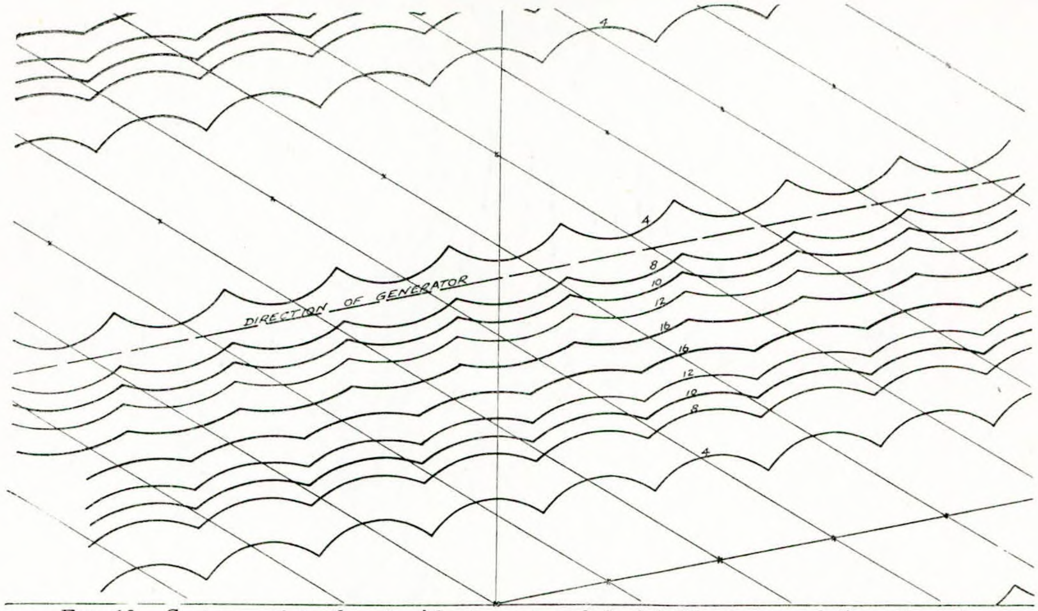



FIG. 13.—*Contours of tooth cut without creep and feed 0·04in. per revolution of work. Contour heights are in hundred thousandths of an inch*

Inserting the smaller value for both $R_1$ and $R_2$ in (9) and assuming $c = 0{\cdot}05$in.,

$$CD = \frac{0{\cdot}05^2}{8 \times 10} = 0{\cdot}00003 \text{ in.}$$

This is a reasonably small variation from the nominal surface but it varies as the square of the spacing of the cuts, and so reduction in feed-rate may considerably reduce the height variation along lines parallel to the axis of the gear.

The distribution of "low-cuts" caused by main worm error in a non-creep machine is indicated in Fig. 13 from which it will be seen that there are ridges of high-spots similar to AB, whose direction is that of the line of contact of the gear and its mate. The maximum height of the high-spots over low-spots is 16 hundred thousandths of an inch.

For a worm creep fraction of 0·46, the distribution is changed to that shown in Fig. 14. There the tendency to ridges is eliminated and the height of the high-spots over the low-spots is only 8 as against 16 for the case of Fig. 13.

Change in feed-rate from 0·04 to 0·08 produces the pattern shown in Fig. 15. This differs considerably from that of Fig. 14 and the maximum height is 30 instead of 8.

This comparison brings out the fact that, with creep-cutting, a change in feed-rate may have a much greater effect on the finish than would the same change with non-creep cutting.

## Appendix IV

### Crossed Axis Shaving

Two helical gears with equal and opposite helix angles, when meshed together on parallel axes, make contact along straight line generators of their tooth profiles.

If the helix angles are not equal and opposite the gears may be meshed together on axes inclined to each other at an angle equal to the difference in helix angle. Straight line generators of mating teeth no longer coincide but are "crossed" at an angle equal to about one-third of the axis angle, assuming that to be about 15 deg. This is the condition of cutter and work in crossed axis shaving.

In a helical gear of 30 deg. helix angle and 20 deg. normal pressure angle, the straight line generator (AB in Fig. 16) is inclined at about 11 deg. to the tip of the tooth. The dotted line CD represents the straight line generator of the mating tooth; the two teeth touch at P.

Owing to the smallness of the angle CPA, the distance between any point on either tooth profile within the areas PAC and PBD, and the nearest point on the mating profile is very small. A small

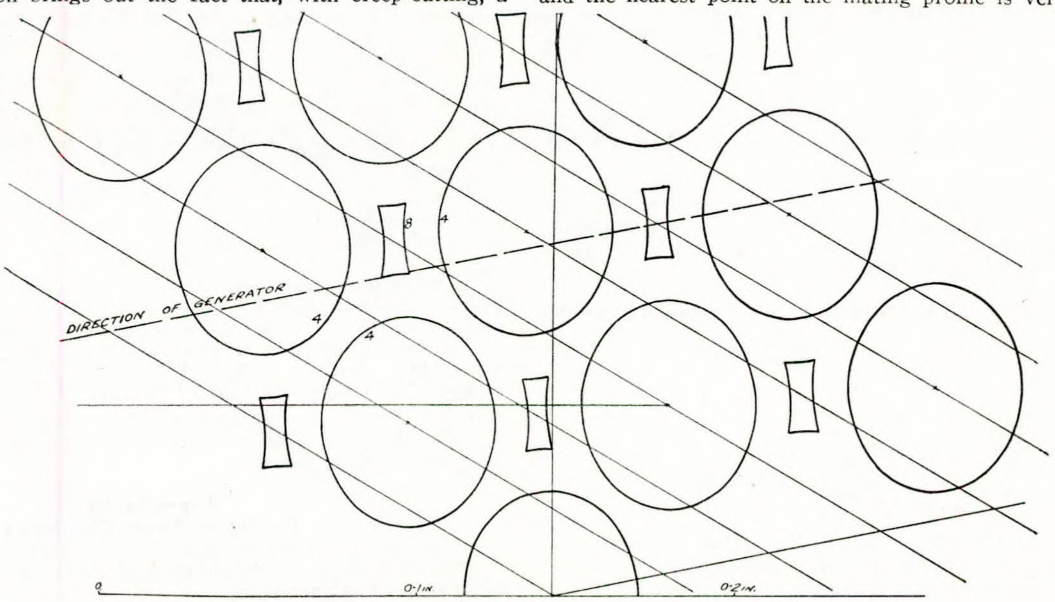



FIG. 14.—*Contours of tooth cut with worm creep fraction of 0·46 and feed of 0·04in. per revolution of work. Contour heights are in hundred thousandths of an inch*

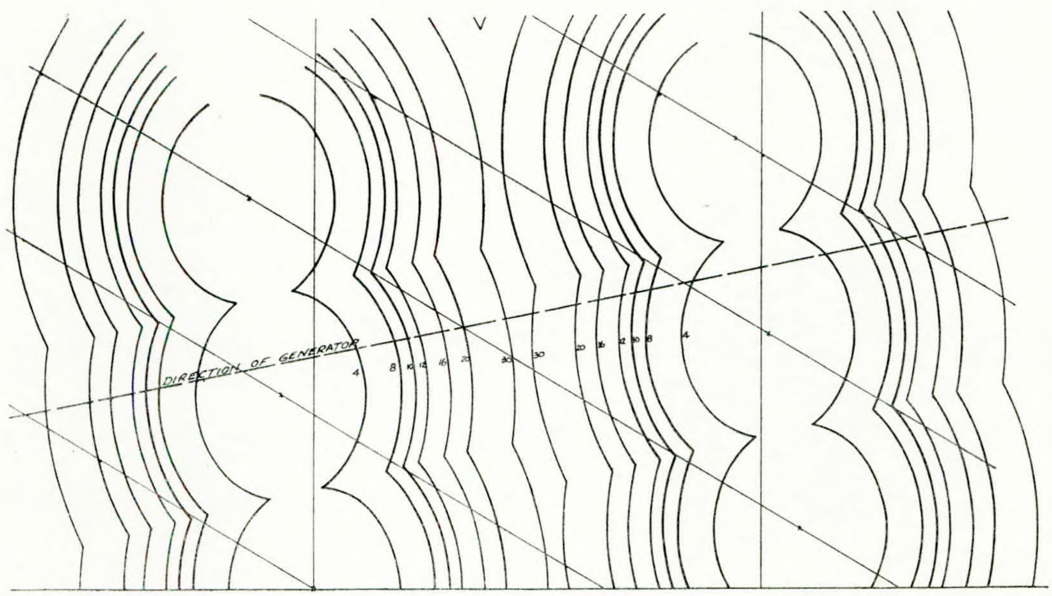



FIG. 15.—*Contours of tooth cut with worm creep fraction of 0·46 and feed of 0·08in. per revolution of work. Contour heights are in hundred thousandths of an inch.*

compression of the surfaces by the load at P produces a contact area in the form of an ellipse centred at P with its long axis bisecting the angle PAC and its other axis very much shorter.

Rotation of the mating gears causes the point of contact to move along a nearly straight line EF inclined at about 30 deg. to the tip

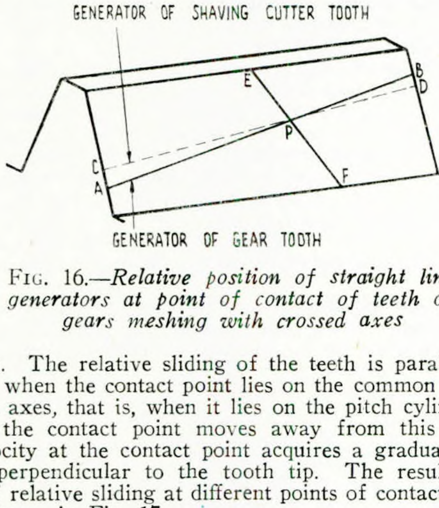



FIG. 16.—*Relative position of straight line generators at point of contact of teeth of gears meshing with crossed axes*

of the tooth. The relative sliding of the teeth is parallel to the tip of the tooth when the contact point lies on the common perpendicular to the shaft axes, that is, when it lies on the pitch cylinders of both gears. As the contact point moves away from this position, the relative velocity at the contact point acquires a gradually increasing component perpendicular to the tooth tip. The result is that the directions of relative sliding at different points of contact are approximately as shown in Fig. 17.

In subsequent passages of the tooth through the contact zone with the shaver, shaving takes place on similar areas centred on a line parallel to AE and separated from it by a distance equal to the feed motion of the cutter per revolution of the work, i.e. about 0·050in. The overlapping of such shaved areas results in a smooth finish on the tooth.

The ratio between the cutting speed in shaving and the velocity

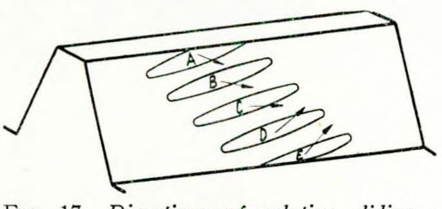



FIG. 17.—*Directions of relative sliding at various points of contact of gear and shaving cutter. Arrows show directions of relative sliding on contact bands in different successive positions*

of the imaginary common rack measured perpendicular to the length of its teeth is the difference between the tangents of the helix angles must be rotated about 90 times as fast for shaving as for hobbing of gear and cutter and is usually of the order of 0·2.

The corresponding ratio, derived in exactly the same way, for hobbing is about 18. Consequently for equal cutting speeds the work and the overlapping of the long shaving cuts in the direction of the tooth helix produces a much smoother finish than is possible by hobbing at any economical feed rate. Shaving is thus equivalent to a very fast fine finishing cut by hobbing.

## Appendix V
## Power Loss in Gear Teeth

The power loss in friction between the teeth of steel helical gears with not fewer than about 30 teeth in the pinion is of the order of 0·5 per cent. of the transmitted power.

Assuming this power is to be dissipated in the lubricating oil with a temperature rise of (say) 20 deg. F., the weight of oil required per second per transmitted horse-power is

$$\frac{0{\cdot}5}{100} \times \frac{550}{778} \times \frac{1}{20 \times 0{\cdot}5} = \frac{1}{2825} \text{ lb.}$$

It must be assumed that all the oil sprayed onto the teeth is accelerated to their tip speed (say V.f.p.s.) and so the change of momentum of oil per second per transmitted horse-power is

$\frac{1}{2825} \times \frac{V}{32{\cdot}2}$ This is the force, in lb. tangential to the tip circle. required to accelerate the oil, and the corresponding power loss is

$$\frac{1}{2825} \times \frac{V}{32{\cdot}2} \times \frac{V}{550} = \left(\frac{V}{7070}\right)^2 \text{ per transmitted H.P.}$$

or $\left(\frac{V}{707}\right)^2$ per cent of transmitted power.

For this to be equal to 0·5 per cent.

$V = 707 \times 0{\cdot}707 = 500$ f.p.s.

or 30,000 f.p.m.

This is distinctly higher than tip speeds for large first reduction gears and shows that as a general rule, the sum of tooth friction loss and oil spray loss is less than 1 per cent. of the transmitted power.

The power required to pump the oil at (say) 10lb. per sq. in. is

$$\frac{10 \times 144}{550} \times \left(\frac{1}{2852} \times \frac{1}{62.5 \times 0{\cdot}9}\right) = \frac{1}{60000} \text{ H.P per transmitted H.P.}$$

or about 0·002 per cent.

This loss is negligible even with a generous proportional allowance for pumping losses.

## Appendix VI
## Testing by Power Circulation

Two pairs of similar gears may be load-tested by the power-circulation method, which, although simple in principle has a number of interesting variations.

The most commonly used method is indicated in Fig. 18. To load the gears, the coupling bolts are removed and the two halves

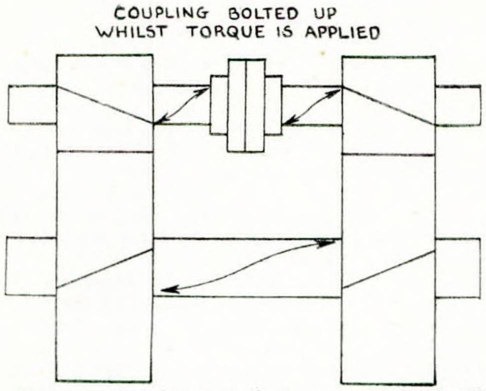



FIG. 18.—*Power circulation in helical gears pre-loaded by torque*

of the coupling subjected to equal and opposite torques of the magnitude desired on the pinionshaft. Whilst the external torques are applied, the halves of the coupling are bolted or clamped together. On removal of the external torques, the torque reaction between the pinionshafts is transmitted through the coupling, and the assembly may then be run under the desired load. Any wear of gear teeth or bearings, or any yielding of loaded members reduces the gear loading. To determine whether such release of load has taken place during a run, the loading procedure should be repeated after the run in order to ascertain whether the original torque brings the coupling halves to the original relative position.

With the helices handed as shown, the axial thrust on the gears are balanced by transmission along the shafts. The directions of

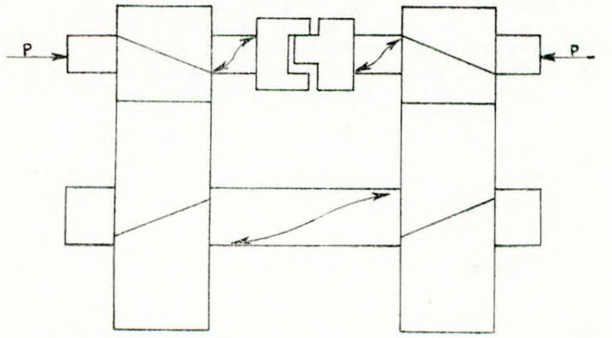



FIG. 19.—*Power circulation in helical gears pre-loaded by axial thrusts applied to wheel shafts*
*Torque set up by application of equal and opposite thrusts P to pinion shafts*

the torques are shown in Fig. 18 by the helical positions of lines that are straight when the shafts are unloaded.

In Fig. 19 the pinionshafts are connected by a coupling that transmits full torque with negligible axial restraint. The gears are loaded by applying equal and opposite axial thrusts P to the pinion shafts. The torque on each gear (neglecting frictional effects) is then

$$\left(\frac{P}{2\pi}\right) \times \text{Lead}$$

In Fig. 20, the arrangement of Fig. 18 is repeated except that the coupling connects the wheelshafts and the axial loads are applied

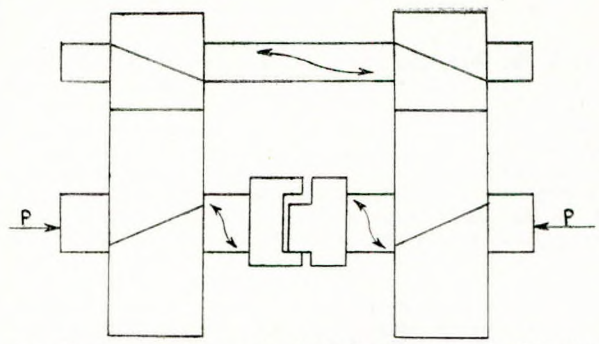



FIG. 20.—*Power circulation in helical gears pre-loaded by axial thrusts applied to wheel shafts*
*Torque set up by application of equal and opposite thrusts P to wheel shafts*

to them. Again the torque on each gear is $\left(\frac{P}{2\pi}\right) \times$ Lead.

In Fig. 21 gears with helices of opposite hand are rigidly coupled together and equal and opposite axial loads P are applied one to the pinionshaft and the other to the wheelshaft. In this case the torque on each gear is $\frac{1}{2}\left(\frac{P}{2\pi}\right) \times$ Lead.

In none of these arrangements is any axial load exerted on bearings in the gear unit.

In all cases except that of Fig. 18, the load can be changed whilst the assembly is running. This is of particular advantage because the start from rest can be made in the unloaded condition,

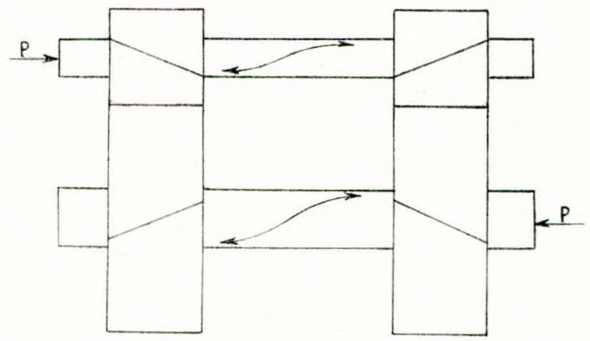



FIG. 21.—*Power circulation in helical gears pre-loaded by axial thrusts applied one to pinion shaft and the other to wheel shaft*
*Torque set up by application of equal and opposite thrusts P one to pinion shaft and the other to wheel shaft*

and the heavy loading associated with static friction is then avoided.

The scheme of Fig. 21 may be applied to a single gear unit fitted with double helical gears but the directions of the bearing loads are then different from those applying during normal operation of the unit.

**Acknowledgments**

The author makes grateful acknowledgment to Messrs. David Brown and Sons, Ltd. for permission to publish this paper and to Mr. S. Fearns and other members of the staff of Messrs. David Brown and Sons, Ltd. for assistance in preparing it.

## Discussion

**Mr. G. W. Richmond** (Visitor) said that in general, the turbine designer was striving to produce turbines of higher speeds and higher efficiency, and also of lower weight. The propeller designer, on the other hand, did not want speeds to go up, because of consequent loss of efficiency. The marine gear designer came between the two and had to produce transmission units of much higher reduction ratios to suit turbines running at much higher speeds and propeller shafts turning at the same or lower speeds than before. In the future, therefore, it might be necessary to consider gear ratios of the order of 25 to 1 or 30 to 1, as being more likely than the lower figure of 10 to 1 referred to in the paper. There was no doubt that the critical attention now being given to marine gearing would enable developments to proceed on sound lines, with benefit both to specialized designs for the Navy and also to the Mercantile Marine.

Inspection of the distribution of tooth marking on marine gears which had been in service for years were made from time to time, and he was certain from his own observation that the proportion of the tooth surface which was actually effective had been in some cases relatively small. An immediate development, therefore, to get increased loading capacity, was to use the surface more efficiently. In that way it was possible to transmit the same power with the same intensity of loading, on a much shorter face length. He thought that the illustrations which the author had shown of crests and valleys on hobbed gears were indicative of the sort of load concentration which was actually applied. The gear designer, with his text book, visualized the line of contact as representing the actual contact between two mating gears; but in fact, of course, it could not be so unless the mating surfaces were geometrically perfect; any departure from that line of contact must entail a serious concentration of loading. It seemed to him, therefore, that the immediate problem was primarily one for the gearing manufacturer.

A closer approximation to the conditions of line contact was the obvious first step, and that that development could save weight, and could save wear and tear both of machinery and of personnel, had already been demonstrated by the considerable improvement in smoothness of running and reduction of noise in some recent large gears to which post-hobbing processes had been applied. If it was desirable to run gears to bed them in, or to use running-in oil, he thought that the marine engineer would much rather in most cases pay the manufacturer a little more to bed them in before installation. The tests on load shown by the author presented the manufacturer with an excellent method of doing so. Probably the marine engineer would not recognize those illustrations as looking very much like his own marine gears, but they were simply diagrammatic, and the power circuit could be readily applied to either single- or double-reduction gears, particularly if the arrangement was the common two-turbine design.

In the introduction to the paper the author put variations in tooth-form first. Personally, he thought that the second factor mentioned, namely higher standards of accuracy in manufacture, was far and away the most important. Improvements which were now going on amounted in fact to major engineering developments, inasmuch as errors which were formerly measured in thousandths of an inch were now being reduced to tenths of thousandths; and even a few tenths of thousandths of an inch on gear teeth could be responsible for a good deal of noise.

In relation to tooth-form, he did not think that the increased depth to pitch ratio should be regarded as an experiment. It had been in use in the Mercantile Marine for a number of years, and was in fact being used now for Naval gears. The proportion recommended by the author was similar to the one which had been adopted for the latter.

Sliding velocity seemed to be regarded by the author as a criterion for surface loading capacity, but there again he saw nothing to fear, because in existing double reduction gears pitch line speeds varying from 30ft. per sec. up to 300ft. per sec. were employed, sometimes in the same gearbox, and in general the defects which arose were more often on the low-speed than on the high-speed gears.

The next point on which he wished to touch was length to diameter ratio of pinions. In some gears which he had seen which were excessively long—there were reasons for that, of course—gears with a pinion length to diameter ratio of 3 or more, there had been clear evidence that the after end of the forward helix had not been carrying any load at all, and it was obvious that that portion could be removed without any disadvantages whatever to performance; in fact, a better performance would be expected because of the reduction in the load concentration at the other end. Some investigations had been carried out in America and published in the Journal of the American Society of Mechanical Engineers, which endeavoured to assess the degree of concentration of loading associated with this length to diameter ratio. It amused him a little because the technicians in the shops had tackled the problem mathematically, whereas one of the professors at the university had tackled the same problem graphically. The actual concentration, of course, must depend on a number of factors, but from the information published it would appear that the optimum length to diameter ratio would be about 2 for a double helical gear, and anything above or below that ratio would mean an increased maximum loading. Any increase in length beyond the optimum proportion would give a higher tooth maximum loading intensity and not fulfil any useful purpose.

Appendix I set out the geometry for assessing the allowable torque on the basis of tooth surface stress, and the table seemed to suggest that although the differences as shown there were small—they would be larger if the range were extended—the straight spur gear would be the best tooth-form to adopt. He felt that there was something there which required a little further exploration, and reference to Fig. 11 showed that an explanation could be found in the approximation made for the length of the line of contact. EF multiplied by the cosecant of the angle gave the diagonal line between two parallel straight lines whereas the tip circles of the teeth restricted the actual line of contact by varying amounts; in Fig 11 by about 25 per cent. He had made some investigations some time ago on a similar basis, and found that the optimum conditions for tooth form of helical gears were given if the pressure angle in the transverse plane was about 20 deg.

In looking to the future, particularly for highly loaded Naval gears, he thought that they would now be running into territory which had not previously been explored, and the use of case-hardened and ground gears was of particular interest, as might be imagined, in view of the fact that it offered so much scope for increasing the intensity of loading; but it introduced a difficulty, because with such a high surface capacity it might well be that the root strength would become the criterion. In that case, it would be necessary to consider the tooth proportions very carefully to ensure that there was not likely to be any failure at the root. He thought that most marine engineers would prefer pitting or scuffing marks on the teeth of their gears rather than broken teeth.

There was one point about the shaving process not mentioned by the author. One of the chief advantages was that the shaving cutter meshed with the gear and was driven by it. It was possible, there, fore, to eliminate altogether the errors arising from the hobbing machine.

**Mr. C. Timms** (Visitor) said that with reference to the production of accurate gear hobbing machines, the author quite rightly emphasized the need for extreme accuracy in pitch of the lead screw controlling the hob saddle traverse. The facilities available in this country for the production of accurate lead screws up to 10ft. in length were extremely limited and it invariably happened that the final process of correction involved transporting them two or three hundred miles to the National Physical Laboratory. Some relaxation in these fine limits of accuracy could be permitted if gear hobbing machines were fitted with a corrector bar mechanism in which the linear errors in the hob saddle traverse were automatically compensated by a suitably formed corrector bar. Such devices were frequently used for controlling the saddle motion of master screw-cutting lathes and he saw no difficulty in similar devices being incorporated in gear hobbing machines. By adjustment of the corrector bar the effect of any progressive wear of the lead screw could be eliminated.

In addition, as an alternative to selective shaving, it would be possible by an appropriate tilt of the corrector bar to apply a correction to the tooth helix angle and so eliminate or offset the effects of load distortion which was referred to on p. 111. It would be of interest to have the author's reactions to this proposal.

The primary object of gear shaving was to improve the quality of finish of the tooth surfaces and from a measurement viewpoint this had the effect of reducing the random variations in measurement which invariably occurred with hob-cut gears. The presence of local surface irregularities completely masked the magnitude of the real indexing errors and in view of this it was impossible to say whether shaving was really effective in reducing tooth-to-tooth errors.

It was agreed that tip or root easing on gear teeth could be secured by appropriate modification of the cutter profile, but an additional modification must also be applied to take account of the change in the cutting action which occurred when shaving gears of different diameter. Such factors as the relative curvature of the gear teeth and the relative sliding between cutter and work must have an important bearing on the amount of metal removed by the shaving cutter. These factors did not depend on whether the gear was shaved by a dual flank or single flank cutter. In view of this, it was difficult to reconcile the author's remarks that the same cutter would produce the desired profile on pinions and wheels of the same normal base pitch.

Finally with reference to Appendix III since the overlapping effect of hob cuts varied as the square of the feed and inversely as the hob radius the use of hobs of larger diameter would permit an increase in the rate of feed without impairing the finish of the tooth surface. The consequent reduction in the machining time would be considerable when cutting large reduction wheels. It was understood that giant hobs had been employed by the firm of Klingelnberg in Germany and had the author any experience in this particular field. It was appreciated that the gear machine would have to be suitably stiffened for this purpose and it might require a new design of hob slide.

**Mr. W. J. Ferguson** (Member) said it might be inferred that a silent gear was not necessarily a good one, all the emphasis being rightly placed upon the accuracy of manufacture. This indicated one of the main difficulties in applying rules for gearing. Everyone was conscious of the fact that the rules at present in force were not what one could wish them to be since no account was taken of the accuracy of the finished gear. It was felt that the time was not yet ripe for the accurate control of the machines but that progress was being made, and that improved accuracy of gear-cutting machines would lead in time to a very useful increase in permissible load.

The author mentioned the advantage of case-hardened gears, and stressed the difficulty regarding distortion. He would like to ask whether the grinding difficulty had been overcome as he had had experience of grinding cracks ruining gears. The author had not mentioned the use of cast steel; was that to be taken as an indication that the author's opinion was that for important gears such as were used in marine applications he would not advocate the use of steel castings?

**Mr. G. Hallewell** (Member) mentioned three points on which he would like the views of the author. The first, he said, was with regard to the ratio of length of tooth to pitch. He had seen various ratios used and personally he had always rather favoured the stub tooth form, which was a short tooth, and in consequence, stronger than the standard form, particularly in reduction gears with a ratio of approximately 3/1. He had in mind Diesel engines operating at 600 r.p.m., developing between 300 and 600 h.p. which he agreed was rather lower than anything to which the author was referring. As an example of a tooth form of abnormal tooth length, he had seen a Junkers engine where the power was transmitted from the top and bottom crankshaft through a series of spur wheels having teeth approximately 30 per cent longer than the standard form. These gears were transmitting a high power but from a cursory examination, the teeth appeared to be of a weak design.

His second point was with regard to the advantages of the helical wheel as compared with the spur wheel. He felt that in some cases helical wheels were used where spur wheels functioned equally well. He believed that the helical tooth originated from the desire to reduce the noise arising from inaccuracies of teeth but where ground teeth were used he felt that in ratios and gears of which he had had experience, the straight tooth answered the purpose quite well.

His third point was with regard to the bearings. He had had experience with ball, roller and plain bearings for supporting the gear wheel shafts and had found that in changing from a ball or roller bearing to a plain bearing, the noise emanating from the gear box, which had not been excessive, was reduced to nearly complete silence. He had formed the opinion that a contributory factor to the reduction of noise was the fact that the diameter of the bearings was approximately equal to the pitch diameter of the pinion and oil emerging from the bearing in fairly large quantities entered the teeth on both sides of the pinion and provided a cushioning effect between the teeth.

**Mr. A. F. Ainslie** (Visitor) who remarked that his recollections went back to the early days of marine reduction gearing, said he thought the first set of marine reduction gears was produced by the author's firm at the end of the last century. Reduction gearing was applied to a 22ft. launch by Parsons in which a pinion drove two wheels with right and left hand propellers. The first cargo ship to be fitted was the "Vespasian", which made one or two voyages from England to Malta and back before the firm would proceed with any manufacture for general purposes. Parsons carried out tests on the loading of the gears by putting a lever on the end of the shaft and using weights to load up the gears to see when and where they deformed.

In the turbine driven ship "Cairnross", the reduction ratio was in the neighbourhood of 26/1. It was the only instance of so large a single reduction for a marine main drive. Afterwards double reduction was always used.

### BY CORRESPONDENCE

**Dr. T. W. F. Brown** (Member) wrote that at this time when so many lines of development in gearing were being followed up it was probably useful to put on paper a statement about the lines which were being pursued in the search for lighter, less noisy, less bulky and in consequence more heavily loaded gears. It would be clear that these desiderata were not mutually compatible and the author's statements on these points were very useful.

Under the heading "Tooth Profile Modification", it was agreed that the provision of end relief in helical gears was instrumental in reducing noise emitted at the contact frequency of the gear teeth—this incidentally being one of the strong components in noise made by gearing. The precise advantage in the form of relief shown in Fig. 4 was not understood. It appeared that full deflexion of the mating tooth occurred first at a distance $p_n$ from the end of the tooth. Did this not require a greater impulse than if the full deflexion were first experience at the end of the tooth as the tooth became stiffer the loading point was moved farther from the end?

Under "Accuracy of Gear Teeth" it was stated that errors of less than 0·0005in. might cause appreciable local stress concentrations. It was suggested that at this state of the art it was errors greater than this amount which would have to be eradicated. The statement that excessive noise was objectionable only because of its effect on human beings and that it did not necessarily imply mechanical inefficiency, or overloading was too general. It would appear better to say that noise was always accompanied by, if not caused by, some vibrational motion of the gear and that the effect of this motion with the corresponding inertia forces on the gears would cause augmentation of tooth loading. If this was true, then excessive noise was generally asociated with excessive tooth load and a silent gearbox should carry with safety a greater nominal load than a noisy one.

He noted that the author only claimed for shaving that it improved the finish of teeth. It was understood that shaving could also be used for a corrective process to remove undulations provided they were of short enough wave length. He would like the author to give an opinion as to the use of shaving to remove hobbing errors such as helical angle correction and to produce silent gears?

In the paragraph "Running-in" the author suggested the use of a running-in oil. It was suggested that in all cases gears should be run in in service before applying the full load. It was not necessary to use special running-in oil for this purpose but in all cases a period of running at lower than full load should be carried out to ensure that polishing of the teeth occurred and consequent reduction of the high spots in the mating surfaces before applying a full load.

In the paragraph on mounting gears, various nominal face width to root diameter figures were given for gear pinions without and with centre bearings. What was the author's figure for pinions in which the drive was taken off by two primary wheels in a locked train gear where bending in the pinion body should be practically absent? The author referred again to the use of running-in oil. Would it not be better to lap the pinion and wheel separately with a cast-iron lap and then to lap the mating gears together before mounting, always providing that the accuracy of mounting for lapping could be reproduced in the finished gear box.

The form of gear wheel shown in Fig. 8 did have advantages in lightness and cheapness but it was suggested that a wheel of this construction had extremely poor axial stiffness—one of the required features in modern marine gearing. The method of using this wheel to correct and even out pinion loading presupposed that pinion loading had already been measured although earlier statements in the paper suggest that the axial distribution of loading on pinions was not readily determinable.

Under the heading "Lubrication of Teeth" and also in Appendix V the author referred to the action of the gear in distributing lubricating oil. Was it not clear that the mating helical gears did in fact act as a type of pump and that the lubricating oil was not only operating radially in and out of the teeth but additionally was being pumped axially along the helices by the action of the continual movement of the line of contact across the teeth. Incidentally, Appendix V began with the assumption that the power lost in friction between the teeth of helical gears was of the order of 0·5 per cent of the transmitted power. It was clear from the efficiency of helical gears that this loss must be of a small order but it would be valuable if the author could adduce any experimental evidence in support of this value. The horse-power loss by axial pumping might be considerably larger than the loss calculated in Appendix V.

Although it was agreed that in marine use turbine gears had also to be fed with oil suitable for meeting the conditions in the steam turbine bearings and that a compromise oil was therefore used, the author omitted to point out that there was still some control of viscosity as between turbine bearings, gearing bearings and gearing sprayers by means of separate oil coolers with bypasses. This arrangement did not require duplication of the drain tanks, lubricating oil pumps, filters, etc. but did ensure that oil differing in viscosity according to the lubrication requirements could be supplied for these three duties utilizing only one grade of oil.

The paragraph on load testing of gears by means of back-to-back tests, together with Appendix VI, would appear to apply mainly to single helical gears, and did not apply to tests on say two similar gear boxes fitted with double helical gears. The single helical gears would presumably be manufactured for test purposes since they were not in general use in marine practice, especially in double reduction gearboxes because of the number of thrust blocks required. In fact one of the chief uses of back-to-back tests was to obtain performance data on actual prototype designs. Had the author no better suggestion for loading in such cases apart from the crude method shown in Fig. 18?

The final statement on tooth form in Appendix I would be more useful if limits were given for the actual values of normal pressure and helix angles instead of a general statement.

**Mr. S. A. Couling** (Visitor) wrote that he proposed to deal with only two points—firstly, hob tooth form and secondly, the contour of tooth surfaces.

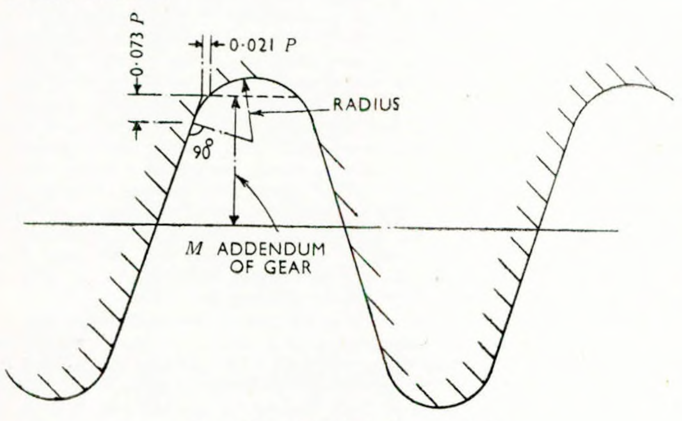



FIG. 22.—*Axial section of hob*

Whilst the author preferred the British Standard tooth form, his firm had used over the last twenty-five years a different one as represented by Fig. 22. The particular feature of this form was the amount and disposition of the tip relief. Tip relief in a peripheral direction on the finish gear was about seven times the amount produced from the British Standard hob, but only about half the amount down the depth of the teeth. This relief, which amounted practically to a radius at the tips of the teeth had been found essential to produce quiet running gears at high speeds. Further, he was not able to agree with the author that tip relief was not so important on helical gears as with spur gears, since with helical gears the tips of the teeth came into continuous engagement equivalent to the number of passages through the zone of contact at a rate equal to the number of teeth of the gear multiplied by the speed in unit time (see Fig. 23). Lack of tip relief, therefore, gave rise to contact frequency noise and this had been demonstrated in tests.

To take care of the extra end deflexions at entry to the zone

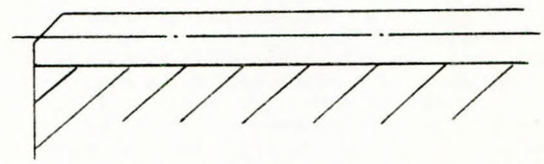


FIG. 23.—*Shows continuous engagement of the tips of the teeth*

of contact, the ends of the teeth were machine chamfered, as in Fig. 24. This had proved quite successful in practice having deflexions of the order of 0·0005 inch.

The author would appear to favour the use of the creep machine to overcome the noise directly attributable to the indexing worm and wormwheel of the solid table hobbing machine. His firm had successfully overcome noise frequency notes by using an indexing wheel with a large number of teeth correctly related to the cut gear so that within the zone of contact upwards of twenty-seven teeth were used in pinions and about four for large gear wheels in normal production.

From calculations made, it would appear that the ridge pattern or "scallops" were much more complex than the author's presentation. The design particulars of a hob were similar to that of a worm except that the worm thread was provided with gashes or flutes to provide cutting faces. Assuming a perfect hobbing machine the fluted hob, due to its down feed and its flutes, would produce a honeycomb pattern dependent upon the number of flutes around the circumference, its diameter and the rate of down feed. With a hob of 5-inch diameter and twelve flutes and assuming no errors in the machine a feed of $\frac{1}{32}$ inch would produce "scallop" or ridges of depth less than 0·0001 inch for the addendum to about half this amount for dedendum region and in a direction of the hob path. The down feed would produce scallops less than 0·00005 inch in the direction of the feed.

On this "perfect" pattern might be superimposed scallops or spots as described by the author produced by the driving worm and wormwheel errors. The pattern was dependent upon :—

(1) The accuracy of the master worm and wormwheel and the number of teeth in the latter compared with the number of teeth in the cut blank.

(2) The depth of the "scallops" relative to the normal hob and feed pattern which generally was less than 0·0001 inch.

It was his opinion that, with the solid table machine, the hob would produce a smoother surface in the finished gear than that from the complicated creep mechanism which would itself have errors and hence a scallop pattern. It was much more common to get pitch noise as distinct from hobber frequency noise nowadays from gears cut on solid table machines, particularly, if the number of teeth in the wormwheel were properly related to the job as explained above. From test records available the pitch noise from creep gears could be louder than the pitch noise plus the hobber noise from a solid table cut machine.

Finally, temperature changes whilst cutting gears, together with hob profile errors and inaccurate setting could produce distorted tooth

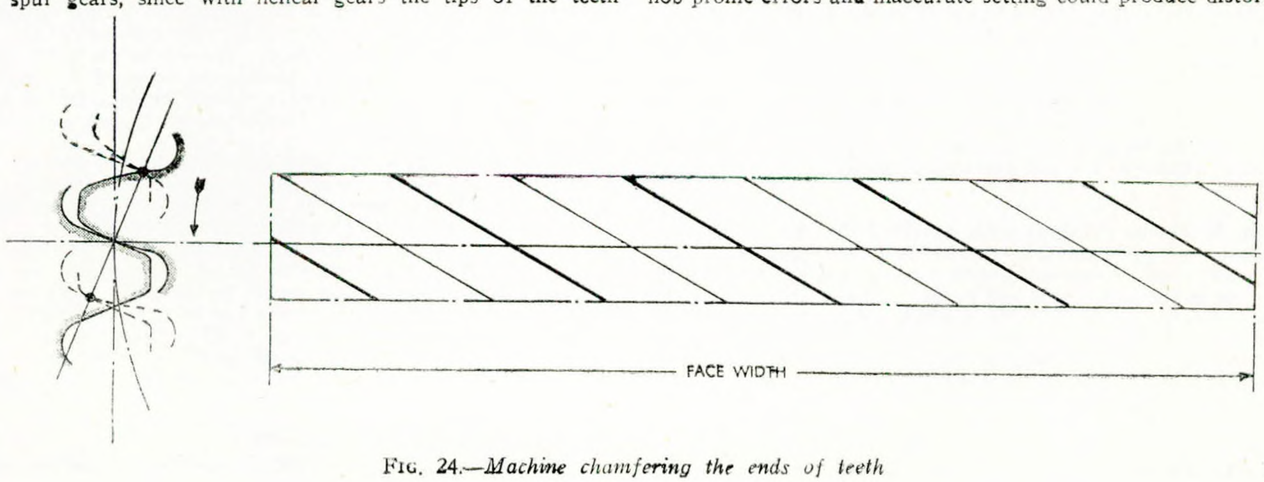



FIG. 24.—*Machine chamfering the ends of teeth*

forms and introduce troubles of more consequence than the worm cyclic error which had been reduced to less than 0·0001 inch in teeth of gear wheels up to 100-inch diameter.

**Mr. A. W. Davis** (Member) wrote that the assumption of novelty in the author's suggestions in relation to developments in tooth form that had in fact already been achieved, taken in association with the depressing reference to the dying art of gear design, led to the realization that the author could not be altogether familiar with developments carried out in the marine field over the past few years, and of the extensive research being undertaken in the realms of design by Pametrada, in association with certain of their member firms.

In a paper published three years ago* the whole question of tooth depth in relation to pitch was referred to in some detail and it was appropriate to quote the following extract:—

"The most outstanding example of a deep tooth form in marine practice is the Parson's 0·7in. pitch deep tooth, which has been extensively and successfully used in secondary gears, the tooth being 44 per cent in excess of British Standard depth for that pitch. For all other types of gear, however, it is considered that insufficient attention has been given in the past to the possibilities of teeth greater in relative depth to pitch than the B.S. form and although the minimum feasible ratio of pitch to pinion p.c.d. will usually result in restricting the depth increase to less than the 44 per cent quoted above, considerable benefit should be obtained by taking advantage of the greatest increase possible".

Since the publication of the paper referred to, two of the tooth forms instanced had been adopted with minor modification and in the case of one vessel in which a tooth form 20 per cent in excess of the British Standard depth/pitch ratio had been employed, nine months service experience had been gained with the most successful results witnessed in his experience. Incidentally the author would find on reference that the hobs for these gears were manufactured by his firm.

The success obtained with the new tooth form had been contributed to by the high perfection of tooth finish achieved by shaving and the absence of the slightest witness of surface distress was reflected by the remarkable silence of the gears in running. This led him to express emphatic disagreement with the author's statement that "excessive noise is objectionable only because of its effect on human beings subjected to it and does not necessarily imply mechanical inefficiency or overloading".

He agreed with the views expressed in favour of the creep device hobbing machine wherein, of course, the word "creep" referred to the motion between the main driving worm of the table and the table itself, but it must be recognized that the effectiveness of this type of drive depended entirely upon a satisfactory choice of creep ratio, a term which he preferred to retain having regard to its accepted usage. The author's brief reference as to the best value to be adopted for the fractional portion of this ratio seemed to be misleading in that he recommended a value slighly different from 1/2. This point was very fully discussed in the paper already referred to and the solution to the problem could be most clearly stated as follows:—

The creep fraction should be chosen so as to give the maximum avoidance of any vulgar fraction comprising simple digits, recognizing that 1 should be regarded as 1/1, and 0 as 0/1, these being the worst possible values and which should therefore be given the widest berth. The next most serious value was 1/2, and a creep fraction approximating to this value was bad, whereas the author stated it to be desirable. The best range was probably to be found between 1/8 and 2/5, or alternatively between 3/5 and 2/3, avoiding such fractions as 3/8 and 4/11 by as great a margin as possible relative to their diminishing importance.

In his remarks regarding the desirable accuracy of hobbing machine lead screws, the author quoted a maximum error which was a function of the axial distance between the points of measurement. This would be a justifiable guide were the machine to be employed for the cutting of spur gearing, but for helical gears, wherein each tooth that was in mesh at any instant had to deform to the extent of the whole inaccuracy of the helical angle over the width of the helix, regardless of what that width might be, the only reasonable criterion for the measurement of the screw was the maximum pitch error over a length corresponding to the maximum width of helix which the machine was likely to be called upon to cut. In the opposite sense the variation in truth of the hob saddle guides should be regarded over a length corresponding to the maximum helix width and not as the author suggested over their total length.

The author's remarks that the weight of the worktable held it in contact with the supporting surface and so prohibited any vertical freedom of movement was, of course, only true if either the supporting surface or the bearing surface of the worktable itself was perfectly true, otherwise there would be a rise and fall during each revolution of the table and this was a most common form of error.

In his various references to the all-addendum form of tooth as adopted by the Parsons Marine Company the author made no mention of the principal reason which led to its adoption, namely, the avoidance of contact at the pitch point where no sliding occurred, it having been found from experience that pitting was most prevalent at that zone of the tooth surface. He was not aware of any theory which satisfactorily explained this phenomenon and it would be of interest to have the author's views on this important point.

In referring to crossed axes shaving, the author stated that the helix angle of the cutter was such that it meshed with the gear with a shaft angle of 10 deg. to 30 deg. In this connexion it should be noted that the very successful results which had been achieved in the deep mesh method of shaving involved a differential shaft angle of only 6 deg.

In the matter of materials for marine propulsion gears the author stated that at present pinions were made of steel with a tensile strength of 60 tons sq. in. and wheel rims 45-50 tons tensile. There might be isolated examples of these types of steel having been used but the normal specification was for pinions to be of 40-45 tons tensile and wheel rims 31-35 or 35-40 tons. With the development of greater accuracy in gear production there was no doubt that higher tensile steels could be employed and he had had experience in experimental work of pinions up to a strength of 80 tons being satisfactorily hobbed. The case for the hardening and grinding of gear teeth might therefore prove less attractive with the development of harder gear steels employed for gears produced by hobbing and shaving.

The author made several references to the employment of running-in oil and there seemed to be some confusion as to whether he was in general referring to oil with an abrasive content, as suggested on p. 111, or an extreme pressure lubricant as referred on p. 112. In his opinion the use of any form of abrasive was to be strongly deprecated and in the production of high quality gears there should be no circumstances arising which might call for its employment. It was noted that in the author's experience avoidance of the use of running-in oil might lead to the necessity for subsequent readjustment of the bearing alignment. In his experience a gear that was properly aligned remained so regardless of the lubricant employed.

**Mr. L. M. Douglas** (Visitor) wrote that the recommendation of a creep fraction slightly different from 1/2 was unsatisfactory without amplification. In the first place if the inclination of the high spot lines to the axis was so small that the axial pitch of adjacent lines on any gear cut on the machine was less than the width of the gear there would be a hiatus between each pair of adjacent lines and the object of the creep would be defeated. Further, no account was taken of the pitch or depth of the secondary creep waves. Prior to the advent of post hobbing processes using a narrow tool, such as in shaving, the length of the secondary wave was not considered important since the amount of protruding metal forming waves of a given depth was unaffected by the pitch. If, however, gears were to be shaved, the depth of the wave was of secondary importance to their pitch, which must be small. At the same time it was desirable to keep the depth as small as possible in case gears cut on the machines were to be lapped or left untreated after hobbing. The depth of the primary wave decreased as the shift of phase per revolution of the work was increased up to a shift of 0·5 but the depth of the secondary wave increased with an increase of phase shift up to a maximum value of 0·5 of the error at a shift of 0·5. The shortest secondary wave was obtained when the secondary shift was exactly half the first but a further tertiary shift was necessary to avoid axial disposition of the lines of high spots and give these lines sufficient inclination to avoid a hiatus between adjacent lines on narrow faced gears. These matters must be taken into consideration if a satisfactory shift of phase was to be obtained.

Crossed axes shaving was stated to be the most useful hobbing process. It was certainly a good and quick process but not all would agree that it was superior to lapping done in a satisfactory way, that was with at least some relative axial motion, which excluded the definitely objectionable process of running gears together with abrasive without any axial relative motion.

Appendix I showed that the load carrying capacity was substantially unaffected by the pressure angle. This treatment applied to the static condition only. With a large pressure angle, the radius of curvature of the tooth flanks was large and the rolling velocity of the tooth flanks towards the point of contact was correspondingly high so that the entrainment of oil was more effective.

* Davis, A. W. 1945 Inst. of Engineers and Shipbuilders in Scotland. Vol. 88, p. 179, "Current Practice in Marine Gearcutting".

The last paragraph of Appendix I and the whole of Appendix II were based on the premise that relative sliding of the teeth was injurious. This was an assumption which could not be justified. In a journal bearing or a pivoted thrust bearing there was pure sliding with no rolling and satisfactory lubrication was attained provided the relative sliding velocity was not too low. If the sliding velocity was sufficiently high oil was entrained by the moving surface to maintain an oil film, and there was no doubt that a gear tooth functioned in a generally similar way substituting rolling for sliding. The principle difference was that in a gear both engaging tooth flanks rolled towards the point of contact and both contributed to the oil film by entraining oil irrespective of whether their speeds were the same or different. It was true that if the oil film was absent, i.e. if the teeth came into metallic contact, the liability to scuffing would increase with the sliding velocity, but this was an abnormal condition to which the gears should never be subjected, except at the beginning when the torque to be transmitted was very small. No high-speed heavily loaded gear could survive having metallic contact of the teeth at speed and load.

In regard to Appendix V the oil was not thrown off tangentially. Once inside the common tangent of the pinion and wheel on the approach side it could not escape otherwise than axially. It was pushed axially and, with a 30 deg. helical angle, at a velocity of $\sqrt{3}$ times the peripheral velocity of the gears, so that the power loss would be three times that calculated. However, the estimate of 0·5 per cent was high. In an ordinary marine turbine gear the total loss was of about this order including the power required to throw off the oil. However, with the adoption of much higher peripheral tooth speeds this loss would require consideration because while there was little information as to this power loss some experiments by his firm indicated that the total loss per cent varied as $\left(\frac{\text{velocity}}{\text{load rate}}\right)^{2/3}$ with a constant oil supply, so that if the velocity was increased without a proportionate increase in load rate the total loss would be increased both by the increase in velocity and by the increased power required to throw off the necessarily increased quantity of oil.

**Mr. H. M. Gemmell, B.Sc.** (Member) wrote that he was interested in the author's remarks on the load testing of gears and regretted that this aspect of the subject had been treated with such brevity in view of the paucity of information on this matter in technical literature.

So far as he was aware the fundamental assumption in back-to-back testing was that the losses in each of two identical sets of gears were equal, and half of the input power on test was allocated as the losses of one set of gears. He would like the author to explain how it was possible to detect difference in performance between the driving and driven members in a power circulation system.

It would be interesting to know the measure of agreement between the loss of power in a set of gears as measured by a back-to-back test, and as computed from the flow and temperature rise of the lubricating oil. Had the author any data available on this matter?

It was noted that Appendix VI dealt only with the simple case of single reduction single helical gears, excepting that the method of Fig. 21 was mentioned as applicable to a single set of double helical gears. This application, however, was admissable only by the acceptance of bearing loads different in direction from those obtaining under conditions of normal operation and, presumably, of tooth loading on the astern faces in one half of the gear.

Consequently, it appeared that load application by means of axial thrusts was useful only for single helical gears, if the conditions of tooth contact occurring in normal operation were demanded on test: thus a ready means of load variation during the test ceased to be available in the case of double helical gears.

Would the author explain the technique of back-to-back tests of double helical single reduction gears, with, say, three pinions to suit a three-cylinder turbine installation and of double helical double reduction gears for a two-turbine installation, and state if there were any simple means of producing part-load conditions on such tests.

In cases with multiple pinions it seemed to be essential for the pinion torques, proportioned according to the contribution of each particular pinion to the designed total power, to be applied simultaneously and the various pinion couplings then bolted together. Otherwise the application of torque between one pair of mating pinions would relieve the load previously imposed on another pair. In these circumstances he was of the opinion that the provision of a short calibrated length of shafting between the main wheel couplings would provide a means of checking that the torque in the main shaft was what it ought to be and give some assurance that the torques in the pinion shafts had not been altered from their intended value during hardening up of the couplings between pairs of pinions. Alternatively, or in addition, calibrated lengths might be fitted between each pair of pinions. Would the author state if such arrangements were desirable or essential for the successful conduct of gearing tests on the type of unit mentioned.

It appeared that only in certain cases would it be possible to test two sets of gears if they were intended for sister ships each with a single screw and that generally "one off and one to opposite hand" would be necessary.

Comment on such aspects of the problem would be appreciated, also any performance data or reference thereto as the only information found so far was that appearing in *Marine Engineering* by Seward published by The McGraw-Hill Publishing Co., Ltd.

**Capt. (E) J. G. C. Given, C.B.E., R.N.** (Member) wrote that the paper discussed problems of design, production and operation of marine propulsion gearing, pointed out some of the means by which improved performance might be obtained from present designs of transmission units, and indicated further problems which might have to be considered in the future.

Increasing output per ton of weight and per cu. ft. of space was an ever present demand for Naval propulsion machinery, and any departure from established practice which would help in this direction was of special interest for future projects.

With advanced designs of power units, it was to be expected that speeds of rotation much higher than present general practice might be adopted, and hence the size and weight of the reduction gearing unit might become excessive, unless a considerable increase in tooth loading capacity could be obtained.

Increase in sliding velocity must result from higher speeds of rotation of the imput shafts, and it would be of interest to know whether a limit must be imposed on the rotational speed of the power units on account of the effect of the increase in sliding velocity. Experience with existing marine gears suggested that pitch line speeds up to 300ft. per sec. were quite acceptable.

In Appendix I, the author suggested that the smallest pitch consistent with adequate root strength should be selected in order to reduce sliding speeds. With the increase in surface loading capacity derived from the use of harder materials, it might be expected that larger pitches than present general practice might be required if fatigue failure at the roots of the teeth were to be avoided. Present experience showed the need for root fillets of large radius to minimize stress concentration at these positions.

Experience with hardened straight spur gears showed that small modifications to the involute profiles were important factors contributing to good performance of heavily loaded gears. While the need for corresponding adjustment to the tooth profile might not be felt with present marine gears, it was possible that with single helical designs of low helix angle and a higher loading, attention to this feature might be essential to obtain maximum loading capacity.

It was frequently assumed that the oblique loading of helical gears entailed some concentration of loading near the mid-depth of the teeth but there was little experimental evidence to support this theory. Investigation of the effect of tooth deflexion on the load distribution along the oblique contact line between mating helical teeth was required to enable the degree of correction of the tooth profiles to be determined. The results of some preliminary work in U.S.A. on this problem had been published recently.*

The justification of the author's inclusion of a higher standard of accuracy in manufacture as a means of increasing loading capacity of marine gears would be evident from visual comparison between the tooth surfaces of existing large marine gear wheels and of high class gears of smaller sizes. There was no doubt that when the surface irregularities, arising from hobbing machine errors, had been reduced to the minimum, something still remained to be done to spread the tooth contact uniformly over the available surface. While the initial service of the gears had been relied on in the past to achieve this bedding in it would appear that with the use of harder materials it would be less effective. Now that shaving and lapping had been applied to large gears in this country with very satisfactory results, it was to be expected that bedding in would be superseded by these post hobbing processes.

**Mr. A. Hoare, Wh.Ex.** (Member), wrote that under the heading "Accuracy of Gear Teeth", it was stated that "noise does not necessarily imply mechanical inefficiency." Would gearing of 100 per cent efficiency make a noise?

The basis of 2·5 for the ratio of face width to root diameter, although generally accepted, seemed to be founded on mere opinion

* Davies, J. A. and Semar, H. W. 13th November, 1947 Paper presented to the Society of Naval Architects and Marine Engineers, New York, "Mechanical Reduction Gears".

rather than scientific reasoning. In view of its controlling influence on the size of the gear wheel there was a need to investigate this part of the problem with greater precision.

On the subject of lubricating gears, *Engineering* on 13th October 1922, published an article where the author showed that a high load carrying oil film was generated at high speeds. This seemed to imply that low-speed gears of the same tooth form could not sustain as high a tooth load as those running faster, which was the opposite of the author's statement.

Nothing had been mentioned of the effect of heat generated at the cutting point of the hob, nor of efforts to design hobs which resulted in the minimum heat transfer to the work from the action of cutting. True all hobbing machines were provided with a generous flow of cutting fluid, but this compound must be directed at the right spot and be capable of taking up the generated heat at the point of cutting. Any heat in excess of the prevailing temperature of the work and hob resulted in a deeper cut and, as inaccuracies of tooth form of 0·0005in. and less were being measured, local temperature rises of 100 deg. F. were sufficiently important to warrant attention. In some operations it had been found that low temperature cutting fluid was advantageous but for gear hobbing this would necessitate conducting fluid to and from the work instead of allowing it to cascade over the tooth face and cutter.

**Mr. W. F. Jacobs** (Member) wrote that the question of noise was particularly important to the operating engineer in that speech must be heard in the engine room without difficulty, and the noises must not be of a pitch to be confused with telephone or alarm bells. Noise did not mean that much power was being wasted, as for instance, a ball bearing was always more noisy than a plain one as well as being more efficient.

The actual loss in gears might be measured by the rise in temperature of the lubricating oil, though in an ordinary system this would include loss in bearings.

Referring to the noise level chart shown by the author it would be interesting to know the conditions under which it was made—the size of the engine room, vacant volume, skylights open or shut, and any other factors that would affect noise level.

It seemed to him that for the highest gear efficiency some form of post hobbing finish was required if only to level down the high spots which must exist in surfaces finished by milling cutters. The development of finish grinding on important gears seemed to be indicated.

**Professor Ewen M'Ewen** (Visitor) wrote that the author's discussion of the possibilities of profile ground case-carburized gears for marine use was timely, although he doubted whether the author had sufficiently emphasized the need for adequate control of distortion in the manufacturing stages preceding grinding. Profile grinding was not a satisfactory salvage operation and it was essential to keep the distortion in heat treatment down to acceptably low limits by adequate control of material, forging and heat treatment. Such control was, as the author pointed out, fairly well understood for automobile and aero-engine sizes of casecarburized gear; it was his contention that it would be better in the long run for marine development if the problems of control of distortion and the necessary expensive equipment were faced from the beginning rather than be forced into such practice as a result of trying to get by without.

The relatively high speed and low loading of marine applications made profile grinding desirable on case-hardened gears for this service since the stress raising effect of residual case tensile stresses was less disadvantageous than the stress raising effects of impact loads due to errors. But it was surely too much to say that no distortion occurred in flame hardening, particularly in view of the local heating involved. The author had shown in Appendix I that the load carrying capacity for surface stress was practically independent of the helix and pressure angles. It was worth while considering the effect of these variables on bending stress capacity:—

$$r_o = \text{contact ratio} = \frac{h}{p_n} \operatorname{cosec}\psi_t \sec\psi_t \cos\sigma$$

$$\text{Length of tooth root} = r_c F \cos\sigma \sec^2\sigma_o$$

$$= \frac{h}{p_n} F \operatorname{cosec}\psi_t \sec\psi_t \cos^2\sigma \sec^2\sigma_o$$

$$\text{Load normal to tooth surface} \propto \sec\psi_t \sec\sigma$$

$$\text{Rate of loading} \propto \frac{p_n}{hF} \sec^3\sigma \cos\sigma_o \sin\psi_n$$

The tooth form was a function of the pressure and helix angles, since everything else being equal an increase in pressure angle or helix angle increased the ratio of the tooth thickness at weakest section to that at the pitch line, but within the limits here concerned, this effect was small and the strength of the tooth might be considered as proportional to $p_n^2/hy$ where y was of the order of unity.

$$\text{Hence stress} \propto \frac{y \sin\psi_n \sec^3\sigma \cos\sigma_o}{Fp_n} = \frac{y \sin\psi_n \sec^4\sigma \cos\sigma_o}{Fp_t}$$

For a given choice of tooth numbers $p_t$ was fixed, and the load carrying capacity was approximately proportional to $\cos^3\sigma$ cosec $_n$.

Case-carburized gears were usually limited by bending fatigue rather than by surface failure, and it was clear from the foregoing that for such gearing the pressure angle had little effect since $y \sin\psi_n$ was approximately constant for pressure angles from 14·5 to 25 deg. It was also, however, very clear that the helix angle should be as low as possible, provided that the overlap ratio was in excess of unity, since the strength rating varied approximately as the cube of the cosine of the helix angle.

**Mr. M. C. Oldham** (Visitor) wrote that he was interested in the references by the author to the use of much higher strength materials than had been customary in the past for large size gears, and the various possibilities in hea treatment methods in order to obtain the results required. Unfortunately, no experience yet existed in the heat treatment, particularly case hardening methods, for such large forgings, where obviously complications arose in respect to possible surface cracking, and distortion outside the permissible grinding limits. The various suggestions made offered a most promising basis for research, involving heat treatment plant, which remained to be developed in a suitable form for dealing with such large gears, handling equipment, means of reducing the distortion which would normally occur, testing, etc. Such work demanded collaboration of the specialists in various types of heat treatment plant, and metallurgists, to deal with the severe problems involved, but no doubt in the not very remote future very useful progress in this direction could be anticipated.

**Mr. C. P. Rigby, B.Sc.** (Visitor) wrote that he thought that the author had shown undue bias in favour of the creep machine. With reference to p. 109, while it might be true that an increase in the number of teeth on the wormwheel of a solid table machine would in theory leave the high spots in the most objectionable distribution, it was nevertheless a proved fact that fitting a fine tooth wormwheel did reduce the noise and vibration at worm periodic frequency to negligible proportions.

This being the case, a gear wheel cut on a solid table machine with sufficient teeth on its wormwheel might be considered to have a distribution of high spots equally as good, for all practical purposes, as a creep cut wheel. The desired result was, moreover, achieved without introducing the additional errors inevitable with the creep mechanism.

As to surface finish, he himself had yet to see a creep cut wheel which would bear comparison, in that respect, with the solid table product. It seemed inevitable that the creep machine should produce a rougher surface because it introduced the cumulative pitch errors of wheels perhaps 12ft. in diameter as undulations on the teeth, whereas in the solid table machine the undulations were due to the smaller tooth to tooth errors of the wormwheel and wobble of the worm abutment.

Noise was an important matter and he had found that many people believed that creep cut gears were silent. The fact was that they made as much noise as solid table cut gears where the hobbing machines were of comparable accuracy. He had found it impossible to hear himself shout in destroyer gearing rooms with creep cut gears. The saving grace of creep had been that the mixed rumble of comparatively low frequency noise was far less objectionable than the worm periodic screams of a gear cut on a solid table machine with an inadequate number of wormwheel teeth. This distinction no longer applied between the products of a solid table machine with a fine tooth wormwheel and those of a creep machine where both were up to the best modern standards of accuracy, because in each case the remaining major components of noise were the tooth contact frequency of the mating gears and harmonics of pinion speed.

The creep machine had at first been a very necessary and useful palliative, but later by reason of its success, it had been a serious brake on progress towards accuracy. With modern standards and the advent of post-hobbing processes he could see little justification for its continued existence.

He thought that some reference should be made to the importance of giving an accurate hobbing machine fair treatment in service. It did not appear to be as widely appreciated as it should be that it was not reasonable to spend time and money on bringing a machine up to a really high standard of accuracy and then use it for heavy work such as, for instance, roughing large mill gears. An accurate machine should be reserved for fine work.

Under "Post Hobbing Processes", he thought the author had dismissed lapping rather casually; in point of fact, the latest method of lapping had resulted in the quietest running destroyer gears in the Navy, being somewhat quieter than a generally similar but slightly smaller set which had been finished by crossed-axis shaving. The latter process was not quite such a simple proposition as the author suggested, because a mathematically correct cutter and angle of crossing had in some cases failed to produce the correct tooth profile and the known existence of this profile error had probably contributed to the tooth contact noise which was predominant in the shaved destroyer gear. This tendency to increase tooth contact noise had also been observed in small gears shaved by the "crowding" method and it would be of interest to know whether the author had succeeded in getting to the root of the trouble.

An important point in connexion with post hobbing processes was that whether lapping or shaving was used, it was most desirable that the undulations on the teeth produced bv the hobbing machine should be as short as possible. This could be achieved by using either a creep machine with a suitable creep fraction or a solid table machine with a fine toothed wormwheel. The latter was preferable, unless the master wheel of the former was either very accurate or also had a large number of teeth, because the tooth to tooth error of this component was impressed directly on the hobbed gear.

**Mr. W. Sellar** (Member) wrote that while the suggestion that increase in the face-width/diameter ratio caused increase in maximum local surface stress/mean surface stress on the tooth flanks ratio was confirmed by actual results, the author did not state the immediate cause of the phenomenon. Sooner or later gear designers would have to treat the distortion of the pinion due to twisting and bending under working load as a major factor in correct design. If the gear cutting was perfect, i.e. when no power was being transmitted, the wheel and its pinion geared together with the teeth in perfect mesh, then the matter of pinion distortion must not be overlooked. That stage of perfection was being reached according to the progress indicated in the paper, but there was more in gear design than good workmanship. Was it possible that if more attention had been paid to this subject twenty-five years ago, the quality of finish now attainable would not have been necessary?

The effect of twisting and bending of the pinion was such that the end of the face-width of the gears nearest the power end of the drive must take the greatest load intensity, while the end farthest away took the next intensity, and the middle part the least. To generalize, the greater the length/diameter ratio, the worse would be the maximum surface stress/mean surface stress ratio, and *vice versa.* If a middle bearing was fitted the effect of bending was greatly reduced, but the effect of twist was still there unchanged. If the teeth were short and rigid the distribution of tooth loading (inferring the maximum stress/mean stress) was bad, but this condition was mitigated if longer teeth were used. This was borne out in practice.

The author said that no conveniently useful procedure appeared to exist for calculating the second of these ratios (maximum stress/mean stress) from the first (length/diameter), but it was not going too far to suggest that a simple solution would be found in studying the effects of the twisting and bending of the pinion.

These remarks led up to the principle referred to at the end of the section "Form of Gear Wheels". Actually it was the same problem. About twenty-five years ago a well-known American firm brought out a laminated disk wheel, the object being to give lateral deflexion by virtue of the axial component of the driving load on the helical teeth, and so relieve the heavy local load intensities. All the disks were of the same thickness, and of course, with the varied deflexions necessary to accommodate the pinion distortion to the local loads no such relief was possible, for if relief were obtainable momentarily the deflexion could not be held and the result would be vibration. Consequently in order to give the correct accommodation to the distorted form of the driving pinion under load, and to hold the deflexion permanently on a uniform distribution loading basis, he suggested that the disks should be thinner and more flexible axially at the sides of the wheel and thicker and more rigid at the middle, in accordance with the condition along the face of the pinion. This design was referred to as the graded disk wheel. A constant and uniform loading would result from this modification, and the uniform loading would produce the distortion of the pinion that would give the uniform loading (a form of mutual understanding) for any loading whatsoever—the ideal condition.

The procedure of cutting the pinion teeth on non-uniform helices referred to by the author was not a solution, it did not fulfil the conditions just mentioned, nor, theoretically, was the running in of the gears at a constant load, for the very reason suggested by the author.

The simplest solution was the graded dedendum pinion. The pinion was manufactured in the usual way, and then the spaces between the teeth were milled out and graded in such a way to make the teeth longer at the ends next the power and shortest at the middle (see Fig. 25). Thus, in the region of the greatest distortion of the pinion the tooth was "softest", and elsewhere in proportion to the variation in tooth deflexion on a conventional pinion for the same theoretical uniform loading. Further particulars regarding these designs might be found in his paper*. There, also, would be found a

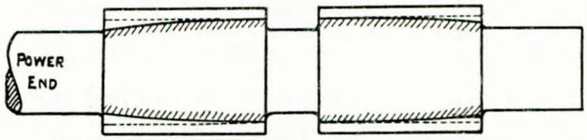



FIG. 25.—*Diagram indicating the lengthening of the teeth at the pinion ends*

suggestion for testing a graded dedendum pinion. Thus the circumference of a pinion of conventional design was marked off into four equal sectors and two alternately opposite sections were formed with graded dedenda as described above, while the remaining two sectors were left unchanged. In this way, under exactly the same conditions of service the relative merits of the two designs could be gauged. Incidentally, with the graded dedendum pinion, thinning down the teeth at the ends could be dispensed with.

**Mr. S. A. Smith, M.Sc.** (Member of Council) wrote that it was essential that limits of accuracy should be laid down to the gear cutting machine makers in order that inaccuracies in machines should not be repeated in the gears that they cut.

He considered that the limits of accuracy to which a machine must conform should be laid down in order that machines which did not conform to such degrees of accuracy should be rejected.

The machines should be placed in an air conditioned room maintained at a uniform temperature. He would mention in connexion with gear cutting machines, that the largest machine, certainly in the British Isles and possibly in the world, had no creep mechanism.

The author had mentioned variations in tooth form in the introduction. He himself had had some considerable experience of various types of tooth form, namely, the enveloping form, the all addendum tooth and the normal involute tooth. This experience had led him to return to the normal form of involute tooth as the most reliable and satisfactory in service.

It would be appreciated if the author would give his views on the adoption of 4/5in. or even 1in. pitch teeth as compared with 7/12in. for large gears.

With particular reference to the all addendum tooth he would like to ask the author why it had been found necessary to increase the pitch line speed to not less than 80 to 85ft. per sec. Was this a necessity for maintaining the oil film?

He thought that lapping of the teeth was advantageous in removing or reducing to a minimum the undulations across the face of the teeth, however, he would like to have the author's views on the relative merits of shaving and lapping.

With regard to gearing for ships he considered that when new, the gearing should always be run in at powers and loading not exceeding 75-80 per cent of the full power, and that this running in period should extend to six months at least.

Limitation of the life of a gear was in his opinion determined by the treatment it received in the early days of its service and gears that had been satisfactorily run in had normally lasted the life of the ship.

In the light of the latest developments in lubricating oils, he thought oils that were inhibited for rust and oxidization would prove in service to be superior to the straight mineral oils previously used but he would like to know the author's opinion on this important question.

The author would be well aware that fabricated gear cases were becoming more and more in use and he would like to have the author's opinion on the advantages or disadvantages of fabricated gear cases and cast-iron gear cases.

He had experienced gears, which after running for periods of two years had developed a grunt every revolution. Could the author give any reasons why such a phenomenon should occur since such a sound had only developed as a rule on one set of gearing in a twin screw ship, the other set being unaffected?

**Mr. A. H. L. Trapnell** (Visitor) wrote that he proposed to refer to the accuracy of gear teeth in conjunction with the accuracy of the wheels, the pinions and their mountings.

From the paragraph on accuracy of gear teeth it was apparent that errors in teeth of 0·0005in. caused appreciable stress concentra-

* Sellar, W. 1924 Trans. I. Mar. E., Vol. 35, p. 525, "A Basis for the Explanation of Marine Gear Troubles".

tions and noise. This being so it was considered that errors in the geometric form of the wheels and pinions and in their alignment, would have a comparable effect, although it was possible that the magnitude of such errors would be larger before equal defect was brought about. It was not known to what extent deflexion of the wheel under load occurred. If it was a built up structure the wheel would be subject to some deformation under load, and even if produced as a monostructure it would be unstable with the passage of time and with variation of operating temperatures. A static wheel or pinion, in a horizontal position, would take on a permanent set when supported at its two ends only, as in the case with a two bearing wheel or pinion.

In this connexion he would point out that most wheels for marine reduction gearing were hobbed in a vertical condition, whilst most pinions were hobbed horizontally. These were later assembled to run as pairs in a horizontal condition. Where shaving was employed this was done horizontally and the objection was overcome. Wheels and pinions were sometimes stored horizontally and in this condition would take on set. He suggested that they should be stored vertically to minimize this possibility. Could any form of structure of the size under review be held within the precise limits which seemed to be indicated if noise and stress concentrations were to be avoided?

It was understood that marine gearing had utilized a central bearing, but that this involved additional complications. It might be worth while to reconsider the incorporation of the third central bearing—a thrust pad as a precaution against deflexion under load—and against static sag.

To avoid errors of alignment was difficult, and it might be preferable to consider incorporating an adjusting device into one or more of the bearings. The third bearing would complicate this proposal.

As a final thought on this subject of rigidity, some amount of attention had been given to mass. There might be room for further consideration of form as opposed to mass. Press forming of mild steels would give them increased resistance to bending and deflexion. and permit a reduction of mass. The forming of heavy gauge sheets would present large problems, possibly insurmountable in this context, but it was suggested that there was a field for research into the form of wheel and casing structure to provide rigidity.

It had been said by the author that the height of the ridges between facets produced by hobbing, were of the order of 0·00002in. Tool marks by any refined process would be of this order of magnitude, except possibly by lapping and honing. It had been suggested elsewhere, that such tool marks were advantageous for lubricated surfaces, as they formed minute oil pockets. The surface elasticity of steel would permit the more minute surface errors, without detriment to quiet running, except where such errors formed a regular pattern in exact phase with like errors on the mating gear.

In conclusion, it was noted that the precision of marine production gearing had reached a standard demanding precise measurement control. Mechanical measuring equipment for the larger sizes used in marine gearing was not readily available. It was suggested that optical measuring equipment would be best suited for this purpose, as it was dependent upon accuracy of lenses and small mechanisms, and not upon large apparatus. There would appear to be a field for further research and development into the metrology of gearing.

## The Author's Reply to the Discussion

**Dr. W. A. Tuplin**, in reply, said that Mr. Richmond had suggested that the position of the first two factors in the introduction should be reversed, and that "Higher standards of accuracy in manufacture" being the more important should come before "Variations in tooth-form". He would not quarrel with that; the order in the paper of the items concerned was that in which they must be considered in design and manufacture.

He was interested to hear that experience was being gained with a depth to pitch ratio of 0·8, which was a probable optimum guessed from what had been successfully used. He thought it was an example of a quantity whose best value could be settled only by actual experience.

The criterion of loading of a lubricant seemed, from some experiments which they had made, to be the product of the surface stress and some power of the sliding velocity, approximately the square root. If, therefore, there were gears in the same box with different sliding velocities, and failure of the lubricant occurred on the one with the lower sliding velocity, that meant that the surface stress on those gears was disproportionately high. It was usual for the tooth loading to be higher on the lower-speed gears, and it might well happen that the product of the sliding velocity raised to the appropriate power and the surface pressure could be higher for the low-speed than for the high-speed gear.

Mr. Richmond referred to an optimum ratio of length to diameter. Personally, he was not sure what was meant by that. To get absolutely uniform distribution, the length to diameter ratio would have to be zero, and as one rose from zero the distribution would depart further and further from uniformity. The ratio of 2½ to 1 which he mentioned had been found satisfactory in general in that the highest local stress with that ratio was still below the permissible maximum. If it were possible to make the ratio of highest to lowest unity it would be possible to adopt higher nominal stresses than were used at present.

Mr. Richmond referred to the calculated variation in load capacity with change in helix angle, and pointed out that from that point of view a spur gear would be better than helical gears. He himself saw no reason to doubt that, if the spur gear teeth were carefully formed so as to reduce shock loading on engagement to a negligible amount. The advantage of the helical gear over the spur gear was simply that the former was inherently quieter, and for equal standards of accuracy would give a gear which was less noisy in running; it was not that the helical gear had greater load capacity than the spur gear except that its smoother action might lead to reduced impact stresses at tooth engagement. Even though the helical form might mean a slight reduction in load capacity, that was a small price to pay for its great advantage for high-speed work.

Mr. Richmond feared that the adoption of materials with very high surface stress resistance, such as case-hardened steels, might lead to failure by inadequate root strength—in other words, bending and fracture of the tooth—but it was possible to avoid that danger by reducing the number of teeth in the pinion and thus increasing the load capacity on the basis of bending strength, or in other words by reducing the bending stress for the same tooth loading.

Mr. Timms's suggestion that the feed screw in the hobbing machine should be provided with a corrector bar was a very useful one. At present the general practice was to send the lead screw for correction on a machine which had such a corrector bar, so that it was logical to suggest that if the hobbing machine itself had a corrector bar there would hardly be any need to send the screw for correction. As. Mr. Timms pointed out, this suggestion had the further advantage that if one had the staff with the necessary skill one could keep pace with wear of the feed screw by checking it up periodically, finding what had happened, and altering the corrector bar accordingly, so that there would be at least the possibility of maintaining the standard of accuracy of the feed screw throughout its life.

It was interesting to find Mr. Timms valuing the shaving process if only because it enabled measurement of the tooth to be carried out with greater certainty. Mr. Timms's point was that the ordinary hobbed surface was of a hump-and-hollow nature, and that when trying to check it with a stylus instrument, if the ball happened to drop into a hollow it might give an entirely different impression from that which it would give if its point of contact happened to be ¼ inch away. It was useful to point out that shaving, by removing the worst high spots, did help in accurate measurement.

His own suggestion that one shaving cutter should suffice for one pitch regardless of the number of teeth was at the moment an ideal rather than an accomplished fact. He mentioned it in that extreme form to draw attention to the difference between it and the tendency to make a shaving cutter for every gear, which was going to an expensive extreme in the opposite direction. If one aimed at using the same shaving cutter for all gears of its pitch one might not always succeed, but it would be more economical than uncritically assuming that every job must have its own shaving cutter.

His firm had tried a giant hob of German origin, but he could not say that they had found any advantage from the point of view of cutting performance. Theoretically it should make it possible to achieve any given standard of finish with a higher feed per revolution, and therefore it should enable the finishing cut to be carried out more expeditiously, but a considerable amount of time was in any case absorbed in the roughing cuts where the finish produced was obviously not of great importance, and it was doubtful whether a giant hob was of any great advantage in reducing the total time required for cutting the gear. One disadvantage of a hob of great diameter was that it increased the torque on the hob spindle, and on many of the components back from the hob spindle, so that when it came to a finishing cut, where the highest accuracy was required, it was at least doubtful whether the bigger diameter was really advantageous.

Mr. Ferguson had suggested that he had said that gears which were silent were not necessarily good. He had not actually said that, although it was probably literally true. The mere fact that a gear was silent did not prove that it was good from every other point of view, though it was at least extremely probable, because if gears were silent they must be very accurate, and if they were very accurate they were likely to be very good from the point of view of load capacity. What he had said was that a noisy gear was not necessarily bad from the mechanical efficiency point of view, although the personnel near to a noisy gear might find it very unpleasant.

The difficulty mentioned in connexion with the grinding of gear teeth, namely the production of grinding cracks, was one which had arisen many times. It seemed that it could be countered by the selection of appropriate material and by appropriate heat treatment, but the essential factor was care in grinding. Some case-hardened steels were said to be less prone to cracking in grinding than others, but he had no evidence on this point.

On the subject of castings as against forged steel, he would not be prepared to commit himself to anything beyond the statement that the possibility of flaws in the material was greater with a casting than with a forging. In fact that operation of making a forging included the closing up of voids and the reduction of certain types of defect in the steel, so that a forged material which started as a casting and had been subjected to corrective treatment was almost bound to be a little less risky in respect of hidden defects.

In reply to Mr. Hallewell, he was afraid that he could not see any advantage in stub teeth. They had been tried a number of years ago, and had gradually disappeared from the gearing field simply because on balance they were found to have no advntage. On the contrary, where the kinematic conditions permitted, the load capacity was increased by the opposite departure from standard, namely by using a greater depth of tooth in relation to the pitch. In gearing for general application, where it might be desired to have pinions with, say twelve teeth, there was a limitation on tooth depth which did not apply so severely when the number of teeth in the pinion need not be less than about thirty, which was roughly what applied to marine turbine gears; so that it was possible that for that relatively limited field of application, where the minimum number of teeth in the pinion was not very small, a higher ratio of depth to pitch could be adopted than would be useful or economic or practicable in a gearing system which had to cover a wide range of numbers of teeth.

The advantage of helical gears, as he had said, was that they were inherently quieter than spur gears. That advantage became more and more important as the speed rose. If one were dealing with gears at only moderate speeds, and if one were able to attain a high standard of accuracy by grinding, there was no reason why spur gears should not be used, and it was well known that they were used a great deal in high duty applications, as for example in aircraft, where, of course, the noise was not very important, simply because it could not be heard above other noises. If, however, the speeds were low, and it was possible to obtain the accuracy necessary for quiet running with spur gears, there was no reason why they should not be used, and the helical gear had no special advantage.

It had been found in the past that the substitution of plain sleeve bearings for ball or roller bearings did tend to reduce the noise produced by any given pair of gears. In the anti-friction type of bearing there was metal-to-metal contact, so that any vibration set up by the gears was transmitted with very little loss to the gearcase, which amplified the noise effects by acting as a sounding board. In the plain bearing the oil film, although it was very thin, seemed to have a perceptible damping effect, and it had been found on many occasions that plain bearings did reduce the noise produced by a gearcase as compared with the noise which the same gears would set up when running on anti-friction bearings.

As a comment on Mr. Ainslie's remarks, it could be said that Sir Charles Parsons was certainly a pioneer in many fields, including the application of gearing to steam turbines. It had been made clear in other ways that he had studied the problems very carefully, and in fact he originated the creep machine, which in itself was a proof that he had gone very deeply into the problem of noise production, and that he had achieved a practical way of securing an improvement.

In reply to Dr. Brown's question about the form of the end relief shown in Fig. 4, the point was that if the oblique boundary of the relieved area were parallel to the generator, contact between the teeth would begin simultaneously at all points on the boundary. By inclining the boundary to the direction of the generator, contact could be made to begin at a point and engagement might therefore be expected to be smoother than would otherwise be the case.

In the author's reference to noise in relation to mechanical efficiency and overloading, he had in mind variations large enough to be of practical importance. Noise of distressing intensity might be caused in high-speed gears by tooth errors too small to produce any appreciable overloading and the energy dissipated in sound was usually quite negligible by comparison with the full-load rating of the gears. Excessive noise did not necessarily imply overloading or loss of efficiency that measurably impaired the performance of the gears.

The shaving process as normally used with either "crowding" or "brake-loading" might be expected to reduce the difference in height above the true profile of any two points on a generator of the tooth to an amount depending on the axis-angle in shaving, on the distance between the points concerned and upon the diameters of cutter and work.

He had seen no sufficiently convincing theoretical analysis of shaving action to enable him to say more than that, but in practice, high spots spaced at distances up to about 0·8 inch were reduced in height to about 0·0001 inch.

By varying the loading on the teeth as the shaving cutter was traversed across the width of the gear, a small change in helix angle might be produced; the success of such an operation depended, of course, on a certain amount of skill and experience.

Since it reduced the errors in the tooth-form, shaving tended to make the gears quieter in operation but it seemed probable that hobbing on a creep machine with an exceptionally fine feed would be equally effective but much slower. Shaving was broadly equivalent to a very fast fine-finishing cut on the hobbing machine.

Use of a running-in oil (which did not contain any abrasive) had the same effect as that of the service oil except that the bedding-down of high spots took place more quickly. If time could be spared for prolonged running-in with the service oil, that was equally effective.

Running-in the gears in the gear-case tended to give automatic compensation for small mounting errors; separate lapping of wheel and pinion offered nothing comparable. If exactitude in mounting could be guaranteed, this point would not be important. For a pinion subjected to no total transverse loading a facewidth/diameter ratio up to 4 should be satisfactory.

In proposing the form of wheel shown in Fig. 8, he had not envisaged the necessity for transmitting axial loading. If that were expected in any particular case, its probable effect would of course need investigation. It was not claimed that the unsymmetrical overhang of the rim could be calculated so as to make tooth loading exactly uniform despite the distortion of the pinion; all that was suggested at this stage was that it would be a step in the right direction.

The power lost in the teeth and the lubricant of helical gears was such a small fraction of the full-load rating that it was hardly worth while to go to special pains to measure it. A test carried out many years ago by his firm on helical gears of 24-inch centre distance, 10-inch facewidth working in a power circulation system with oil-bath lubrication showed that the loss in teeth and oil was about 0·3 per cent of the full-load rating of the gears. This figure was probably subject to considerable proportional variation dependent on details of the lubrication and the 0·5 per cent quoted in the paper might be taken as a safe upper limit for normal conditions in turbine reduction gears.

If viscosity were the criterion of load capacity of a gear lubricant, it would always be possible by use of oil coolers to allow heavily loaded gears to run with oil taken from a common system, and in fact that practice might often be successful, but it would be too much to suggest that an oil could be made suitable for gear lubrication simply by cooling it.

The gear loading schemes illustrated in Appendix VI applied to single helical gears. The scheme shown in Fig. 18 could also be applied with double helical gears; he could suggest more elaborate schemes but nothing simpler.

He found it necessary to question Mr. Couling's comment that "a radius at the tips of the teeth had been found essential to produce quiet running gears at high speeds". This could not be accepted as a general statement as it was known that some helical gears with no tip-relief whatever ran quietly at high speeds. Mr. Couling's remaining comments in his first paragraph were interesting but did nothing to invalidate the statement that tip relief in helical gears was not required for the purpose that made it imperative in fast-running spur gears.

He agreed that the main wormwheel of a hobbing machine (whether a "creep" machine or not) should have the largest practicable number of teeth as that minimized the effect of main worm error in producing irregularities in the tooth-profiles of gears cut on the machine. The suggestion that there was a special value in making the angular pitch of the table gear teeth small compared with the angle subtended at the axis of the gear by the ends of the zone of contact failed, however, to withstand critical examination based on appreciation of helical gear contact conditions. If it were sound, it would imply that helical gears should be of coarse pitch so that the zone of contact should be wide.

Teeth cut on a perfect machine would show more scallops than indicated in Figs. 13 to 15. Those diagrams referred, however, to the condition in which the main worm had an error so great that the low cuts produced one for every revolution of the worm removed the surfaces of shallower cuts. No matter how great such error might be, the resulting tooth surface was as shown, its peak-to-valley heights being determined by the dimensions of the hob and the feed-rate.

Gears cut on a creep machine would always tend to be quieter than similar gears cut under similar conditions on a similar non-creep machine of comparable accuracy, because of the more favourable distribution of high-spots on the creep-cut teeth. A non-creep machine, if made sufficiently accurately, might nevertheless produce acceptably quiet gears.

The "assumption of novelty" in the suggestion to use deep teeth was entirely Mr. Davis's as the advantage of that feature had long been well known, i.e. well known to some people. The writing of this paper was taken as an opportunity to give a geometrical analysis that demonstrated the importance of the depth/pitch ratio and showed how little was to be expected from changes in pressure angle and helix angle.

He was well aware that his firm had manufactured deep tooth hobs for marine purposes. He was also aware that they had manufactured hobs with tooth forms of many and diverse characters to clients' specifications and this had led him to the depressing conclusion that a great amount of ingenuity and money were being wasted by designers in pursuing ideas that had been tried and abandoned many years ago.

He confirmed that he was not aware of the results of any research in this direction by Pametrada or member firms.

The author was gratified to learn that Mr. Davis had observed what seemed to be excellent performance by deep-tooth gears. However it must be borne in mind that the excellence of the gear performance of the gears in a certain vessel might, or might not, be a consequence of the use of deep teeth. Although extremely tempting, it was not safe to accept an isolated example as evidence in support of a preconceived theory.

It was true that perfect gears would be at once enduring, silent and free from power loss, but to conclude as Mr. Davis did, that noticeable departure from the ideal in one of these respects must also imply noticeable departure in the others, was hardly rational. The facts were firstly that the power loss in noise was quite negligible and secondly that noisy running was no guarantee of short life.

Mr. Davis's comments on creep fractions, although broadly correct so far as they went, were evidently based on an incomplete appreciation of the problem involved. What had been overlooked was the fact that a creep fraction of 0·5 gave the maximum overlapping effect between deep cuts and shallow cuts on successive lines of cuts by the hob and so minimized the roughness resulting from any particular error in the machine. A creep fraction of 0·5

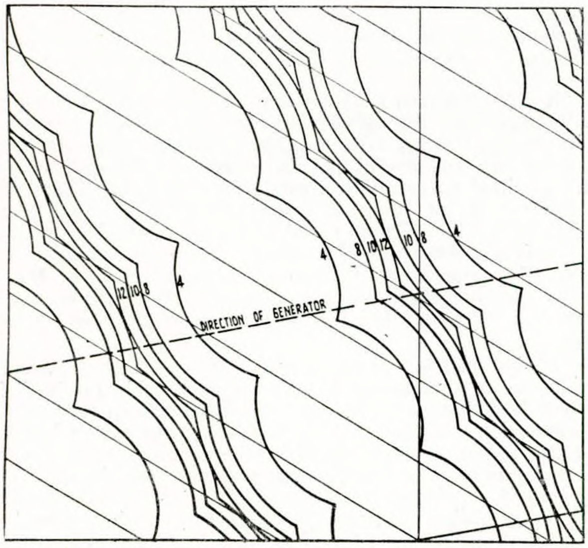



FIG. 26.—*Contours of part of tooth surface at heights 4-8, 10, and 12/100,000 inch above low spots.*
*Creep fraction $q = 0{\cdot}36$. Axial speed $f = 0{\cdot}04$ inch per revolution*

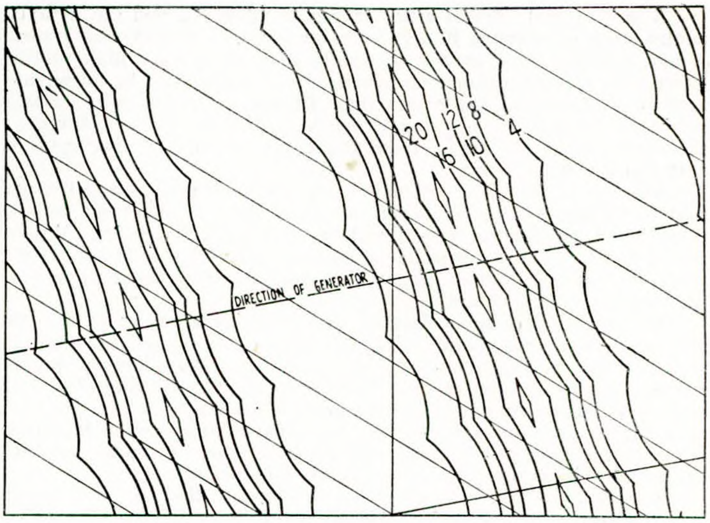



FIG. 27.—*Contours of part of tooth surface at heights 4/100,000 to 20/100,000 above low spots.*
*Creep fraction $q = 0{\cdot}285$. Axial feed $f = 0{\cdot}04$ inch per revolution*

was not ideal but it was very much superior to a creep fraction of zero.

The ideal was to depart from 0·5 by the least amount that would angularly displace the lines of high spots corresponding to 0·5 sufficiently to allow the line of contact to bridge them. The departure necessary for this purpose was surprisingly small, and for average tooth proportions and rates of feed in hobbing, the desired effect was secured if the creep fraction lay just outside the range 0·47 to 0·53.

The feed rate of 0·04 inch per revolution of the work, in conjunction with a worm creep fraction of 0·36, which lay within the range suggested by Mr. Davis, gave the contours shown in Fig. 26. The tooth surface, although smoother than for zero creep (Fig. 13) was inferior to that produced with a creep fraction of 0·46 (Fig. 14). The effect of a creep fraction of 0·285 was shown in Fig. 27.

Mr. Davis's remarks about the necessary standard of accuracy in hobbing machine feed screws were a little obscure. If the machine were to be used solely for cutting spur gears it would hardly be necessary to set any limit at all on the inaccuracies of the lead screw as their only effect would be to cause irregular spacing of the feed marks on the teeth of the work. The effect of lead screw error on the positional errors of helical gear teeth cut on the machine was proportional to the sine of the helix angle.

Any method of fixing tolerances must take into account practical possibilities in manufacture and that was why it was usual to specify that the error permissible over a certain length of the lead screw was greater than that permissible over a smaller length. As it must be presumed that the full length of the hob saddle guides might have to be used at one time or another and as their total lengths in all machines bore an approximately constant relation to the maximum widths of gears that could be cut, there seemed to be little advantage in introducing the maximum width of gear (in place of the length of the guides) into the specification of permissible error in the guides.

His own view had long been that the real reason, as distinct from the published one, for adopting the all-addendum form of tooth was that as there was sliding at all points on its surface, it facilitated the lapping that was necessary to correct the best tooth profiles that could be produced by the "creep-cutting" machines available at the time of its introduction. This virtue vanished, however, when the lapping had been accomplished.

Pitting usually occurred on gear teeth near to mid-depth, whether the pitch line was in that vicinity or not. The mere shifting of the pitch line to the root (or tip) of the tooth did not in itself affect the tendency to pitting at or near mid-depth.

The occurrence of pitting near to the mid-depth of spur gear teeth was explained by the fact that the greatest surface stress occurred at one end of the phase of engagement in which there was only one line of contact. The position of the point of maximum surface stress in the depth of a helical gear tooth could not be deduced theoretically with the same certainty as for a spur gear, but the position at which pitting was observed to begin in helical gears might perhaps be a practical answer to the question.

The fact that crossed axis shaving could be carried out at axis-angles outside the range 10 deg. to 30 deg. was, of course, well known. A figure within that range was usually adopted where a high rate of production was important; elsewhere, there might well be an advantage in adopting an angle less than 10 deg. as the slower cutting might be accepted as a price to be paid for the higher degree of accuracy that might result from the closer contact of the teeth of cutter and work.

The use of 60-ton nickel chromium steel for pinions and 0·4 per cent to 0·5 per cent carbon steel for wheel rims in industrial turbine gears had been standard practice at his firm for many years. Whilst aware that materials of lower load capacity were used in many marine gears, he had been loth to suggest that marine gear practice in general was lagging to that extent.

He agreed that application of any form of abrasive to finished gear teeth was most undesirable. A running-in oil did not contain abrasive but permitted more rapid wear to take place at highly stressed points of contact than did a normal oil. Bedding-down of high spots during running-in of the gears when finally mounted tended to compensate automatically for any unavoidable small departure from parallelism of the shafts, and this was true, of course, whether the running-in process ws slow or relatively fast.

In reply to Mr. Douglas, he would say that he had never been satisfied that attempts to describe the form of the surface of a helical gear tooth by "wave-lengths" measured on a line parallel to the tooth tip were at all reliable. As an example he would draw attention to Fig. 14. This represented the contours of a tooth surface for which (so he was led to understand) the calculated "wave-length" was several inches, whereas the most prominent high spots on the surface were spaced at about 0·08 inch, i.e. about twice the feed per revolution of the work.

On a helical gear tooth of British Standard proportions and 30 deg. helix angle, a creep fraction of 0·47 slewed the high-spot lines corresponding to 0·5 sufficiently to cause a contact line always to bridge two such lines, irrespective of the facewidth of the helix, provided that it exceeded about 2½ times the transverse pitch.

Crossed-axis shaving gave relative motion of the coontacting surfaces parallel to the tooth tip at all phases of engagement, quite apart from any feed motion of the cutter parallel to the axis of the gear.

At the beginning of the fourth paragraph Mr. Douglas stated that the assumption that relative sliding of the teeth was injurious could not be justified. This statement conflicted with the fact, well established by many fully authenticated cases, of gear teeth spoiled by "scuffing" which was a consequence of relative sliding under pressure in a combination beyond the capacity of the lubricant to handle. In such cases scuffing invariably started where the sliding velocity was highest and never where it was zero or a very small fraction of the maximum to be found in the teeth.

Mr. Douglas was correct in pointing out that some at least of the oil sprayed on to the teeth might be projected axially at a speed higher than the peripheral speed of the teeth. Not all the oil was thus treated, however, and if any considerable proportion of the total emerged axially from the teeth, it was a sign that the oil was being sprayed at an unnecessarily high rate. The essential fact was that the power loss need not exceed 0·5 per cent of the transmitted power at any peripheral speed likely to be used in the near future.

In reply to Mr. Gemmell he had to say that while there was theoretically a small difference in tooth efficiency between a pair of gears operating under certain conditions with pinion driving and a similar pair under the same conditions but with wheel driving, it was not sufficient to explain the dfference that was usually observed in a power circulation test. This remark referred to efficiencies calculated on the assumption of a constant coefficient of friction between the teeth at all phases of engagement. If this assumption was appreciably in error, it was possible that the difference in efficiency might be explained by the fact that the gear with the long addendum drove the one with the short addendum in one assembly whilst the opposite was the case in the other.

The most reliable method of determining the loss in each gear assembly would seem to be that based on meaurement of the rate of heat convection by the oil and the division of the difference between this and the measured power input (i.e. the power lost from the gear cases by radiation convection and conduction) between the gear units proportionately to their observed temperature rises above atmosphere.

In back-to-back testing of multiple-pinion units it would seem best to insert in each connexion between pinionshafts some torque developing device whose effect can be adjusted whilst running. He had no actual experience with such devices, but a hydraulic torque-loader developed at Pametrada Research Station was understood to have given satisfaction on trial. With any scheme of this general character a torsion dynamometer should preferably be provided in each pinion shaft so that no reliance need be placed on the loading device as a means of measuring torque.

In reply to a point raised by Captain Given, he would say that increase in power transmitted per unit of weight involved increase in maximum surface stress on the teeth, or increase in maximum sliding velocity, or both. This meant that lubrication became more difficult and eventually it might become necessary to use more highly specialized oils than were necessary for marine gears today.

Owing to the fact that a change from direct-hardened steel to case-hardened steel gave a greater increase in load capacity as determined by surface stress than as determined by bending stress, the full advantage of case-hardened steel was obtainable only by adopting a smaller number of teeth for the pinion than was appropriate for direct-hardened steel.

In reply to Mr. Hoare it might be said that whilst gears of 100 per cent efficiency would be silent, gears of 99·99 per cent efficiency might easily be extremely noisy.

The face-width/diameter ratio of 2·5 for pinions had come to be accepted because general service results with many gears over long periods suggested that it was a satisfactory limit with the nominal working stresses that had been used in the past. The higher the nominal stress for any given materials, the smaller must be the limiting ratio of facewidth to diameter.

The suggestion that increase in speed of a pair of gears led to increase in tooth-load capacity was opposed to experience.

He had no evidence that heat generation at the cutting edges of a hob taking a clean finishing cut had any measurable influence on accuracy of the product. Such heating was unlikely to vary appreciably between different points on the tooth surfaces and so any effect it did have might be expected to be uniformly distributed and thus to be unimportant.

In reply to Mr. Jacobs, the noise level readings were taken in an engine room open to the level of the upper deck in a vessel of 12,000 tons; the height from floor to skylight was about 40 feet.

Professor McEwen was correct in stating that the greatest difficulty in the way of producing case-hardened steel gears for marine main propulsion would be that of restricting distortion in heat-treatment to very small amounts. He might be assured that those responsible for certain projected experimental work in this direction were under no illusion on this score.

It seemed certain that case-hardened steel marine turbine gears would need profile-grinding to attain the degree of accuracy neecessary to permit the gears to compete in respect of load capacity with gears not heat-treated after hobbing. There was some doubt (possibly based on pious hope) whether the same necessity would apply to flame-hardened gears, although of course no one claimed that flame-hardening produced no distortion at all.

He agreed with Mr. Oldham in emphasizing that the production of marine main propulsion gears of case-hardened steel was likely to be beset by many difficulties, at least until techniques were delevoped to overcome them. He thought that the gain in load capacity by the substitution of case-hardened steel for (say) 60-ton steel might be less in large gears than had been found in small gears, for example those in automobile gearboxes. To establish exactly what advantage was obtained, carefully controlled comparative tests would be necessary.

Mr. Rigby's comments, at first rather immoderately "anti-creep" practically cancelled themselves out on the "creep or non-creep" question. He said in his fifth paragraph that the creep machine had been a necessary and useful palliative but then he endeavoured to make its admitted success into a cause for criticism. There was no object whatever in pursuing accuracy merely for its own sake, and any artifice that achieved the desired end more easily than by extreme accuracy was a worth-while alternative. The creep principle proved its worth when standards of accuracy were much lower than they were today and an evaluation of the creep principle could be made only by comparing the products of equally accurate creep and non-creep machines.

Mr. Rigby "had yet to see a creep-cut wheel which would bear comparison as to surface finish with the solid table product" but it should be added that such creep-cut wheels have been seen.

In connexion with the shaving versus lapping comparison it was of little use to cite a single example of a pair of gears in one ship that were thought to be quieter than a similar pair in another ship. Neither was it convincing to mention that crossed-axis shaving had in some cases failed to produce the correct tooth profile because (a) it was doubtful whether small changes in tooth profile of helical gears had any influence on noise production, (b) there were differences of opinion as to what was the "correct tooth profile" for helical gears, (c) lapping also occasionally failed to produce either the correct tooth profile or the intended tooth profile.

He had no detailed information about any case in which crossed-axis shaving could be suspected to have caused an increase in tooth contact noise.

A creep machine with a suitable worm creep fraction produced high spots more closely pitched and more favourably situated than those produced on a non-creep machine with the same number of teeth in the wormwheel and the same hob-feed per revolution of the work.

Mr. Sellar had explained very clearly the effect of the inevitable flexibility of the pinion. As theoretical analysis of the problem appeared to be extremely difficult, practical test was likely to be a more reliable means of determining the variation of surface stress across the width of the teeth.

The details given of the laminated disk wheel were extremely interesting. With rise in nominal tooth loading, such a construction was likely to be more valuable in the future than was the case twenty-five years ago. The "graded dedendum" pinion would seem to be equally valuable and of course considerably simpler.

Mr. Smith's suggestion that the first step in producing accurate gears must be the use of hobbing machines maintained within specified limits of error was one that would find general agreement. British Standard limits for this purpose had already been drafted.

The pitch adopted for any pair of gears was preferably that which equalized the load capacities as determined by surface stress and by bending stress. This meant that for any particular combination of materials, the number of teeth in the pinion depended on the velocity ratio of the gears. It was thus impossible to say that 0·5-inch pitch or 0·8-inch pitch or 1-inch pitch was either a fine pitch or a coarse pitch. What was fine for gears of 80-inch centre distance might be coarse for similar gears of 40-inch centre distance.

He was not aware of any necessity to specify a minimum pitch line speed for gears. On the contrary it was desirable to keep down the pitch line speed so far as was consistent with economical proportions of the gear unit as a whole.

Crossed-axis shaving could do more quickly all that was possible by lapping and without the disadvantage of possibly leaving abrasive embedded in the teeth.

There was every reason to suppose that careful running-in over a long period was essential if gears were to give their very best possible performance and he was interested to learn that Mr. Smith's experience was in agreement with this view.

The author would expect that specially inhibited oils would give superior service because of avoidance of the deterioration that the inhibitor was intended to combat, but would not anticipate that the inhibitor would improve the performance of the lubricant when first set to work.

The advantages of a fabricated steel gear case over a cast-iron case were that the former could be made lighter for the same strength and rigidity and that it was less expensive to make unless several identical cases was required, when cast-iron might become the cheaper owing to spread of the pattern cost.

He could not advance any general suggestion as to the cause of development of periodic noise in gears after some years of service.

Mr. Trapnell's contribution raised a number of interesting points and it certainly did seem possible that steel plate wheels designed for lightness might develop dimensional changes with the passage of time. As these changes would probably be of such a nature that they were less likely to produce local tooth errors than accumulated pitch errors over large arcs, they need not be expected to give rise to objectionally noisy running or to high local stresses, but they might well be responsible for a noise "surge" of the type mentioned by Mr. Smith.

It would certainly seem preferable to store gears with their axes vertical rather than horizontal, but he could not say that he had any direct evidence that the second alternative had been the cause of any serious trouble in subsequent service.

The possibility that facets on a tooth surface acted as pockets for oil was one that had often been mentioned but he was inclined to believe that the suggestion might have arisen from a subconscious desire to make a virtue out of a necessity. On perfect tooth surfaces, lubrication demanded no more than the merest film of oil and the possible advantage of relatively deep pools seemed likely to be outweighed by the stress concentration on the boundaries between the pools.

## Sea Water Contamination of Boiler Fuel Oil and its Effects

By Eng'r Rear Admiral C. J. Gray and Wycliffe Killner

The authors wish it to be noted that they unfortunately omitted to acknowledge the valuable help given by Dr. A. S. C. Lawrence when replying to some of the more complicated chemical problems which arose during the discussion.

## MEMBERSHIP ELECTIONS

### Date of Election, 10th May 1948

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Francis Doonan
Edward James Kenneth Goldsmith
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Kaikhosrow Adar Irani
George Sidney Jackson
Edward Turner Kennaugh
John William Ladyman
Donald Edward McGuinness
Philip Wreaks Mason
James Melrose
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Lewis Daniel Norton
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Edward Henry Roberts
George Sydney Roberts
Stanley Watson Swain
Gilbert Basil Taylor
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Thomas Barr Malcolm
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**Graduates**

James Stewart McAllister
Charles McKinstry

**Students**

Geoffrey Herbert Fuller, Constr. Sub. Lieut., R.C.N.C.
James William Geoffrey Hall

**Transfer from Associate to Member**

Alfred Louis Giordan, Lieut.(E), R.N.
Neville Henry Westacott

## OBITUARY

T. C. Barry (Member) who was elected a Member in 1928 has been reported missing after the unfortunate mishap to the S.S. "Samkey" on the 3rd March 1948.

A. G. Cornford (Associate Member) was born in London in 1904, and educated at Dulwich Hamlet School. He served his apprenticeship at Blackwall Iron Works, Poplar. He then entered the firm of Messrs. Mitrovitch Brothers, London as junior engineer on motor boat trading from Liverpool to Carnarvon and West Country ports.

He then joined Messrs. Nourse and Sons, leaving Liverpool at Christmas 1925 on S.S. "Explorer" to join S.S. "Sutlej" at Calcutta. In 1927 he returned home from Havana. In June 1928 he joined Messrs. Andrew Weir and Co., and served two years in Aruba, and then continuously except for six months, when employed by United Baltic, by permission of Messrs. Andrew Weir and Co., before sitting for the Board of Trade examination. In the early stages of the 1939-1945 war he was 2nd engineer on M.V. "Inverilen", and was later appointed chief engineer on M.V. "Empire Marvell" until he suffered a temporary breakdown. In May 1946 he travelled to Rotterdam to bring the M.V. "Inverbank" to Falmouth, leaving in June for U.S.A., Australia and New Zealand, trading from Nauru Island to various ports. In December 1947 after leaving for Auckland, New Zealand he had an attack of dengue fever. On arrival at Auckland he was taken to hospital, and eventually flown home, arriving in February 1948. He was elected an Associate Member in 1933. He died at his home in North London on the 6th April 1948, and leaves a widow.

A. C. Cruickshank (Member) was born in 1885. After a five years' apprenticeship he served as engineer to Messrs. Bullard, King and Co., and Messrs. Jardine, Matheson and Co. In 1910 he was appointed chief engineer at the Shalimar Paint Works, Calcutta, and from 1913 to 1916 he served in the same capacity, the Lodna Colliery Company, Calcutta. From 1916 to 1919 he served as Lieutenant, and later, Acting Major in Basra on the Headquarters Staff of the Royal Engineers. From 1919 to 1921 he was superintendent engineer of Messrs. Turner, Morrison and Co., Basra, and from 1921-24 as chief engineer of dredgers and tugs on the South African Railways and Harbours, Durban. In 1924 he opened his own electrical and radio business in Durban, but in 1928 he was appointed chief engineer of the Durban Brewery, and in 1938 of the Johannesburg Brewery, of Messrs. Ohlsson's Cape Breweries. In 1943 he was appointed Government Inspector of Factories (Engineering), and in 1946 became chief engineer in charge, Mechanical Engineering Laboratory, Witwatersrand University. He was elected a Member in 1946. He died on the 25th March 1948.

E. J. Edge (Member) was born in 1902, and served his apprenticeship with Messrs. Swan, Hunter and Wigham Richardson, Ltd., Newcastle-on-Tyne. In 1923 he became a seagoing engineer with the British India Steam Navigation Company until 1928 when he joined the Western Telegraph Company until 1932. In 1936 he joined the Eastern Telegraph Company, Singapore, sailing on the cable ship "Recorder". He was elected a Graduate in 1921, being transferred to Associate in 1923 and Member in 1932. He died on the 22nd April 1948.

R. J. M. Gibbs (Member) was born at Sunderland in 1877, and educated at Bede Collegiate School, Sunderland. He served his apprenticeship with the Scotia Engine Works, Sunderland, and served at sea in various vessels owned by Messrs. Westolls from 1897 to 1901. He gained his First Class Certificate in 1901. From 1902 to 1905 he was draughtsman on the staff of Messrs. Parsons' Steam Turbine Co., and from 1905 to 1917, as draughtsman with the North Eastern Marine Engineering Co. From 1917 to 1928 he was foreman engineer with Messrs. Glengall Iron Works, London. When this firm closed down in 1928 he continued in various engineering activities until 1933 when he joined the staff of Messrs. J. Russell and Co. on marine repairs. After a serious motor accident in 1942 he retired. From 1917 until his retirement in 1942 he was actively engaged in marine repair work on the river Thames. Since his wife died in October 1946 he had been in failing health, and for the latter six months of his life he was confined to bed. He died on the 17th April 1948. He is survived by two sons, R. M. Gibbs and B. O. Gibbs, both Members of the Institute.

T. Jefferson (Member) served his apprenticeship with Messrs. Kitson and Co., Leeds from 1900 to 1906. From 1906 to 1910 he was a draughtsman with the same company. In 1910 he was appointed chief draughtsman with the Antafogasta and Bolivia Railway until 1918 when he was appointed assistant locomotive superintendent with the Anglo-Chilian Nitrate and Railway Company. From 1923 to 1927 he was chief mechanical engineer of the Nitrate Railways, Iquique, Chile. He was a Member of the Institute of Locomotive Engineers. He was elected a Member of the Institute in 1927. He died on the 2nd April 1948.

A. Menhennet (Member) was born in 1902, and served his apprenticeship with Messrs. Harland and Wolff, Ltd. and Messrs. Dunsmuir and Jackson of Glasgow. From 1923 to 1936 he served at sea with the Asiatic Steam Navigation Company of Calcutta and the Straits Steam Shin Company of Singapore. In 1936 he was appointed power engineer of the Takuapa Tin Dredging, Ltd., Siam. Whilst in Malaya he became a Member of the Malayan Dredging Association. Just prior to the outbreak of hostilities with Japan he joined the Singapore Harbour, and on the 12th February 1942 he evacuated with the Harbour Board staff. On reaching Colombo he joined the Port Commission, but after a few months he became works manager of Messrs. Hoare and Company, Colombo. At the end of 1945 he returned to England, and in 1947 was to have joined the staff of the Moller Line (U.K.). Ltd. as superintendent engineer in Hong Kong, but before proceeding, he was sent to Portland, Oregon to superintend some repairs, and was about to leave for Hong Kong when he had a heart attack from which he died on 3rd April 1948. He was elected an Associate of the Institute in May 1938 and transferred to Member in September 1938.

A. B. Moir (Member) was born in 1885, and served his apprenticeship with the Clyde Shipbuilding Co., Port Glasgow In 1932 he was appointed engine works manager at the Taikoo Dockyard, Hong Kong. In 1942 he was interned in Amoy. He was elected a Member in 1932. He died on the 17th January 1948.