Some Considerations of Wear in Marine Gearing

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The paper deals with the design, manufacture and operation of three sets of marine double reduction gears, having a tooth loading indicated throughout by a Lloyd's K value of 120. They were also the first set of marine gears to be shaved throughout by the author's company. The first two sets of gears had a transient wear problem associated with the dedenda of the second reduction pinions and wheel which cured itself on continued operation and has now ceased. The third set was fitted with hardened and ground pinions but soft main wheel. This had no wear problem.

No proven explanation of the wear is forthcoming, and the investigations of lubrication, surface stresses, torsional vibration and manufactured profile are given for record purposes. Shaving as a post hobbing process gives accurate and consistent profiles which produce quiet gears, but as with other metal cutting processes the surface layers are stressed in the process. It is not known whether this surface stressing contributed towards the wear experienced. Running in to produce a good suitable surface finish is probably of increasing importance with more highly loaded gears which are not hardened and ground.

INTRODUCTION

The three sets of marine gears in this paper were for the vessels *Nestor, Neleus* and *Theseus* of the Blue Funnel Line. The general machinery has been referred to by Baker and Falconer[1] and in the technical press, and will not be detailed here. These vessels operate between the United Kingdom and Australia, the sea miles covered amounting to 26,000 per voyage, of which about 24,000 are steamed at approximately full speed. The maximum speeds and powers of the turbines and gears were as shown in Table I.

TABLE I.

	Power, s.h.p.	1st reduction pinion, r.p.m.	2nd reduction pinion, r.p.m.	Propeller, r.p.m.
H.P. turbine	2,330	6,000	806	112
I.P. turbine	2,760	4,500	806	112
L.P. turbine	3,160	3,000	806	112

One set erected on the test bed is shown in Fig. 1, and a general arrangement of the plant in Fig. 2. The essential difference between these gears and units of comparable powers, designed by the same team, is the increased K loading factor. The resulting reduction in size and weight is shown in Table II.

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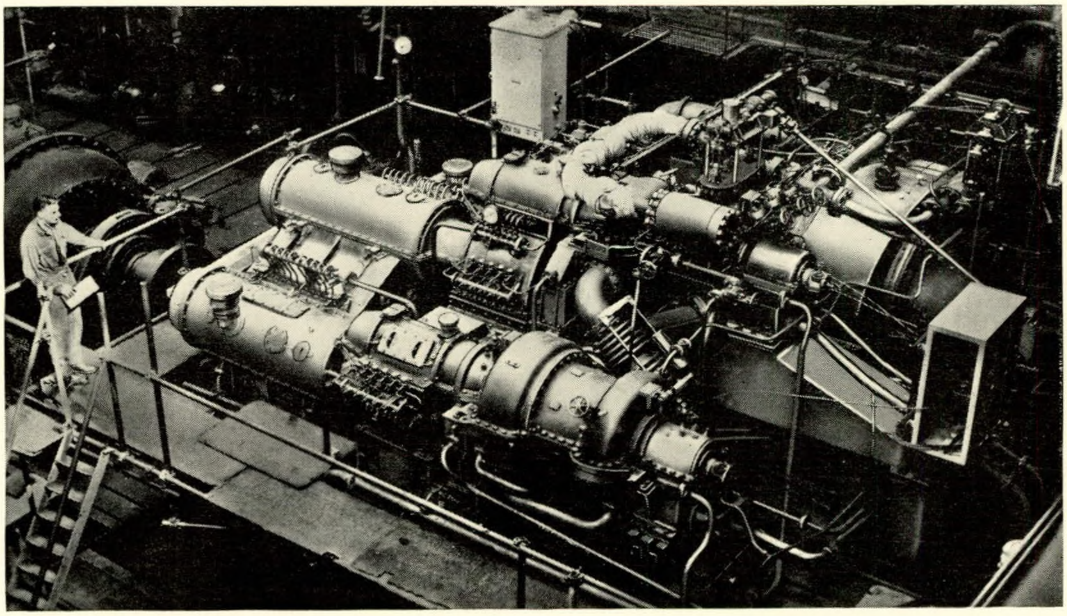


FIG. 1—*The* Nestor *propulsion machinery on test at the manufacturer's works*

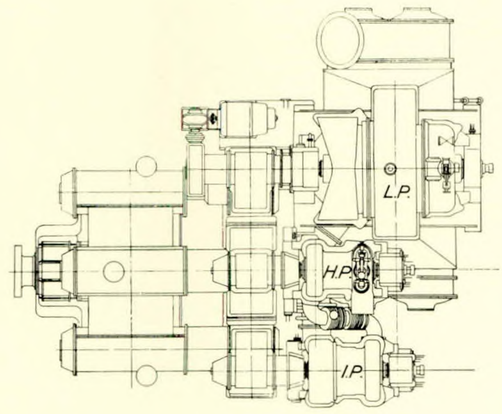


FIG. 2—*General arrangement of* Nestor *class propulsion machinery*

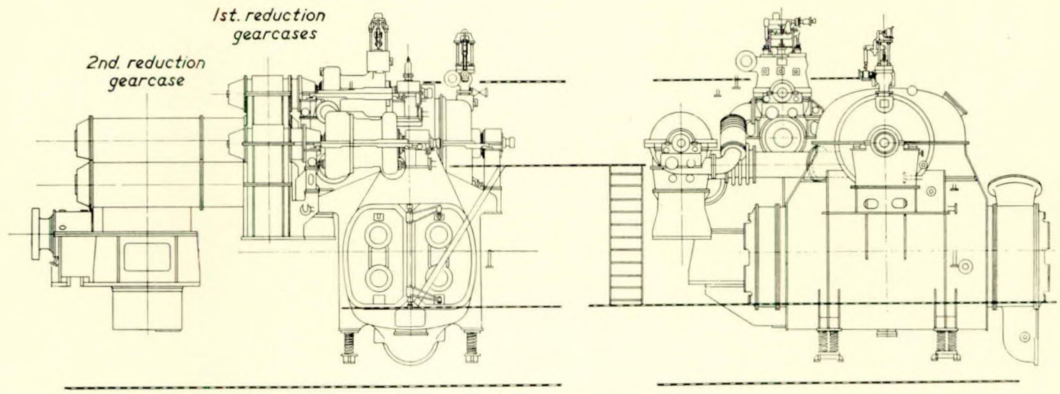


TABLE II.

Contract	Date	Approximate s.h.p.	Propeller speed, r.p.m.	1st reduction K	2nd reduction K	Weight/h.p., lb.	Vessel
A	1942	7,500	120	77·5	48	17·9	Royal Fleet Auxiliary
B	1953	8,200	109	76	69	18·3	Orthodox mercantile marine
Caledon for Blue Funnel	1948	8,000	112	119	118	12·3	Advanced mercantile marine

TABLE III.

Material	Element	U.T.S., tons/in.^{2}	Y.P., tons/in.^{2}	Hardness, Brinell	Elongation, per cent
Modified En28	Pinion	65/70	52	300—330	15
	Wheel	58/63	48	260—290	15
En39 (core)	Pinion	85	—	—	12
En39 (case)	Pinion	—	—	600—640	—

DESIGN

The material used for all pinions and wheel rims on all three sets, with the exception of the three second reduction pinions of the *Theseus*, was Modified En28—3½ per cent NiCrMo; the three *Theseus* pinions were made in En39 and case hardened. General physical properties are given in Table III.

The hobs used were as shown in Figs. 3 and 4. There were two with a normal D.P. of 5 for the first reduction elements and two with a normal D.P. of 3 for the second reduction elements. For each pitch there was a bulbous roughing hob to relieve the fillet to avoid fouling during the post-hobbing process, and a finishing hob which left material on the flanks for the shaving process only. Backlash was produced by the hob without necessity for additional cutting by sinking in further. Initially, consideration was given to a longer second reduction tooth to increase the length of contact at the expense of increased sliding, but ultimately comparable dimensions on both first and second reduction elements were used.

General design details of the rotating elements are shown in Table IV.

MANUFACTURING

Rotating Elements

The wheels were cut on the non-creep hobbing machine referred to by Newton[(2)], and the pinions on a smaller non-creep hobber by the same makers. The number of teeth in the respective master worm wheels were 528 and 180 and the feed screw pitch was 0·5 inch for both machines.

The first reduction wheels and pinions did not present any unusual difficulties during manufacture, but the second reduction wheels and pinions were rather troublesome.

Excessive wear was experienced on the roughing hobs, where the major part of the metal is removed by a few teeth at the tapered end of the hob. The hobs were then "super"

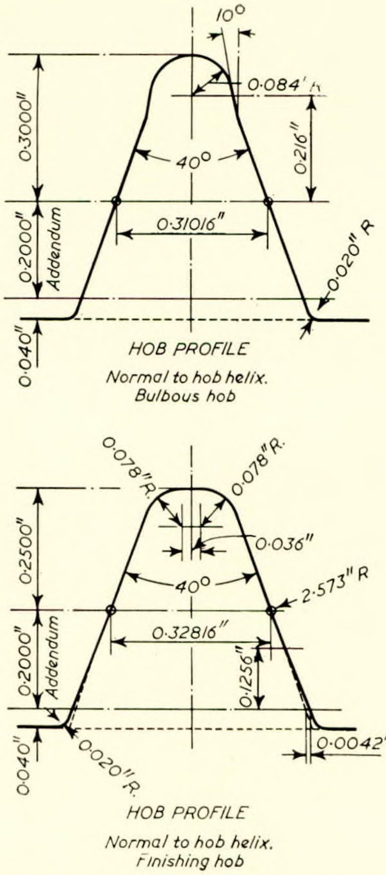


FIG. 3—*Pre-shaving hob profiles:* Nestor *class primary gears*

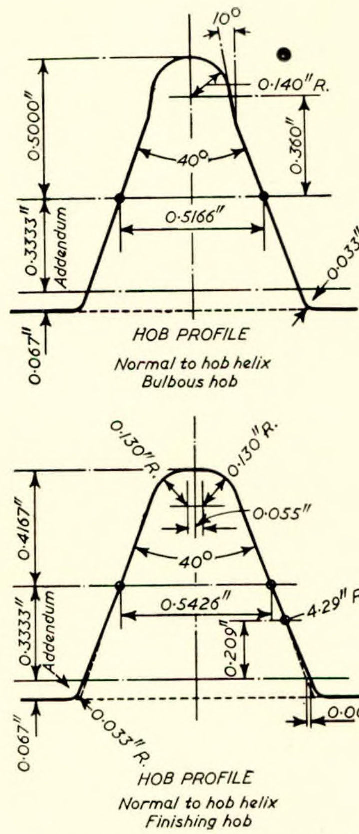


FIG. 4—*Pre-shaving hob profiles:* Nestor *class secondary gears*

hardened, but since there was some distortion and the skin formed was only about 0·002 inch thick, they could not be ground after processing. This process was not considered suitable for the finishing hobs.

Hob wear on the roughing hobs (as indicated by material removed during sharpening), could be assessed only after each cut. It varied from 0·025 inch over the tip when cutting pinions to 0·100 inch when cutting the wheel, which operation took eighty hours.

The first set of elements were cut using water soluble cutting oil, but an E.P. lubricant having sulphur additives was used for the second and third sets since it was thought that a good lubricant rather than a coolant was necessary. Whilst hob wear was not eliminated with this lubricant, it was reduced

TABLE IV.—NESTOR CLASS MAIN GEARS 8,000 S.H.P. AT 112 R.P.M.

	H.P. 1st reduction Pinion: Wheel	I.P. 1st reduction Pinion: Wheel	L.P. 1st reduction Pinion: Wheel	2nd reduction Pinion: Wheel
Gear Details				
Number of teeth	29 214	37 206	52 191	36 259
Normal pitch	5D.P.	5D.P.	5D.P.	3D.P.
Axial pitch, in.	1·2567	1·2567	1·2567	2·0944
Normal pressure angle, deg.	20	20	20	20
Helix angle (approximately), deg.	30	30	30	30
Pitch circle diameter, in.	6·697 49·421	8·545 47·573	12·009 44·109	13·856 99·688
Addendum, in.	0·200 0·200	0·200 0·200	0·200 0·200	0·333 0·333
Dedendum, in.	0·300 0·300	0·300 0·300	0·300 0·300	0·500 0·500
Centre distance, in.	28·06	28·06	28·06	56·77
Face width + gap, in.	2×6+3	2×6+3	2×6+3	2×13·5+3
Velocities, full ahead				
K value (allowing for end relief)	119	118·5	112·5	117·5
Velocity of pitch line, f.p.m.	10,400	10,000	9,300	2,910
Sliding velocity at entry, f.p.m.	1,784	1,400	992	404
Sliding velocity leaving, f.p.m.	1,599	1,285	938	368
Rolling velocity at entry, f.p.m.	2,466 4,250	2,707 4,107	2,823 3,815	772 1,176
Rolling velocity leaving, f.p.m.	5,445 3,846	4,983 3,698	4,341 3,403	1,451 1,083
*Specific sliding at entry	—0·723+0·420	—0·517+0·340	—0·351+0·261	—0·533+0·344
Specific sliding leaving	+0·294—0·416	+0·258—0·347	+0·216—0·276	+0·254—0·341

$$\text{*Specific sliding} = \frac{\text{Sliding velocity}}{\text{Rolling velocity}}$$

× *1,050*

FIG. 5*(a)*—*Reflection electron micrograph: hobbed gear*

× *1,050*

FIG. 5*(b)*—*Reflection electron micrograph: lightly shaved gear*

to 0·030 inch on the wheel. This problem of hob wear has never been satisfactorily solved, and the author considers that under present commercial conditions the maximum U.T.S. of the gear material which can be satisfactorily hobbed for large gears is limited by the hob materials available, to a figure of about 70 tons per sq. in. (Brinell, about 330). This problem may be eased by high speed hobbing or alternative cutting lubricants, but to date these have not shown any further improvements for large gears.

The wheels were shaved on a shaving machine constructed by the author's company, having a capacity for wheels up to 156 inches diameter. The machine is arranged for crossed-axis shaving, using either both cutter faces by "crowding in" or a single cutter face by loading it, using an electrical generator and resistances.

× *1,050*

FIG. 5*(c)*—*Reflection electron micrograph: heavily shaved gear*

The shaving cutters, which were involute in form with no tip correction, were arranged for 15 degrees between work and cutter axes. On set 1, the method used was to feed the cutter and work into mesh radially until zero backlash was obtained, after which the work was revolved. The cutter was then fed into mesh by a further 0·001 inch (producing a cut of about 0·0003 inch on each flank), after which the cutter was traversed across the wheel. Surface tearing occurred during the first cut and surface quality further deteriorated with each successive cut. Cutter examination indicated minute metal particles welded to the cutter flank, which were attributed to lubricant failure. The neat lard oil lubricant was changed to light lubricating oil, but this resulted in such complete failure of the surface that the wheel had to be recut to restore the geometry and surface finish, and ultimately an E.P. lubricant with sulphur additions had to be used. With this lubricant, although a slight deterioration in surface finish still occurred, the process was considered satisfactory enough to enable the

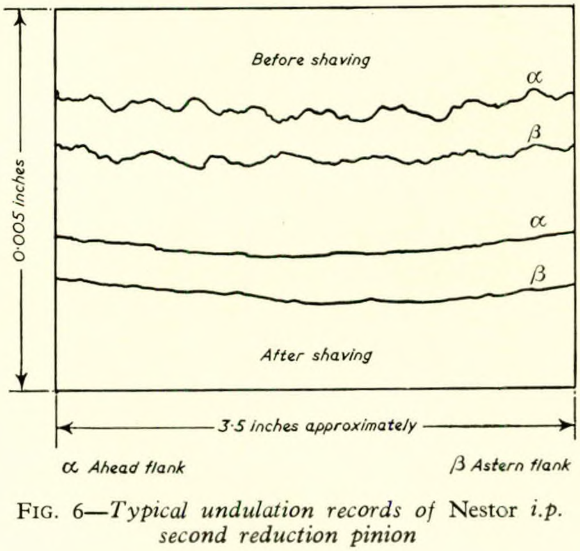


FIG. 6—*Typical undulation records of* Nestor *i.p. second reduction pinion*

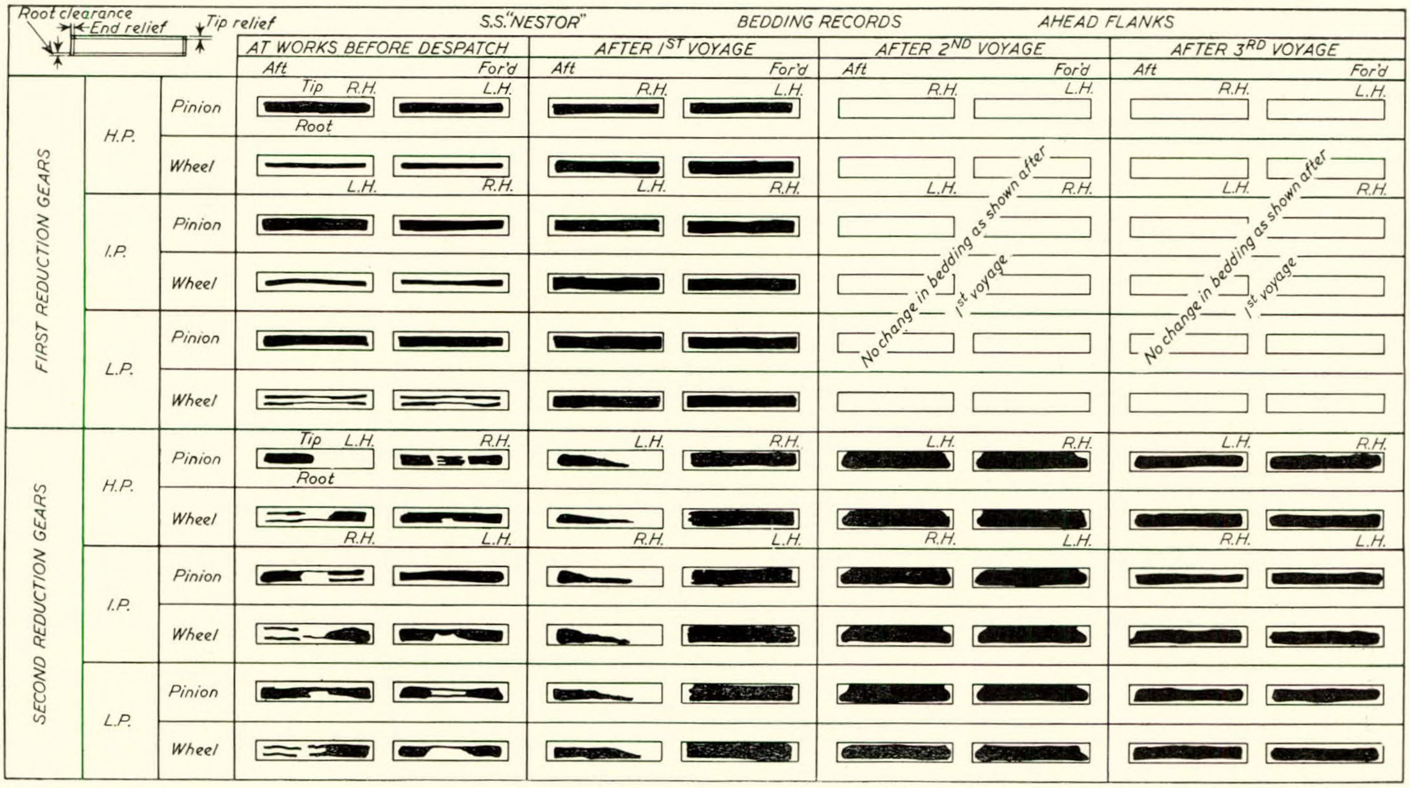


FIG. 7—*Record of bedding after service: s.s.* Nestor

remaining elements to be shaved. Some idea of hobbed and shaved surfaces viewed, using methods described by Haliday[3], are shown in Fig. 5. These only view a very small area and are not necessarily representative of the whole surface.

Helix correction by selective shaving to match wheels and pinions was applied, using the "crowding in" method. This was found to be a very "hit-and-miss" process, particularly at the ends of the teeth, where cutter run-out produced a variation in metal removal and hence helix angle and tooth thickness. This and gear case distortion probably contributed to produce the heavy end bedding which was corrected by hand work on set 1.

On the second set (for the *Neleus*), friction brake shaving on one flank only was used initially with backlash between the cutter and work. This was very promising, but the brake heated up the whole cutter head causing the torque load to fluctuate, so that cutter tooth deflexion and hence finished tooth profile were variable. Finally, an electrical loading method, with heat dissipation in external resistances, was used (Hadcroft and Towle[4]), and a fine control of cutter load obtained at any speed. This method was used to shave all gears for both *Neleus* and *Theseus*, with the exception of the second reduction pinions for the latter ship.

The shaving process was not only very successful in

TABLE V.—TYPICAL MEASUREMENTS: NESTOR CLASS GEARS

	H.P. 1st reduction Pinion: Wheel		I.P. 1st reduction Pinion: Wheel		L.P. 1st reduction Pinion: Wheel		2nd reduction Pinion: Wheel	
Adjacent pitch error, in. After shaving	0·0001	0·0001	0·0001	0·0001	0·0001	0·0001	0·0001	0·0001
Cumulative pitch error, in. After shaving	0·00015	0·0007	0·0002	0·0008	0·0002	0·0008	0·0002	0·0017
Undulations, in. Before shaving After shaving	0·0003 0·0001	0·0002 0·0001	0·0003 0·0001	0·0002 0·0001	0·0003 0·0001	0·0002 0·0001	0·0003 0·0001	0·00025 0·0001
Surface finish, C.L.A, in. $\times 10^{-6}$ Before shaving After shaving	16 21	11 20	13 17	12 18	13 15	12 15	20 20	20 21
Surface finish, C.L.A, in. $\times 10^{-6}$ After 3rd voyage addendum After 3rd voyage dedendum	26 26						50 98	48 130
Surface finish, C.L.A, in. $\times 10^{-6}$ After 4th voyage addendum After 4th voyage dedendum	36 36						50 105	50 106

C.L.A. = Centre-line average.

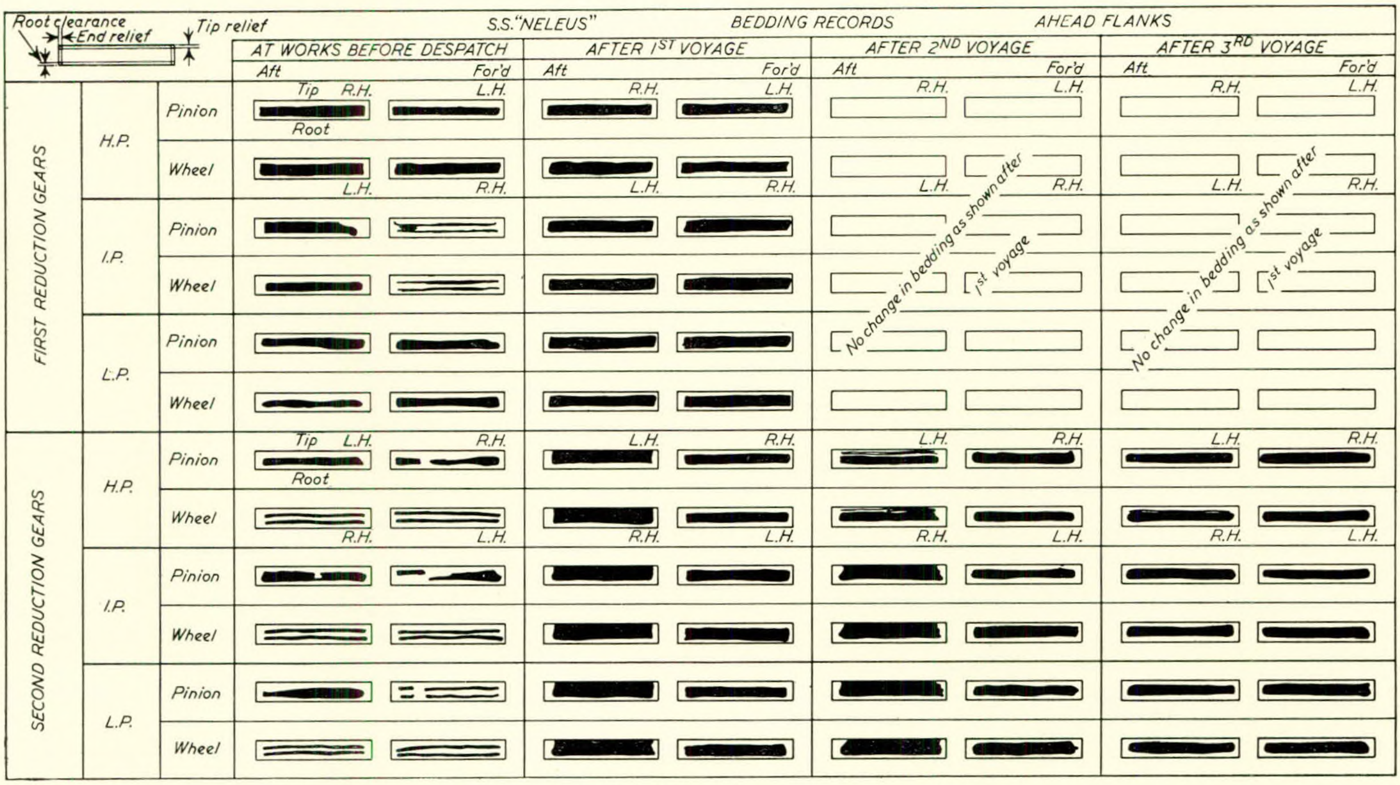


FIG. 8—*Record of bedding after service: s.s.* Neleus

removing undulations, and correcting profile, but was also excellent on sets 2 and 3 for matching the beds of wheels and pinions. Some final inspection records are given in Fig. 6 and Table V.

The case hardened second reduction pinions for the *Theseus,* which were geometrically identical with those supplied for the earlier ships, were ground by Vickers-Armstrongs, Ltd., and The Fairfield Shipbuilding and Engineering Co., Ltd., who undertook this work at short notice; and their help is gratefully acknowledged. The main wheel for this set was

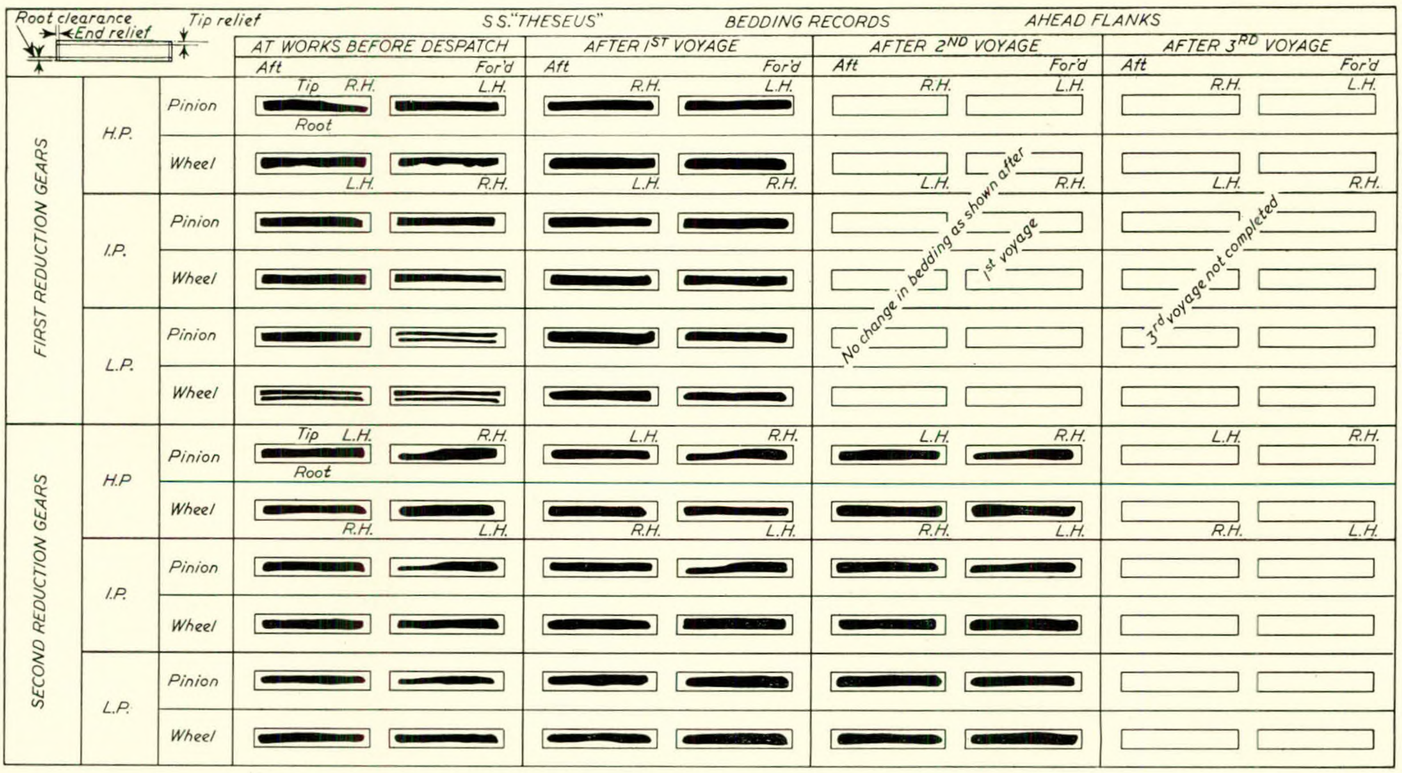


FIG. 9—*Record of bedding after service: s.s.* Theseus

shaved and axial pitch measurements taken. The three pinions were then ground to match as closely as possible, after which the main wheel was selectively shaved to obtain the best possible bed. Bedding records shown in Fig. 9 indicate the success of this method.

Gear Case Line-out

To ensure repeatability of gear case line-out on test and site for both *Nestor* and *Neleus,* an optical method, similar to that referred to by Deakin[5] for gear case joints, was adopted. A number of brackets were dowelled round the gear case, and a rotating table mounted on the top of the gear case to carry the telescope. By mounting three numbered columns on three numbered brackets, and sighting the telescope on to a single illuminated target placed in turn on top of each of the three columns, the telescope mounting screws could be adjusted to give a constant reading on all three columns. This established an optical datum surface, and by moving a single column in turn round all the brackets fitted to the casing and reading the target indicator, the vertical relationship of the various brackets to this surface could be established.

For the *Theseus,* the line-out was controlled by this method from the initial boring operation, through the complete manufacture of the gear case, and on test and site.

Bedding Records

A record of the bedding due to load running, for the three sets, is shown in Figs. 7, 8 and 9. This bedding was obtained by lightly painting all pinion teeth and bands of wheel teeth with Talbot Blue, and rotating the gears.

OPERATING EXPERIENCE

Nestor

The *Nestor* has now completed seven voyages.

The engines were installed at Dundee but no opportunity occurred for inspection of the gear bedding until berthing at Glasgow, when the first reduction elements were found to be very satisfactory; the second reduction elements had a slight amount of pitting on the aft end of the aft helix of both i.p. pinion and wheel. Local high spots were relieved by hand scraping the i.p. pinion only at Liverpool, prior to the maiden voyage.

After the first voyage, the ahead flank of the forward helix was ridged along the pitch line with a polished addendum and a matt dedendum. The aft half of the aft helix was similar, but the remaining part of the aft helix was polished, with normal bedding. The tip relief on all pinions appeared to be reduced somewhat but that of the wheel was untouched.

× 80

FIG. 11—*A particle suspended in ethylene glycol showing the clean cut nature of its surface*

There was a considerable amount of grey dust collected from the magnetic filter in the lubricating system, which was analysed and photographed (see Figs. 10 and 11). It was shown to be metal from the gear teeth and appeared to consist of flakes of varying sizes all about 0·0001 inch thick. The initial pitting had increased very slightly but was not progressing at any serious rate.

The usual visual bedding observations were considered inadequate in this case and more accurate information of both profile and surface finish was obviously required if the wear of the gears was to be investigated properly.

A considerable background of knowledge was available on plastic replicas for determining surface finish (Sawyer and McGubbin[6]), but there was a lack of precision apparatus for measuring profile of gears such as these second reduction elements in the gear case. Experiments, using resins or graphite and sulphur mixtures, to duplicate tooth form and finish, indicated a tendency on the part of the materials used to shrink considerably on setting, but ultimately a sulphur-graphite

× 400

FIG. 10—*Particles mounted in bakelite*

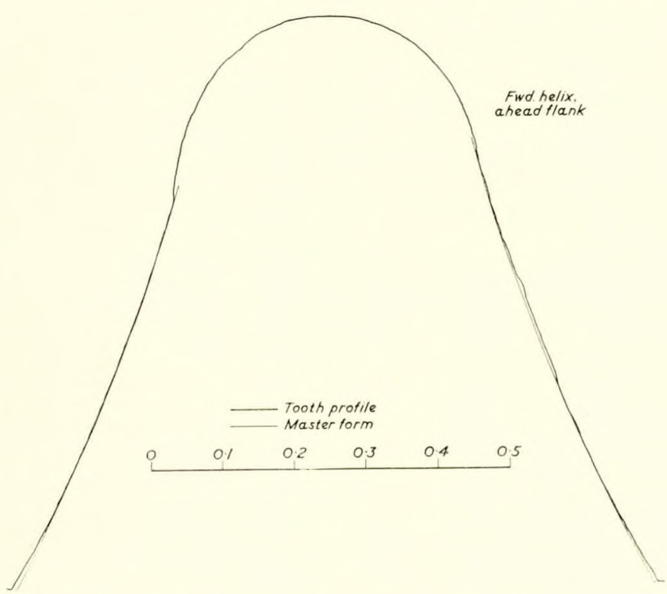


FIG. 12*(a)*—*Inspection record: i.p. pinion after first voyage*

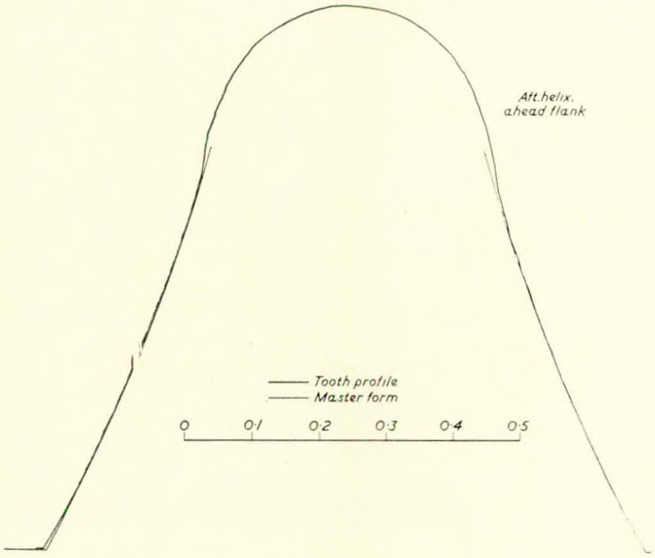


FIG. 12(*b*)—*Inspection record: i.p. pinion after first voyage*

mixture was made, which reduced the shrinkage to about 0·001 inch and was adequate for surface finish reproduction. A later development, using a metal core with a thin resin surround, gave the true form with negligible shrinkage. The moulds taken were pantographed and traced for comparison with a 25-times full size theoretical profile. The original profile was shown to be correctly involute within the accuracy of the method (believed to $\pm$ 0·0005 inch) by recording the astern flanks of the h.p. and i.p. pinions, since these transmitted no load and were unmarked. Some typical results, taken after the first voyage, are shown in Fig. 12. Fig. 13 shows an alternative method of plotting for easier comparison after each voyage. Tip relief was restored as well as possible by hand. It should be noted that all first reduction elements were in excellent condition.

After the second voyage the ridge on the second reduction elements had progressed so that it was uniform across both helices of all second reduction pinions and wheels, and profiles

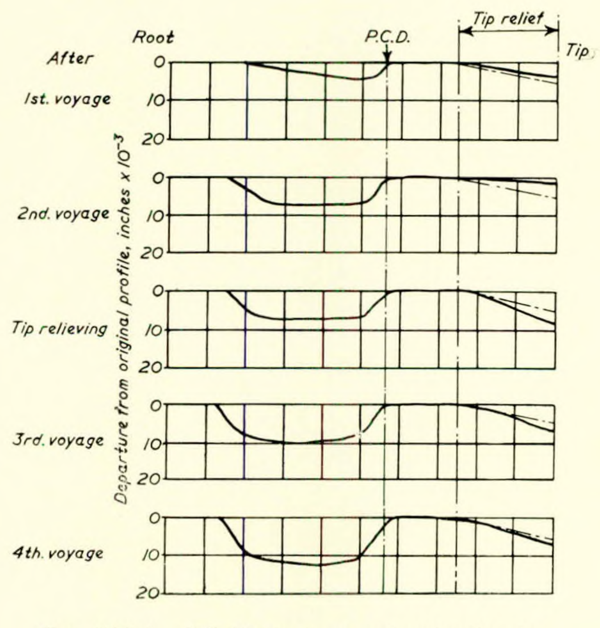


FIG. 13(*a*)—*S.S.* Nestor: *second reduction i.p. forward helix, ahead flank*

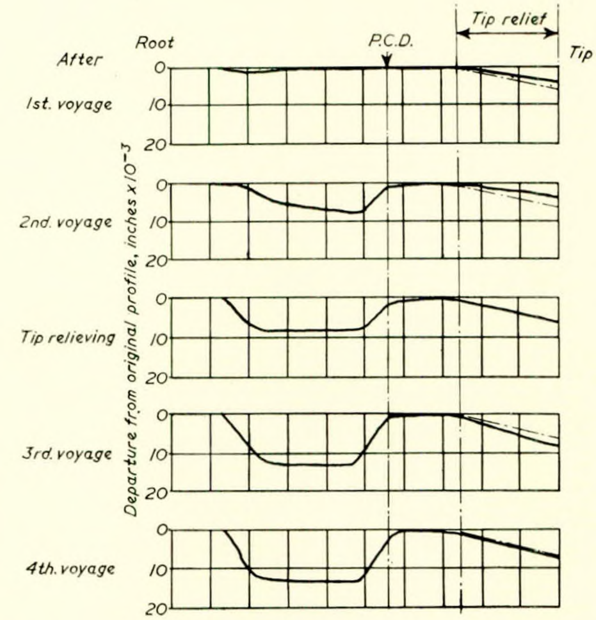


FIG. 13(*b*)—*S.S.* Nestor: *second reduction i.p. aft helix, ahead flank*

indicated further wear, with again slight loss from tip relief (Fig. 14). First reduction elements were still excellent. An impression of the teeth at this stage can be obtained from Fig. 15.

After the third voyage there was no change in appearance despite increased dedendum wear, and the tip relief was unaffected.

Fig. 16 shows the variation of wear to date. It can be seen that on this basis of measurement the wear has ceased. There was from time to time some doubt (due to the apparent loss of tip relief on the first two voyages) whether the addendum portions, although polished, were also wearing.

A check was made on tooth thickness for the h.p. and i.p. pinions, using a tooth thickness vernier gauge (see Fig. 17). It can be seen to confirm the pantograph results. No measurable wear was shown on the addendum, which indicated that it (if it existed) was certainly less than 0·0005 inch. It is interesting to note that the h.p. and l.p. pinions now show some signs of polishing on the hitherto matt dedendum. The

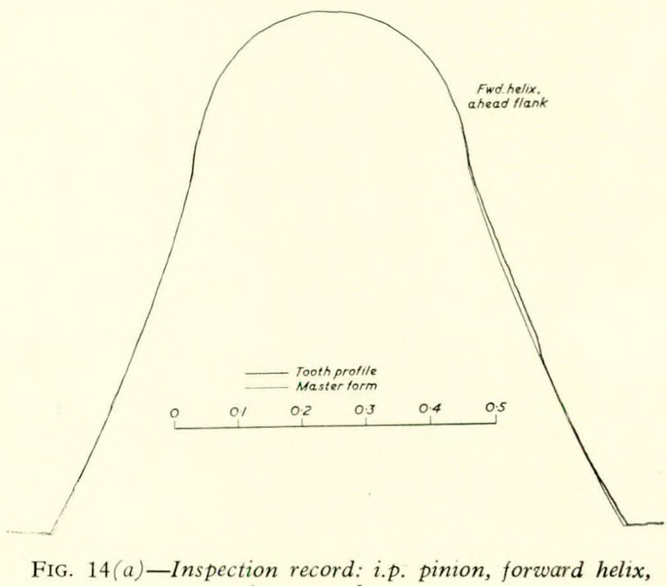


FIG. 14(*a*)—*Inspection record: i.p. pinion, forward helix, after second voyage*

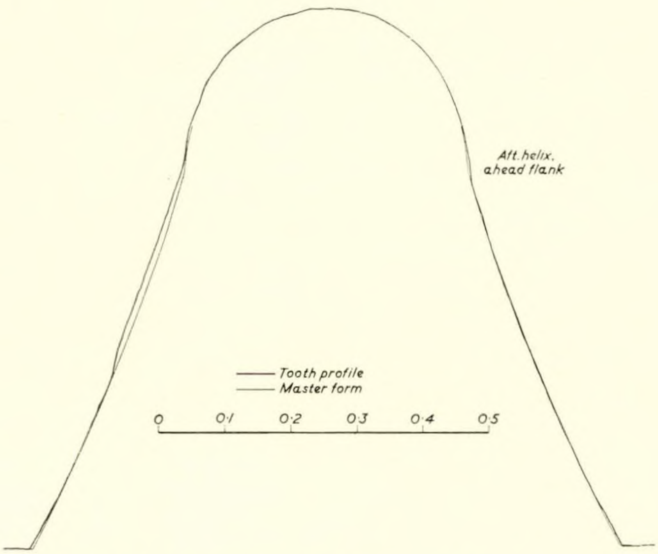


FIG. 14*(b)*—*Inspection record: i.p. pinion, after helix, after second voyage*

i.p. pinion dedendum is still matt as if this were the only surface now liable to wear.

Neleus

The *Neleus* has now completed five voyages and the wear of the gears has followed a similar pattern to that of the *Nestor,* and has now also ceased (see Fig. 16). In this gear case better bedding was obtained on the test bed due to improved shaving technique, and no subsequent hand correction was necessary. The writer is certain that this is why pitting on this set has been negligible. One difference which has not been explained was that the ridging at the end of the first voyage was on the after helix and the forward part of the forward helix, and not,

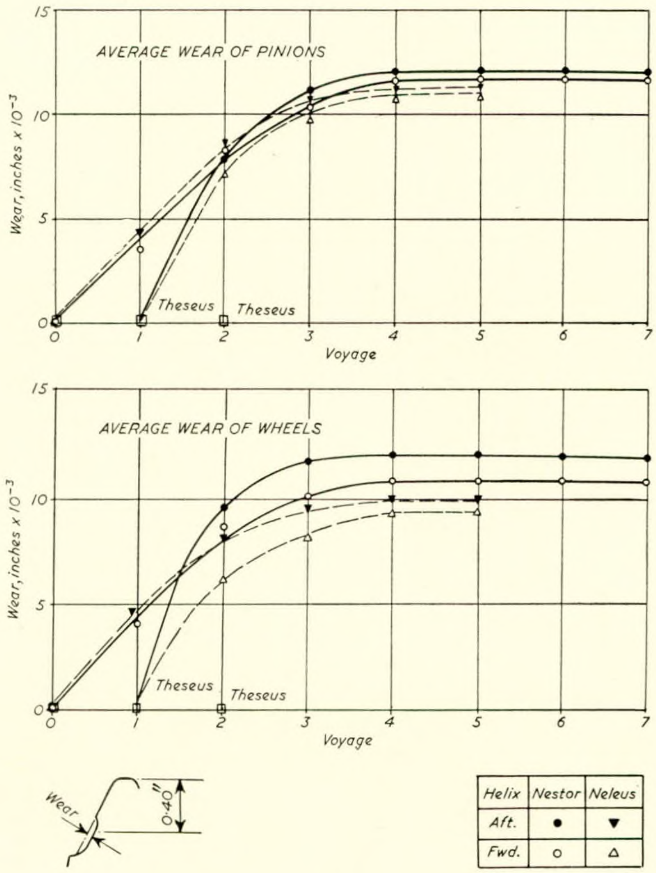


FIG. 16—*Wear of second reduction gear*

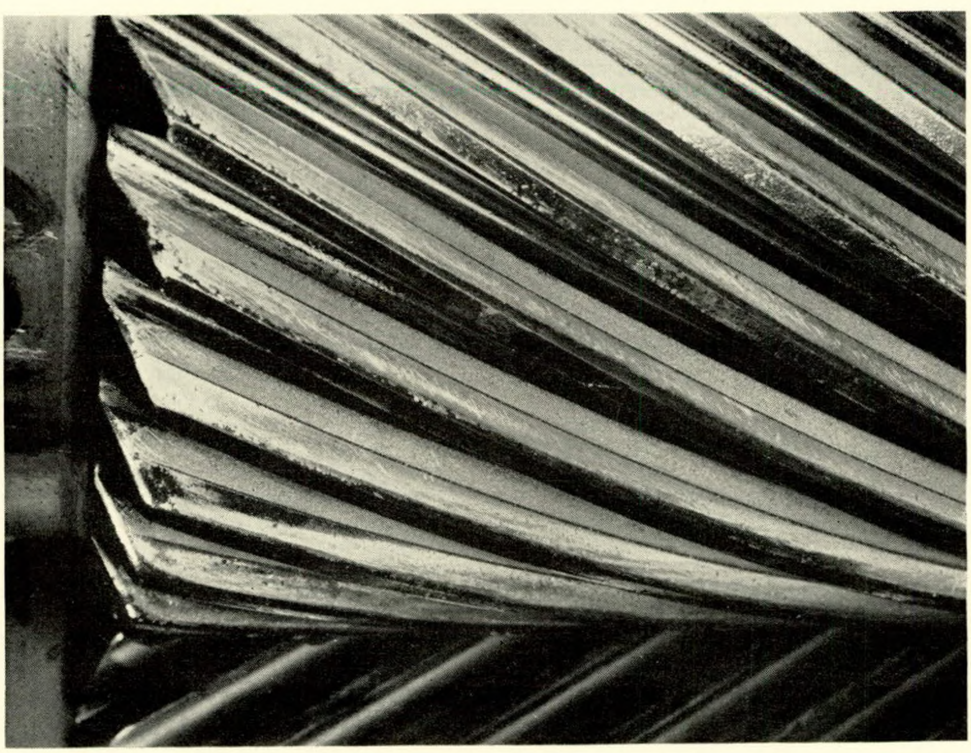


FIG. 15

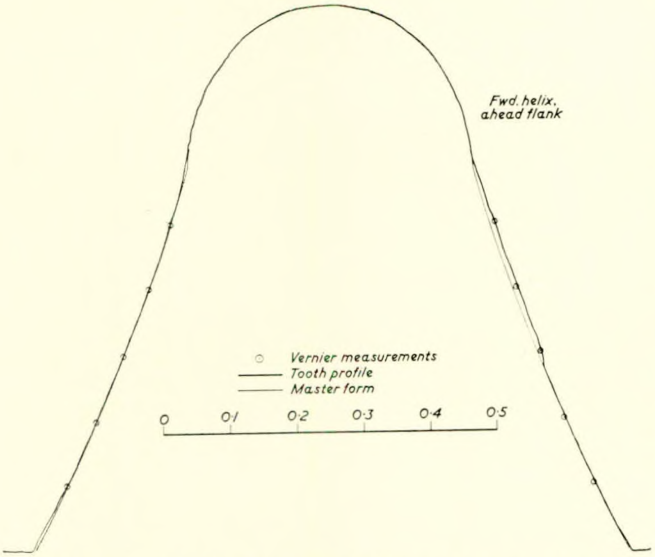


FIG. 17—*Tooth profile with Vernier confirmation, after second voyage*

as in the case of the *Nestor,* on the forward helix and the aft part of the aft helix. H.P. and i.p. tooth couplings between the pinions and their corresponding flexible shafts on both vessels, have been examined and found unmarked.

Theseus

At the time of manufacture of the *Theseus* gears the *Nestor* and *Neleus* were making their early voyages, and rapid wear was occurring. There was then no indication that the wear rate would decrease, and to increase the wear resistance of the second reduction pinions of the third ship it was decided to make them hardened and ground but identical to the other sets in all other respects.

When the gears were examined after the ship's second voyage, which was shortly before the completion of this paper, they were found to be in excellent condition on both the first and second reduction units. The bedding of the secondary pinions and wheel was continuous over both helices and over the full flank, except that the depth of the bedding on the forward helices of the i.p. and l.p. members was reduced to about 60 per cent over the after 40 per cent of their length. The filters contained no discernible detritus from the gears.

Noise Level

For all three sets of gears the noise level was low. A typical noise spectrum obtained on board the *Theseus,* which because of the hardened and ground elements was probably slightly noisier than the first two sets, is given in Table VI.

TABLE VI.

Frequency, c/s	Sound pressure level, db above 0·0002 dynes per sq. cm.	Phons	Probable source
7.0			Propeller blades
15	102	—	Propeller blades × 2 Main shaft r.p.s. × 8
34	80	53	Propeller blades × 5 Main shaft r.p.s. ×20
467	88	85	Main wheel contact
930	91	91	,, ,, ,, ×2
1,400	92	93	,, ,, ,, ×3
2,600	98	110	I.P. contact
3,450	85	91	
5,600	83	80	

The conversion from Sound Pressure Level to Phons has been made according to the M-V Equal Loudness Contours.

WEAR INVESTIGATION

General Remarks

In the early stages when both pitting and wear were occurring on the *Nestor,* these were assumed to be separate problems. The pitting was mild, involving a small number of large pits which progressed in a manner expected by experience; since the bed was improved by hand scraping, the spread of pits slowed down and finally ceased. Confirmation of this assumption was obtained from the *Neleus,* where selective shaving produced an adequate initial bed, and pitting was negligible.

A gradual change of profile after some years' operation, due to relatively more wear from the dedendum than the addendum, had been experienced in the past with marine installations, and was demonstrated experimentally by Bailey[7] in 1927. In all such cases, however, the mating surfaces had been polished over the whole tooth flank and some wear, although admittedly very small, had occurred on the addenda.

The features of the present problem, which had not been encountered before, were:—

(*a*) Rapid wear occurred on two sets of machinery between all second reduction pinions and the main wheel, while the three first reduction elements were unaffected, although materials and method of lubrication were common to both.

(*b*) Rapid wear occurred on two sets where the mating surfaces were En28, but not on one set where the mating surfaces were En28 and En39 case hardened.

(*c*) Rapid wear occurred in both dedenda but neither addenda of mating teeth.

(*d*) Wear progressed rapidly initially but ultimately changed to a negligible rate.

(*e*) Detritus from the wear process was in the form of dust and flakes rather than pyramidal pieces due to pitting or lumps due to scuffing.

Design aspects which it was felt required investigation, were lubrication, material condition and wear resistance, accuracy of manufacture and surface loading. Any one of these is sufficient to occupy a large research and development team for a long period of time and the notes which follow under their various headings are not meant to replace the long term experimental programmes which have been and are going on in certain of these fields. They are given rather to indicate the work carried out by, or for, the author in connexion with these somewhat more than usually highly loaded gears to obtain guidance for the designer until a more complete and proven case explaining the wear is presented.

Lubrication between Gear Teeth and Comparable Discs

The lubricants used for all the gears were typical proprietary oils, representative of normal marine practice. The failure of the dedenda was not recognizable as either pitting or scuffing of the usual type. The surface appeared to be breaking up into dust, although photographs (see Fig. 10) showed this dust to consist of flakes. It was difficult to visualize failure of one of two identical mating surfaces by the flaking away of surface layers, while the other surface was untouched, if oil was absent from the contact. It did not, therefore, seem to indicate a complete breakdown of lubrication due to failure of either the method of lubrication (which had proved satisfactory over a wide range of applications), or the type of oil (although consideration was given by the author to the adoption of oils with E.P. additions). Again, since the wear was initially greatest near to the pitch line, where the sliding was small, scuffing did not appear to be likely. It was at one time considered that the failure might be caused by rapid microscopic pitting due to local overloads, since the detritus looked dusty. Photographic analysis, however, showed no indications of the recognizable pyramidal shape characteristic of pitting detritus.

Several investigators have developed theoretical expressions to determine the limiting conditions for hydrodynamic lubrication between mating gear teeth.

In most cases the problems have been simplified by considering the teeth as two discs having the same radii of curvature and velocities. With helical teeth these quantities vary along the line of contact. The figures calculated below are applicable for the pitch point only, where relative sliding is zero. Experimental verification of this theoretical work is difficult, since the oil is usually contaminated with air or detritus and the surface finishes used in practice may be of the same order as the calculated film thickness. Cameron[8] has developed expressions for rigid discs with constant viscosity oil; Cameron[8] and McEwen[9] have considered rigid discs with oil viscosity varying with pressure; and Lewicki[10] has considered discs deformed by the loading with constant viscosity oil.

Some values of film thickness calculated from these sources are given in Table VII for two oil temperatures and both first and second reduction gear conditions, together with the C.L.A. value of the surface finish of the mating surfaces, all values being in micro-inches.

pitted, and the dedendum was pitted but not scuffed. If this is parallel to the existing case, then the present addendum is apparently not heavily loaded enough for scuffing, while the dedendum should be pitting, although lubrication may be adequate. Way[12] investigated pitting of steel rollers, and concluded that the presence of tiny cracks, having a certain restricted direction in the surface, was a fundamental requirement for pitting when associated with oil of the right viscosity. There were no cracks in any of the pinions as manufactured, or when operating, or in the test blanks mentioned elsewhere. This, coupled with the lack of recognizable pitting detritus, seems to throw doubt on pitting as an explanation of the failure, unless it was of an unusual type.

Thus, while the first reduction elements are more likely to maintain a lubricating film than the second reduction elements, there is not sufficient evidence to indicate whether lubrication did, in fact, fail and permit surface contact, introducing a metal-to-metal wear problem, or whether the surface condition was such that fatigue failures of surface asperities produced the detritus while lubrication was still maintained.

TABLE VII

Case		First reduction	I.P.	Second reduction		Remarks
	Temperature, deg. F.	110	220	110	220	
1	Rigid discs	65	6	18	2	Viscosity constant. Maximum steady load.
2	Rigid discs	226	34	110	16	Viscosity varies with pressure.
3	Deformed discs	41	13	28	9	Viscosity constant.
	Surface finish, C.L.A., in. $\times 10^{-3}$					
	Pinion	13		20		
	Wheel	17		20		

The variation of film thickness with pressure, temperature, and the different mathematical approaches renders it difficult for the designer to calculate with any degree of certainty whether or not hydrodynamic lubrication is, in fact, existing under any particular conditions of loading, velocities and surface finish. He requires a workable formula for oil film thickness, including the effect of sliding, variation of oil properties, and surface finish. Until this is available his best guide is the measurement of film thickness experimentally at the contact zone for known loadings, velocities and surface finishes on two discs representing the surfaces under consideration. This in itself is a difficult problem because, for the same set of loads and velocities, the effect of surface finish on indicated film thickness requires special study. So far as the author is aware, all work on film thickness measurements has been carried out using specially prepared surfaces, with surface finish measurements well removed from attainment during normal gear manufacture, but which may occur after a long careful "running in" period.

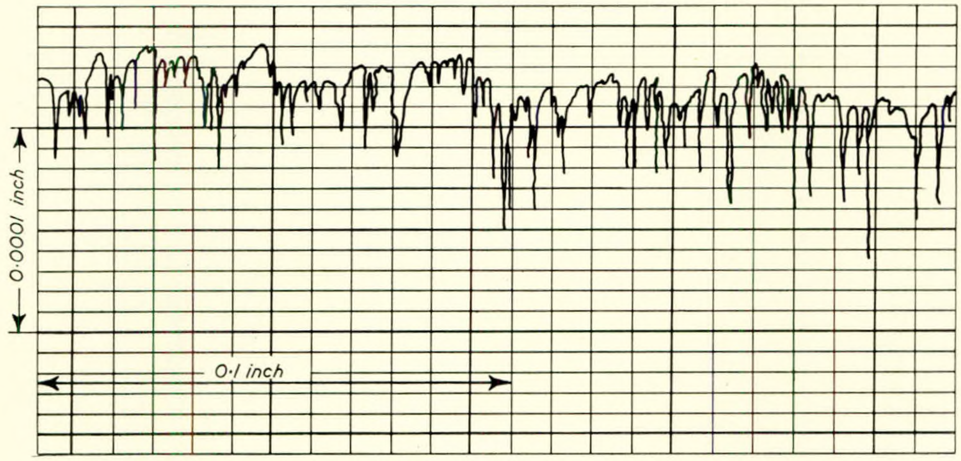
It is possible from the calculated figures above to say that for the given surface finish readings, the second reduction elements are more liable to fail due to lack of lubrication than are the first reduction.

In addition to the problem of maintaining a film of oil the motion of the mating surfaces about that film also plays an important part in gear tooth behaviour. Bailey[7] refers to this and Tuplin[11] points out that on a helical gear tooth contact of the type considered, the slide-roll, or slide-sweep ratio, also known as specific sliding, is between zero and plus one on both the addenda, and zero and minus one on both the dedenda, where the positive sign indicates sliding in the same direction as the passage of the contact point over the surface. He demonstrates that negative values are best for lubrication since they ensure the cleanest, coldest oil to the contact zone. They do not, however, affect the residual oil film which is always present on any gear tooth, and whose contamination and temperature are affected by past contacts. It is interesting to note Tuplin's reference to a helical gear, which indicated a difference in lubricating conditions above and below the pitch line. The addendum was scuffed but not

Material Combinations

The high U.T.S. material combination used had also been used successfully on the first and second reduction elements of the gear box for the British Railways Turbomotive, where the maximum peripheral speeds/K loadings, ignoring starting conditions, were 395ft. per sec./73 and 197ft. per sec./91 respectively, although these values did not occur together. In the design stage, rig test experience at the lower speeds and higher surface loadings required in this present case (174ft. per sec./120 and 48ft. per sec./120) was not available, but as soon as wear commenced, a programme of tests was initiated on a 6in. D.B.S. disc machine with this and other material combinations. The initial test loadings used were very high (K = 500-600) to obtain an accelerated test: a brief résumé of certain of the tests carried out is given in Appendix I. The test conditions are too far removed from those of the gears to permit more than general observations. Longer term tests now proceeding at loads comparable with the gears will be reported in due course. The present tests show a significant difference in performance when a disc of En28 is mating with En28 and when it is mating with a carbon steel. This does not prove that carbon steel would be satisfactory in the present case, but only that the performance of different material combinations cannot be forecast for any operating conditions, and needs approval by experience or test.

The tests show a definite connexion between permissible loading and surface finish, since, given care with the "running in", it was possible to operate the discs without scuffing up to the highest loading, with the worst combination of materials. This seems at first sight to indicate that there is no unique scuffing load attributable to any pair of materials. It may be that different materials fail because of different surface conditions produced by the same manufacturing process. Surface finish measurements on the discs which had been run-in show a profile with most of the asperities worn away (see Fig. 18). It is suggested that the altered surface conditions probably allow a thinner oil film to operate satisfactorily since the



(a) After running in

(b) Before running in

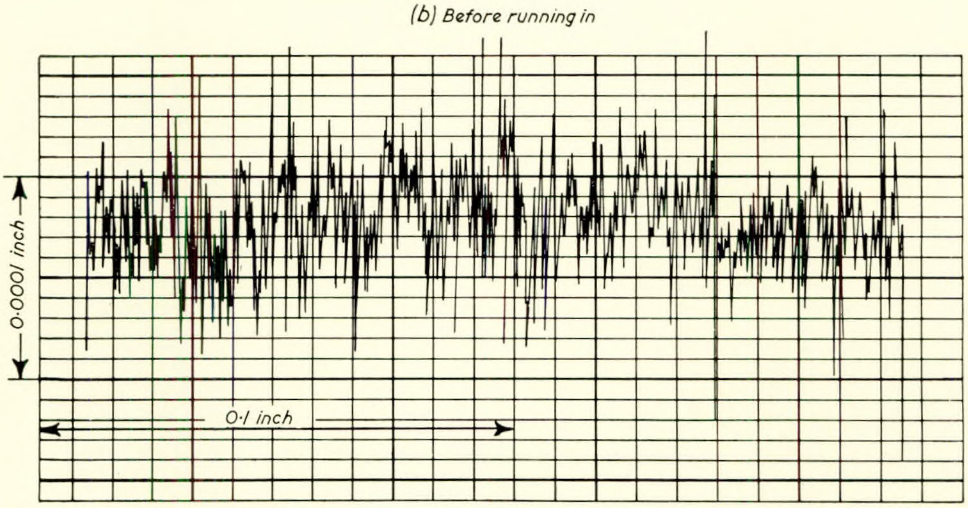


FIG. 18—*Surface finish records of discs on a 6-in. machine*

asperities are not so likely to pierce it. The behaviour of different materials, when contact is made only by asperities, and not by the main bulk of the material, may be the guide to explain the behaviour of different material combinations.

Stress Condition of the Surface Layers, Including That Due to Manufacturing Processes

There appears to be little or no information published on the surface stresses in a material which has been subjected to either hobbing alone or hobbing plus shaving. To determine the surface stress conditions a dummy pinion was made from six 1in. slabs of En28 bolted together on a shaft centre and machined to represent one helix of the second reduction pinions. This was hobbed in the normal way, using the best lubricant, after which two slabs were removed; it was then lightly shaved corresponding to current practice with this process and two further slabs removed; the remainder was then heavily shaved to simulate malpractice with this process and the final two removed (see Fig. 19).

A recent examination of surface stresses due to grinding and milling had been carried out with small square plate specimens, and this was extended to cover thin, approximately flat, plate sections, cut from the tooth surfaces on the above slabs. For details of the method used see Appendix III. The results are given in Fig. 20.

As a check (since the metal was not of uniform thickness) the surfaces were examined by X-ray diffraction methods, and close agreement on depth of affected layers was obtained between the two methods.

The hobbed specimens had some indeterminate stresses

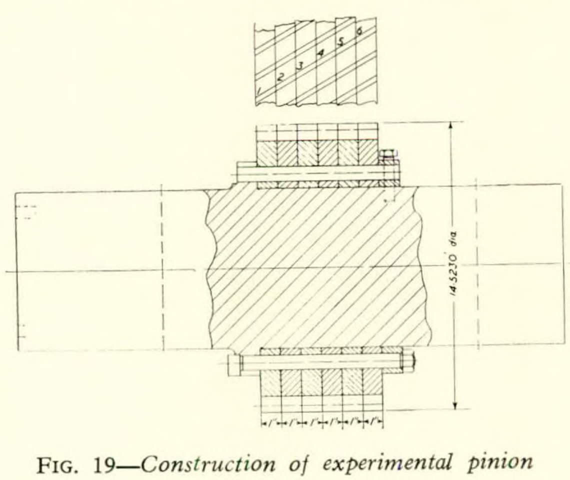


FIG. 19—*Construction of experimental pinion*

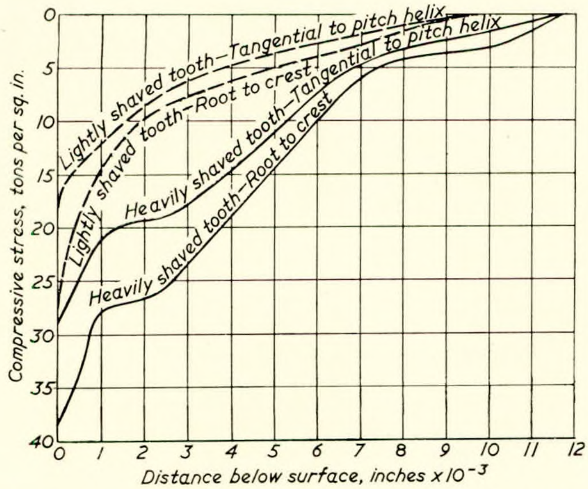
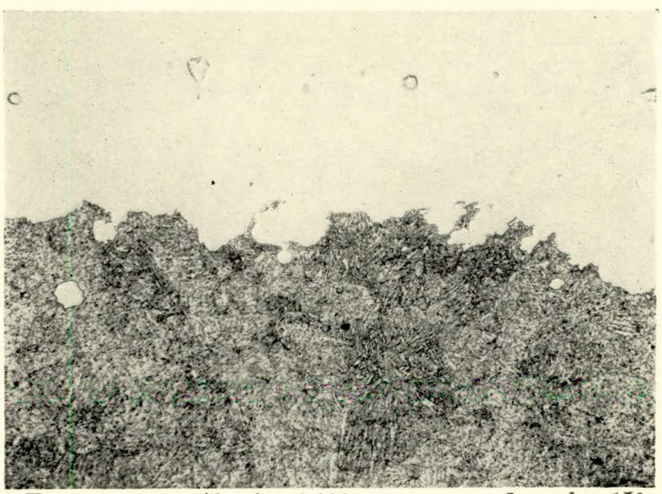
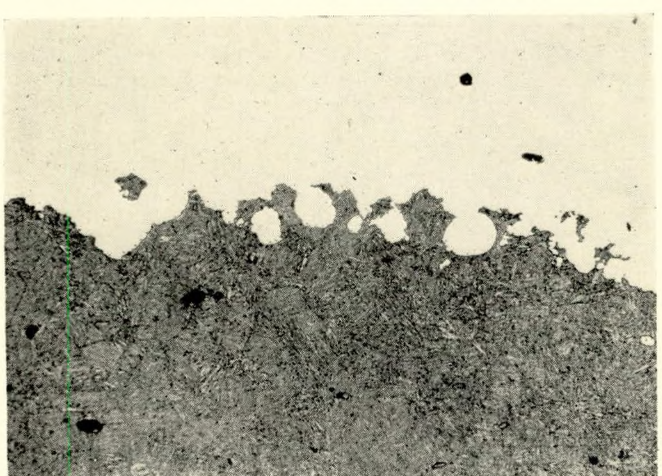


FIG. 20—*Residual stress measurement, experimental pinion: all stresses measured at centre of tooth surface*

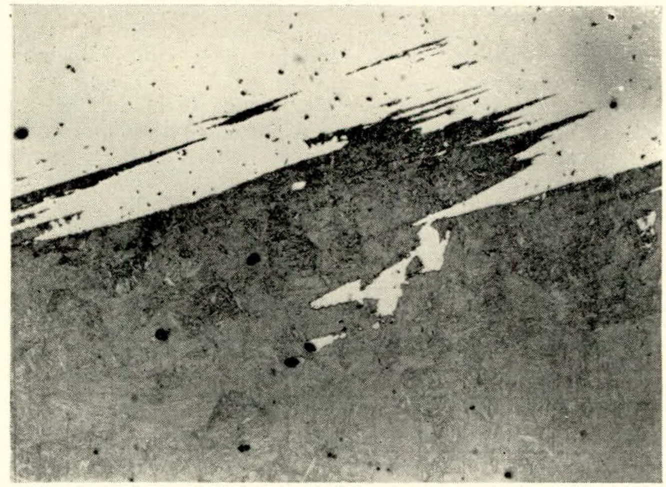
in a very thin surface layer of the order 10^{-4} inch and negligible stresses over the remaining range of surface depth examined. The stress values for the very thin layers on the surfaces were



Transverse magnification 3,000 *Lateral × 150*
FIG. 21*(a)—1:20 taper section of as-hobbed gear tooth*



Transverse magnification 3,000 *Lateral × 150*
FIG. 21*(b)—1:20 taper section of lightly-shaved gear tooth*



Transverse magnification 3,000 *Lateral × 150*
FIG. 21*(c)—1:20 taper section of heavily-shaved gear tooth*

difficult to determine, but there is confirmation of their presence to a maximum depth of 0·001 inch in the X-ray examination.

Taper sections were also made of the surface layers to examine the microstructure for grain flow along the surface as reported in other machining processes (Chisholm)[13], since Cameron[14] mentions cracks which started at the surface and followed grain flow. No serious alteration or deformation was found due to any of the three processes (see Fig. 21(a), (b) and (c)).

It has not yet been possible to examine a hardened and ground tooth for comparable stresses. The only evidence the author has on this process is that from the plate specimens mentioned earlier, in which the metal removed per pass by grinding was much greater than is normal in gear practice. The results are shown in Fig. 22, but they are not strictly comparable with the other curves, or necessarily indicative of the conditions of the *Theseus* pinions.

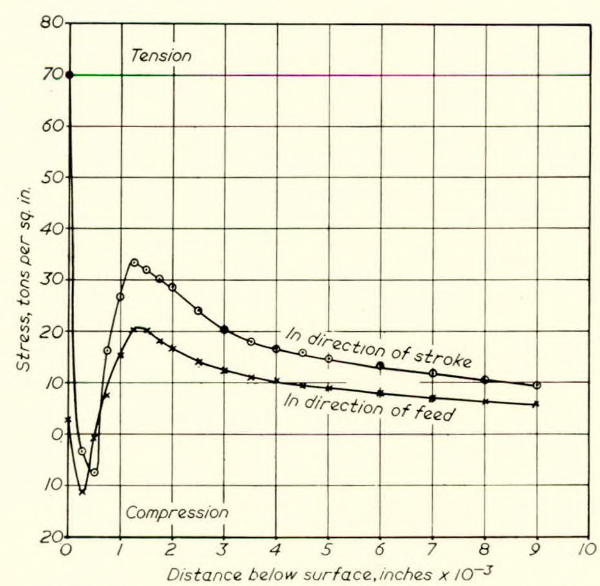


FIG. 22—*Residual stress distribution after grinding*

The actual stress system in the surface layers of a gear tooth is very complex due to several sets of stresses acting at the same time. The main stresses are listed below:—

(*a*) Residual stresses due to machining or heat treatment processes. For shaved gears these are all compressive, rising to moderate values in the surface layers. A ground hardened profile has probably tensile grinding stresses in the surface, which may change at some depth to residual compressive stresses caused by heat treatment.

(*b*) Contact stresses (both direct and shear) due to the deformation of the two surfaces. These have been calculated from Beeching and Nichols[15] for the centre of contact zone of the second reduction wheel and pinion combination (see Fig. 23). When combined with the residual stresses due to (*a*) and allowing for the different principal axes, the resulting stresses are shown in Fig. 24.

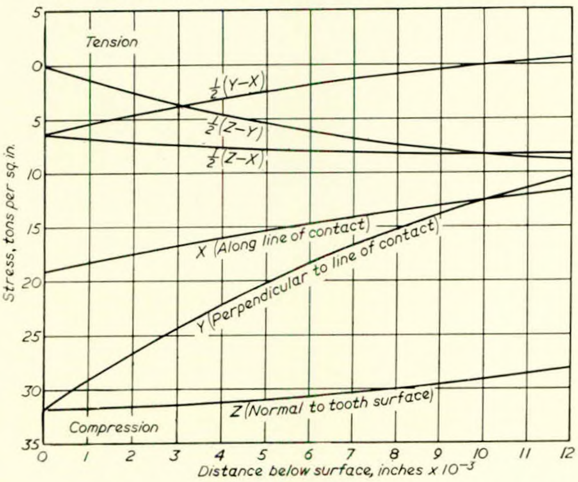


FIG. 23—*Stress components due to applied load*

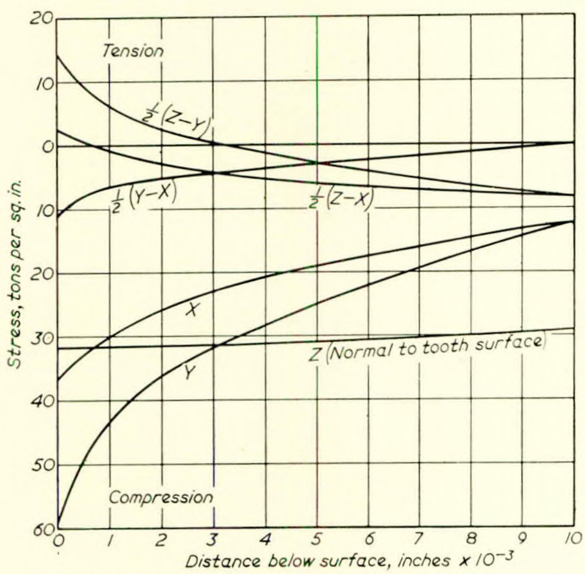


FIG. 24—*Components of combined residual and applied stress*

(*c*) Tensile bending stresses due to the tooth loading along the area of contact. These decrease from root to tip and from surface layers inwards to the neutral layer, while varying along the tooth between areas of contact. They are increased by stress concentration at the fillet radius, but are quite small compared with the residual stresses in (*a*) or the contact stresses in (*b*).

(*d*) Shear stresses in the surface layers due to either the oil film, if lubrication is effective, or metal-to-metal contact, if the lubrication has failed.

Thus it is not difficult to imagine the surface layers of the material at the area of contact being subject to fluctuating compressive stresses over the range 30 to 60 *T* per sq. in. in the direction from root to crest; 18 to 36 *T* per sq. in. along the tooth; and 0 to 32 *T* per sq. in. normal to the surfaces. It is somewhat more difficult to imagine a serious tensile component producing the more familiar stress reversal and hence fatigue failure. The behaviour of this type of steel under fluctuating compressive stresses is not known in detail: such evidence as exists suggests that the permissible stress range is, if anything, higher than the endurance limit under alternating stress, which is approximately 37 tons per sq. in. The residual stresses due to shaving only increase the (compressive) mean stress and are thus unlikely to impair the resistance of the material to fatigue under the loading stresses. It would appear improbable, therefore, that fatigue in the surface layers was the cause of failure. It is, of course, desirable that development of shaving cutters and the shaving process should proceed with the object of reducing the residual surface stresses to a minimum.

In the region immediately below the centre of the contact area the three principal stresses are of the same order and the shear stresses are thus small. This condition is unlikely to produce plastic deformation. Any overloading due to torsional conditions will increase the contact stresses but not the residual stresses. This will alter the shear stresses somewhat but the change should not be serious in this case.

In the surface surrounding the contact zone there are only two principal stresses, since the normal stress is zero, and plastic surface deformation could occur if the combined stresses exceeded the mechanical limits of the material. This could account for the loss of tip relief during the first two voyages when metal may have been rolled along to fill up

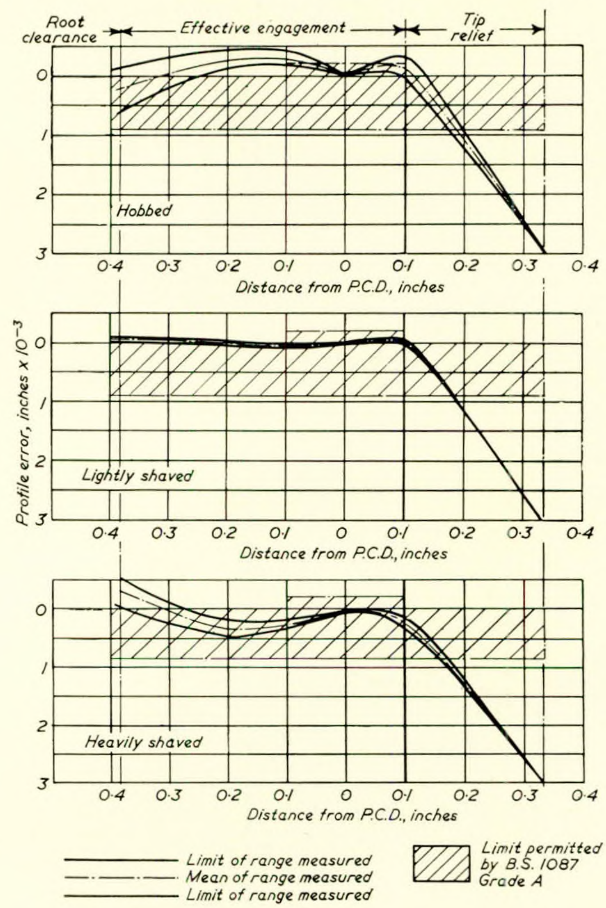


FIG. 25—*Profile records from test pinion based on Maag P.H. 60 measurements*

the tip relief space. This would produce an effect similar to that of uniform wear over the addendum, which contradicted the tooth thickness measurements.

Profile and Surface Finish

Since both the first reduction and second reduction pinions were too heavy for mounting on the P.H. 60 machine, no profile records were taken of the pinions or wheels during manufacture. When the three pairs of discs representing the second reduction pinion were made for stress analysis, opportunity was taken to measure the profile of the teeth produced by the three processes. The results are shown in Fig. 25, the mean profile and the range of profile variation being shown. The hobbed profile was as would be expected with small hob setting errors (see Merritt[16]). The shaved profile was good when shaved correctly but when heavily shaved tended to become hollow.

If the profiles are assumed to be representative of both pinion and wheel, then by redrawing the two addenda so that any point on a tooth profile is the same distance from the pitch point as its mating point on the other tooth, a pair

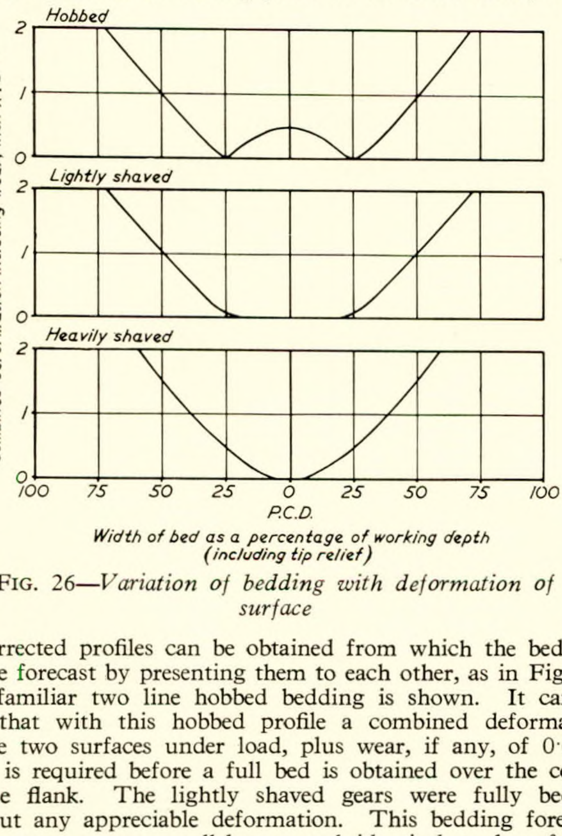


FIG. 26—*Variation of bedding with deformation of surface*

of corrected profiles can be obtained from which the bedding can be forecast by presenting them to each other, as in Fig. 26. The familiar two line hobbed bedding is shown. It can be seen that with this hobbed profile a combined deformation of the two surfaces under load, plus wear, if any, of 0·0005 inch, is required before a full bed is obtained over the centre of the flank. The lightly shaved gears were fully bedded without any appreciable deformation. This bedding forecast, of course, assumes parallel axes and identical angles of lead on mating helices. The initial bedding on the test bed (Figs. 7, 8 and 9) shows that the shaving of the actual gears was not as complete as that of the lightly shaved test pinion since some two line bedding is indicated on some helices.

A further advantage of good quality shaving is demonstrated by the narrowness of the range of profile measured, which indicates consistency of profile and hence reduced dynamic loads. A departure from correct profile of a helical gear, if constant across the helix, affects only the bedding and does not excite any torsional disturbances.

The surface finish produced by the three methods are given below:—

	C.L.A. value. in. × 10^{-6}
Hobbed	35
Lightly shaved	22
Heavily shaved	37

Examination of the System for Torsional Vibration

Since one cause of the overload on gear teeth is torsional vibration of the system, it was decided to measure the torsional and axial accelerations by mounting appropriate accelerometers on the coupling ends of the pinions of the i.p. branch. Dorey and Forsyth[17] referred to the need for accurate calibration up to 5,000 cycles per second for investigating the effects of gear errors, but their apparatus was only calibrated up to 200 cycles per second for torsional use. Experience with a similar apparatus and pick-up in the Research Department of the author's firm, led to the development of accelerometers for direct torsional and axial acceleration measurement. This work was not completed in time for test bed examination of the gears concerned but, by the courtesy of the shipowners, some results were obtained on two pinions at two propeller speeds for torsional accelerations only during a coastal voyage of the *Theseus*. It was not possible to change over to the axial accelerometers without stopping, and hence no readings for this type of tooth loading were obtained. A brief description of the apparatus used is given in Appendix III, the readings taken being shown in Figs. 27 and 28.

A torsional study of the system during the design stage enabled it to be tuned to arrange the first critical (due to propeller excitation) to be at 34 r.p.m. and the second, 78 r.p.m. of the propeller shaft. For this purpose the system was considered as a number of equivalent masses connected by mass-less shafts, the main masses being corrected to include the masses of the interconnecting shafts, and the shaft flexibilities to include the flexibility of the pinion, wheel, or turbine shaft concerned up to the mass centre. In the case of the gear elements, this was assumed centrally placed between the helices.

When the system was examined for excitation due to possible gear errors, the pinions were considered as a six-mass system, with massless shafts as in Dorey and Forsyth[17]. For the effect of first reduction gear errors, the high speed portion from turbine to high speed pinion inclusive was considered, and for the second reduction gear errors, the whole branch from turbine to low speed pinion inclusive was considered. In the latter case, to simplify calculations, the first reduction pinion was again considered as a single mass. By assuming the excitation to occur at any one helix of the branch system it was possible, by means of the Holzer method of tabulation, to determine at any frequency the torque on that helix when the coupling end of the pinion had unit acceleration. These values of torque have been shown in Figs. 27

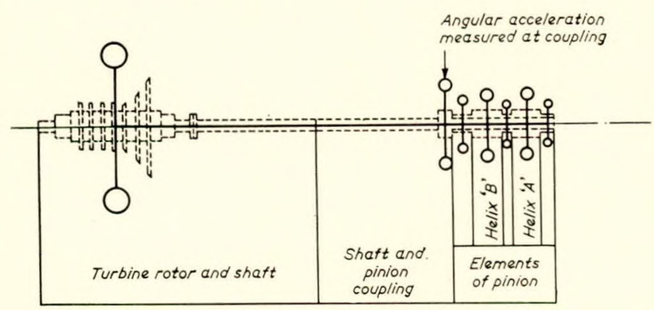


FIG. 27(*a*)—*System associated with i.p. first reduction pinion*

and 28 as a smooth curve on a base of frequency for both helices of the two cases considered.

By this method it is not possible to say which helix is causing the measured acceleration and, hence, both must be calculated to determine the highest possible loading at any frequency. The assumption that the pinion is a multimass system coupled by massless shafts, is not strictly correct, and is only acceptable at frequencies below the natural frequencies of vibration of the individual shafts in the pinion

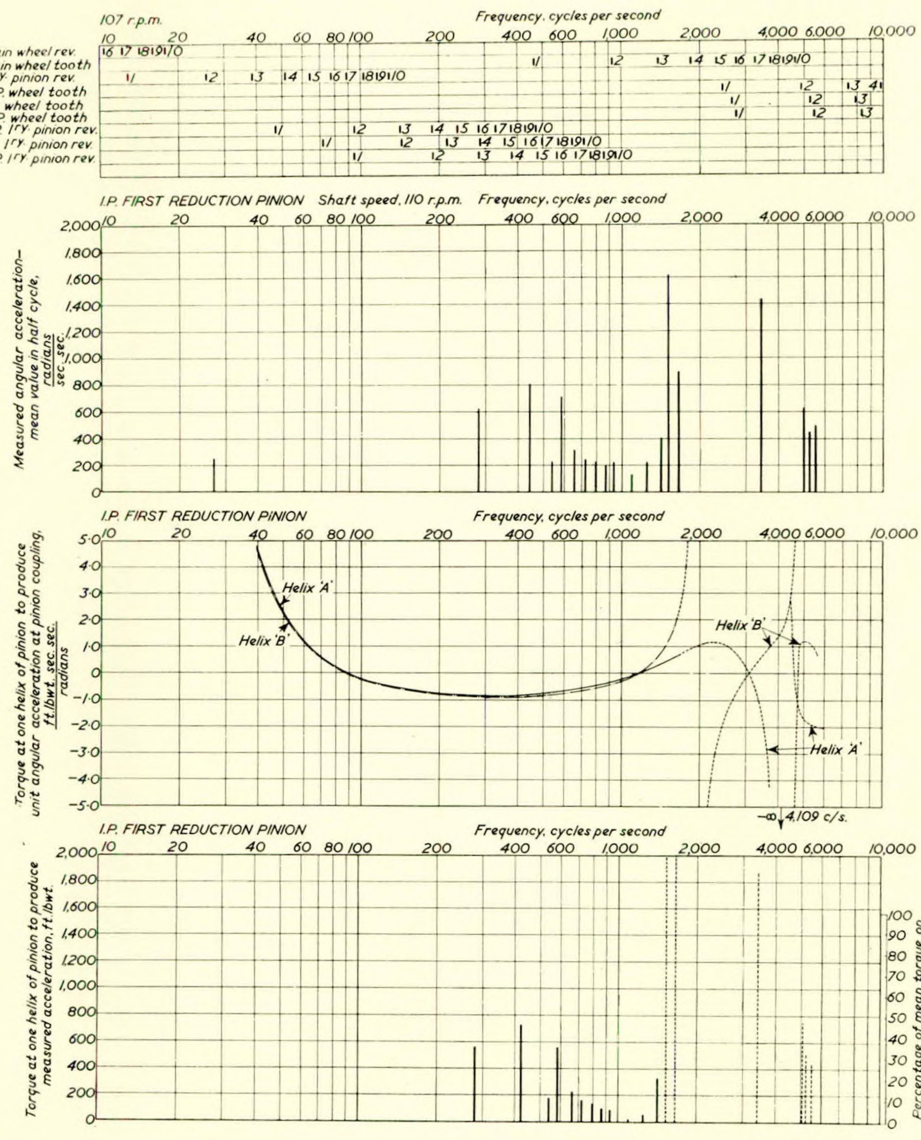


FIG. 27(*b*)—*Torsional analysis of system associated with i.p. first reduction pinion*

system, which are assumed massless. The difference in value of the first anti-resonant frequency of the pinion system when the pinion is assumed to be either a six-mass system or a uniform continuous shaft, is shown in Table VIII for the two cases considered and for excitation due to either helix.

TABLE VIII.

Pinion	Excitation	6-mass system	Continuous shaft
1st reduction	Mass 2	4,100	3,450
	Mass 4	1,952	1,730
2nd reduction	Mass 2	1,610	1,485
	Mass 4	884	835

Thus, for frequencies above, say, 1,700 cycles per second for the high speed pinion and 800 cycles per second for the low speed pinion, the values of torque to produce unit acceleration, calculated as mentioned above, are liable to large errors, and cannot be used to convert measured accelerations to torque, with any reasonable accuracy.

If the tooth loadings due to excitations at higher frequencies are required, then a closer approximation to the actual system must be made by dividing the system up into an even greater number of masses with shorter, stiffer shafts connecting them. This renders the calculation very much more formidable.

It can be seen that for the two cases considered over the range applicable, the maximum torsional loading due to

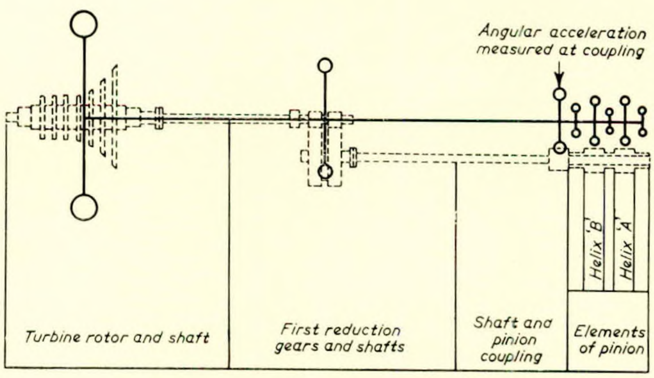


FIG. 28(*a*)—*System associated with l.p. second reduction pinion*

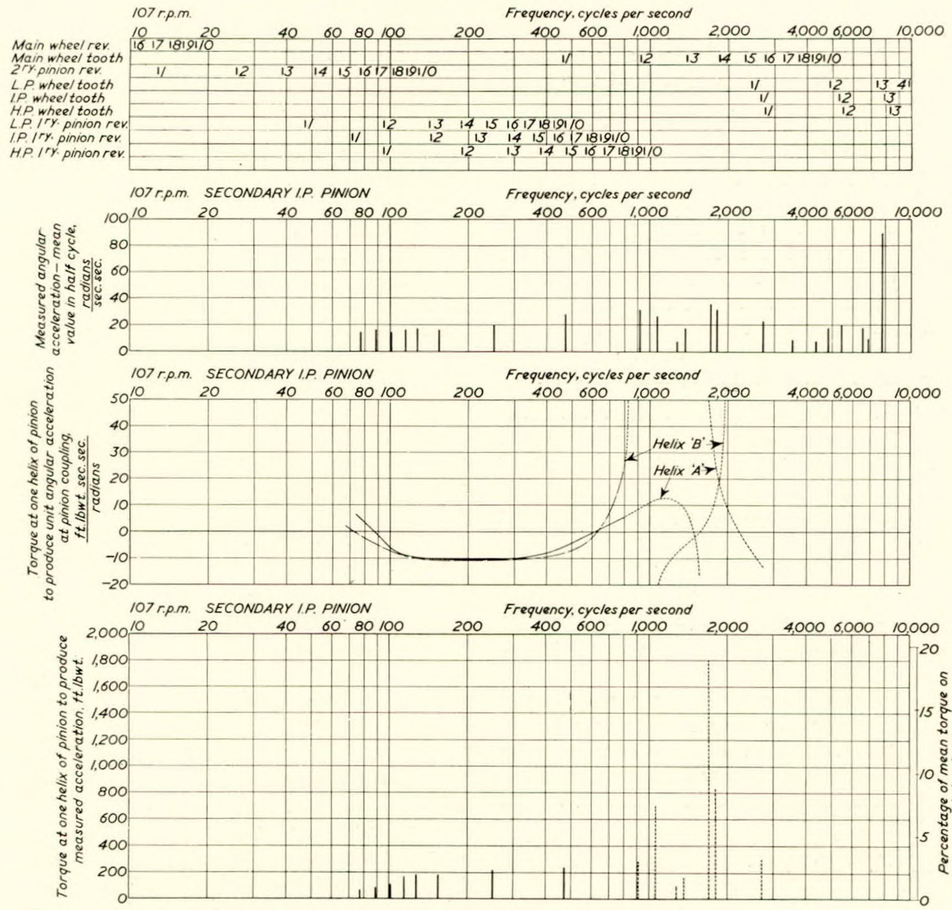


FIG. 28(*b*)—*Torsional analysis of system associated with i.p. second reduction pinion*

any single measured acceleration, expressed as a percentage of steady torque, is 2½ per cent for the second reduction gears and 47 per cent for the first reduction gears based on "averaged" values of acceleration. The analyser gave an averaged reading of acceleration at any frequency, but since acceleration was a recurrent wave form of approximately sinusoidal shape with continuously varying peak values, it was difficult to convert an "averaged" acceleration to a "zero to peak" acceleration. Assuming a sinusoidal shape, the conversion is a simple multiplying factor of 1·4, but, in fact, due to the departure from sinusoidal, and the fluctuation of the peak values, it is probably more likely 2 or even greater. It can thus be seen that for the first-reduction tooth contact, the maximum loading at any one frequency due to torsional vibrations possibly induced by gear errors is about 100 per cent of the steady value, while for the second reduction the figure is 5 per cent.

The negligible effect of the overload on the first reduction gear surface is probably due to the better oil film conditions. Based on the i.p. branch test the gear errors do not apparently contribute seriously to the tooth loadings of the second reduction elements, unless some of the very high frequency accelerations measured, which have not been converted to torque, are serious. Thus gear tooth profile errors or pitch-to-pitch errors do not appear to contribute seriously to the surface failure by causing torsional overloads.

GENERAL CONCLUSIONS AND REMARKS

(1) The use of K values of 120 are commercially satisfactory and show appreciable advantages in size and weight for double reduction mercantile marine gears. The one set of gears with no transient gear wear problem contained hardened and ground En39 second reduction pinions and these could probably have operated satisfactorily with even higher K values.

(2) With the "soft" second reduction elements a transient gear wear problem occurred, which disappeared with continued running. The cause of the wear is not known. The lubrication, which is quite satisfactory for the final worn profiles, has not been changed throughout, and if the initial conditions permitted lubrication failure and metal-to-metal contact, they must have been due to the initial material surface conditions. These must have improved with time of operation. There is some indication from rig tests using discs that for any particular set of velocity conditions the permissible load is dependent on a correct type of surface finish produced by careful "running in", for after this had been obtained, exceptionally high loads were carried quite satisfactorily.

(3) It is not clear how the residual compressive stresses produced by the shaving process could cause the failure of the material, either alone or combined with the operating stresses. For highly loaded "soft" gears where surface stresses are usually more important than bending stresses, further development of cutter design and machine operation, is required to reduce these residual stresses while maintaining profile and improving surface finish.

(4) The shaving process when carefully carried out produced a very consistent profile which approached closely the true involute form. If the shaving was overdone, however, the surface could be heavily marked and the profile seriously affected.

(5) The gears were very quiet, due to the consistency of profile of the shaved gears, the accuracy of tooth

pitching, and the torsional characteristics of the system. There were no serious torsional overloads due to gear errors over the calculated range. The effect of those torsional accelerations measured at very high frequencies was not determined.

(6) Further rig test work is required using disc machines before the behaviour of certain metal combinations can be explained or their performance under load forecast. This work requires the development of satisfactory methods of film thickness measurement, and the interpretation of the results obtained, for surface finishes likely to be met in practice and not under laboratory conditions. Short term tests, using higher loadings, should be regarded with caution, since any experience obtained may be of little use in predicting the performance to be expected under long life conditions.

(7) The doubt regarding the adequacy of the lubrication for the second reduction elements in their initial condition brings out the necessity to consider lubricating oils with E.P. additives on two counts. Firstly, if, as in the present case, the lubrication with a straight oil is ultimately adequate due to some changes at the surface of the material with time, then the additive oils may provide an increased margin of safety against either initial wear while the surface is changing, or subsequent wear due to an accidentally marred surface. Secondly, if the lubrication is quite adequate with straight oils and the materials are satisfactory in all respects, then the use of an additive oil may give either an added margin against contingencies or enable the loading to be further increased with the same margin, and additional gains in space and weight to be obtained.

This is not to be interpreted as a recommendation by the author in favour of lubricating oils with additives, on which he has had no experience. It is rather to suggest their consideration as a possible aid to progress. The designer must consider and accept this aid only after due trial and satisfactory performance, as he would other aids to progress in the form of better materials and manufacturing techniques.

APPENDIX 1

MATERIAL WEAR TESTS USING 6-IN. D.B.S. DISC MACHINE

Some tests were carried out on a 6-in. D.B.S. machine to endeavour to reproduce with discs the peculiar wear condition obtained on the second reduction pinions and wheel of the *Nestor*. The relative radii of curvature are given in Table IX for the discs and gears and the limitation of a 6-in. disc machine can be seen.

TABLE IX.

	Entry Gears	Entry Discs	Recess Gears	Recess Discs
Relative radius of curvature, in.	1·70	1·43	2·91	1·38
Disc load as percentage of gear load to produce same stress	84		47	
Disc stress as percentage of gear stress produced by same load	109		145	
K value of disc/K value of gear for same load	1·09		1·45	

Since the test conditions were those at entry, the K values for both gears and discs are almost the same when the loadings are the same. The tests are summarized in Table X. The very high K values used to accelerate the tests throw doubt on the usefulness of the results in this particular case, since the type and cause of failure may be different from that occurring on the gears under normal operating conditions. There are, however, one or two general observations which probably apply to both cases. If the surfaces are carefully run in using the oil film thickness as a guide, then it is possible even with the most difficult material in combination to obtain satisfactory operation up to a K of 700. This verifies the frequently ignored advantage to be gained by "running in" highly loaded gears carefully in the early stages. Given satisfactory surface finish there is apparently no unique scuffing load for any pair of materials over the range of loads envisaged in practice at the present time. For practical surface finishes, however, the tests indicate considerably more difficulty in operating En28 with En28 than in operating it with either a 0·3 per cent or 0·4 per cent C steel. When the discs scuffed the failure started in the centre, not at the outer edges where the load would be greater. This may be because the disc temperature (which in turn influences the residual oil film temperature) is the main controlling factor in determining oil performance in the contact zone.

It is suggested that odd asperities make initial contact and heat up the disc and hence the residual oil film on the surface; this allows more asperities to touch and so on. According to the shape of surface finish as well as its value, this may result in a well run-in surface with good lubrication when all the major asperities have gone, as in Fig. 18, or a progressively increasing metal-to-metal contact culminating in scuffing.

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APPENDIX 2

MEASUREMENT OF THE MACHINING STRESSES IN GEAR TEETH

In this method a plate type specimen is used, one face of which is the machined surface under examination, the other being polished optically flat. The change in curvature of the specimen as successive thin layers are removed from the machined surface is measured on the polished face by optical interference.

To apply this method to a gear tooth, a plate-type specimen must be cut from the tooth, one face of which is the tooth surface itself, the other being polished optically flat and parallel with the tangent plane to the tooth surface at the point where the stresses are to be measured. The surface stresses actually measured on this specimen will consist of the original machining stresses modified by any distortion the specimen may have undergone as the result of the cutting out and polishing procedure.

A second (steel) optical flat was cemented to the tooth surface before cutting out. The cemented-on flat was flexible compared with the specimen, and was hand-fitted to the tooth surface to keep the thickness of cement to a minimum. Final polishing of this flat was done after the cement had set and before cutting out the specimen. Subsequent distortion due to drying-out of the cement was found to be negligible, in four weeks the change being only one fringe (0·00001 inch). This flat was used to measure the distortion of the specimen when it was cut out and polished. This second flat was then removed, leaving the surface ready for the layer removal process. The corrections to be applied for distortion during cutting-out and polishing were found to be small in all cases (of the order of 1 ton per sq. in.).

Layers of metal were removed from the machined surface by etching, the thickness of the layer being about 0·0001 inch in the early stages, afterwards increasing as the limit of the stressed region was reached. After each etch the curvatures of the specimen in two mutually perpendicular directions (from root to crest and tangential to the pitch helix) were measured and plotted against total thickness of metal removed. From smoothed values taken from these curves the corresponding stresses were calculated, using Letner's[17] formula. All measurements were made at the centre of the tooth face, in which

TABLE X.—SUMMARY OF DISC TESTS.

Test no.	Details of large disc	Peripheral speed, f.p.m.	Details of small disc	Peripheral speed, f.p.m.	Loading	Total revs., small disc	Remarks
11	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=908	$3{\cdot}5\times10^3$	Entry conditions. Both discs scuffed after 2 mins.
12	7·69 in. dia. wheel En 28	1082	4·31 in. dia. pinion En 28	1451	K=581	7×10^5	Recess conditions. Discs scuffed within 3 mins. (4×10^3 revs.) continued despite noise and vibration.
13	7·69 in. dia. wheel 0·3% C.	1082	4·31 in. dia. pinion En 28	1451	K=581	$4{\cdot}2\times10^6$	Recess conditions. No scuffing. Pinion disc burnished—no appreciable wear. Wheel pitted after $3{\cdot}6\times10^6$ revs. and increased to end of run. Track otherwise original ground finish with odd burnished spots (presumed to be tips of asperities).
14	7·69 in. dia. wheel 0·4% C.	1082	4·31 in. dia. pinion En 28	1451	K=581	1×10^7	Recess. No scuffing. Pinion disc burnished—no appreciable wear. Wheel disc showed signs of incipient pitting which commenced at 9×10^6 revs. Original ground finish retained.
15	7·33 in. dia. pinion En 28	772	4·67 in. dia. 0·3% C.	1176	K=563	8×10^6	Entry. No scuffing. Pinion disc had original ground finish—no appreciable wear. Wheel dia. had slight wear and slight pitting which started at $5{\cdot}3\times10^5$ revs.
16	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=176	$8{\cdot}3\times10^6$	Entry. No scuffing or pitting on either disc. Pinion disc had original ground surface. Wheel disc burnished and some wear.
17	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel 0·4% C.	1176	K=563	$1{\cdot}3\times10^7$	Entry. No scuffing. Pinion pitted starting at 8×10^6 revs. Small no. of pits were usually 0·1 in. dia. × 100 at 0·020 in. dia. otherwise surface was as ground. Wheel disc burnished and worn by $1{\cdot}5\times10^4$ on radius.
18	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=176 to 563 in steps	1×10^6	Entry. This is an attempt to "run in" surfaces. Load was raised to maximum but scuffing occurred at $1{\cdot}7\times10^4$ revs. No pitting or appreciable wear. Ran without damage, K=491.
19	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=308	$3{\cdot}6\times10^4$	Entry. Mild scuffing at 6×10^3 revs. and thereafter ran satisfactorily.
20	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=425	$2{\cdot}1\times10^3$	Entry. Scuffed after $1{\cdot}9\times10^3$ revs.
21	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=248	$13{\cdot}3\times10^6$	Entry. No pitting, scuffing or perceptible wear. Pinion disc as ground. Wheel disc as burnished.
22	7·33 in. dia. pinion En 28	772	4·67 in. dia. wheel En 28	1176	K=248 by steps of about 70 to 718	$1{\cdot}2\times10^7$	Entry. No scuffing or appreciable wear. 1 pit appeared in pinion disc after 7×10^4 revs. but no further pits developed.

position the axes of measurement mentioned above were found to correspond closely with the principal axes of stress.

The values obtained, corrected as described above for distortion of the test piece during cutting-out and polishing, are plotted on Fig. 20 against distance from the surface. No claim can be made for their absolute accuracy because the method of calculation applies strictly only to specimens of constant thickness. From the comparative point of view, however, the results should be useful since the dimensions of all test pieces used were almost identical.

APPENDIX 3

S.S. THESEUS: NOISE AND VIBRATION MEASUREMENTS

Total noise measurements were taken, using a General Radio Sound Level Meter Type 759B on the flat weighting. This consists essentially of a Rochelle Salt microphone, flat amplifier and rectifier meter, the sensitivity in terms of sound pressure being constant, ± 3 db, from 50 c/s to 5,000 c/s. The sound pressure level of the main components was measured using a General Radio Sound Analyser Type 760A in conjunction with the above equipment. The analyser has been modified to cover the frequency range of 7·5 c/s to 7,500 c/s and the necessary corrections have been applied to compensate for changes of sensitivity.

The angular, or rotational vibration was measured by a Metropolitan-Vickers Type 20 Rotational Accelerometer (see Maguire[18]) feeding through a cathode follower into a Muirhead-Pametrada wave analyser. This rotational accelerometer is of the inertia piezo-electric crystal bar type. A non-magnetic inertia bar is rotated from the shaft through two symmetrically disposed pairs of quartz crystals, so arranged that they are sensitive to rotational oscillation, but insensitive to linear vibration. The couple necessary to accelerate the bar produces charges on the crystal faces proportional to the acceleration. The resulting potential difference is passed by capacitance type slip rings to the cathode follower, the output voltage being measured on the wave analyser. The lowest resonant frequency of the instrument is in the region of 5,000 c/s with a magnification of less than ten times. The instruments have been calibrated by an electro-optical method at low frequencies and by a reciprocity technique up to at least 10,000 c/s, the sensitivity being constant, ± 1 db, from 5 c/s to above 3,000 c/s.

Magnetic tape recordings of the rotational acceleration were made over a wide range of propeller speeds so that they could be replayed in the laboratory and used to assist in the detection of resonant conditions. To help in identifying the shaft responsible for specific components a Muirhead-Pametrada correlator was used in conjunction with the frequency analyser.

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