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Trends in the Development of Marine Reduction Gearing

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A number of papers have been presented during the course of the last few years dealing with the problems involved in the manufacture of large high-speed reduction gears and the ill consequences of inaccurate production. Some of these papers have served to emphasize the need for further development and others have reflected the nature of the progress made, which in some cases has been such as to show that the difficulties of accurate production might be regarded as historical in the manner of their solution.

It is in fact time that this position should be recognized more widely and that the forces of development should be directed to the various advantages that can be taken in materials and design with the availability of accurate gearcutting. The nature of some of these possible developments form the subject of this paper.

Ten years ago it was generally recognized that tooth surfaces would be rough to a degree which would only provide for contact between pinion and wheel teeth on a series of unevenly distributed points, as clearly evidenced at light loading, and anyone versed in gear production at that time will recall how necessary it was that, in order to give a respectable semblance of marking on the pinion teeth, gears in the course of meshing should be revolved for a considerable period in order to ensure that each pinion tooth should engage with a large number of gear wheel teeth and thus with a large variety of gear wheel tooth profiles, and the effect produced was greatly helped in many quarters by a liberal allowance of red lead marking. The loading for which such gears were designed was based on the assumption that contact would be badly distributed and that the inertia effects resulting from bad pitch distribution would be incalculable but considerable. Papers have been presented from time to time analysing the intensity of loading brought about by bad tooth profiles, bad helical angles, poor pitch distribution and surface irregularities and as a subject of scientific analysis the problems involved have indeed provided a lucrative field for theory and there is no doubt but that the tremendous improvements which have been achieved are consequent upon the interest thereby provoked. On the other hand the time has now arrived when it must be appreciated by all concerned that the scene has changed. Errors have now been reduced to a degree where it can be justifiably claimed that 80-90 per cent of the theoretical line of contact is in fact operative over the full face width, pitch errors are reduced to an insignificant value and the surface finish of the teeth has become a matter of considerable satisfaction. This is not to be complacent about the possibilities of further progress, nor is it to suggest that these attributes can be achieved without proper care and inspection, even given the necessary modern equipment of production to which reference is made in Appendices 1 and 2, but the point is that this high quality is now within the definite control of the producer and it is the business of the designer to proportion his gears in the knowledge that such results can be achieved and of the superintendent engineers to be satisfied with no lesser standard.

Perhaps it would be appropriate to couple this last injunction with a warning that high quality gears cost more per ton weight than their less accurate predecessors, but it is to be noted that development has not overtaken economic saturation in that the reduction in size rendered possible by higher accuracy more than offsets the increased cost of production per unit weight, disregarding, of course, increases in costs of labour and material that have applied simultaneously over the period of development. On the other hand it is to be recognized that if too conservative a view is allowed to prevail in limiting tooth loading rates to figures more generally applicable to out of date gears, an uneconomical article is produced by reason of the fact that it becomes unnecessarily large for the duty expected of it.

The principal lines on which developments are now progressing and their influence on design may be treated conveniently under the following headings:—

(1) Increased loading rate to take proper advantage of accurate production.
(2) Adoption of higher duty materials as rendered possible by advent of accurately finished gears and permitting in its turn a further increase in loading rate.
(3) Improvements in tooth form permitting increase in loading rate.
(4) Effect on gear proportions introduced by the above.

The possibilities of development are also discussed in respect of the following:—

(5) Adjustment of helical angle to absorb pinion distortion.
(6) Single helical gears.
(7) Flexible couplings.
(8) Reversible gears.

INCREASED LOADING RATE TO TAKE PROPER ADVANTAGE OF ACCURATE PRODUCTION

Development in this direction is a matter of reducing what has become an exaggerated margin of safety to more reasonable limits. The difficulties of assessing such limits under sea-going conditions are numerous and well known, and their determination is bound to involve the experience of years and can only be partially assisted by experimental work on land. A typical analysis of the position is given by Appendix 3.

ADOPTION OF HIGHER DUTY MATERIALS

Development in this direction introduces both practical and economic problems.

The materials commonly employed today were generally adopted for their purpose about thirty years ago and reflect in some respects the serious difficulties which arose at that time on account of the great inaccuracy of the engaging teeth. Only materials of considerable ductility and toughness would stand up to the punishment imposed upon them without risk of failure from fatigue, although the best possible compromise was made to minimize tooth surface distress by the employment of a normalized and tempered nickel steel for the pinions with a moderately low maximum stress.

With the advent of high accuracy cutting, gears of material having a higher maximum stress and with greater surface hardness are envisaged, involving a permissible sacrifice in ductility but allowing higher surface and root stresses to be borne.

It would be represented as a failure to face the realities of the position if a certain hazard represented by the adoption of higher duty materials and heavier loadings was overlooked, in view of the inadequate experience which has been obtained in that direction with large gears up to the present time. Clearly, the adoption of higher duty materials, in conjunction with too conservative a measure of their capacity, would lead to the production of uneconomical gear units and, perhaps, no very spectacular advance is likely to be achieved in the materials employed for merchant gears within the next few years. During this time the demand will probably be for selectively shaved gears with materials of conservative maximum stress.

In the case of naval design, the demands of minimum weight and space tend to outweigh the considerations referred to and not only materials of high maximum stress but also surface hardened and profile ground gears demand the designer's attention.

When a gear is surface hardened after cutting the teeth the distortion which occurs is inevitably such as to necessitate re-profiling the teeth and this of course can only be done by grinding. While the surface hardening of pinions is carried out in a hardening furnace under properly controlled conditions and can be a thoroughly satisfactory process, the hardening of wheel teeth, except in the case of small primary wheels, can only be carried out by a process involving a travelling flame introducing conditions which are difficult to control in ensuring uniformity.

At the same time it is to be recognized that the duties which pinions perform are generally much more arduous than those of the wheels with which they mesh and that there is little to justify the case hardening of wheel teeth, since suitable heat treated steels are available wherein the hazard of flame hardening can be avoided and which can incidentally be finish processed by hobbing and shaving.

It is to be observed that gears often referred to as hardened and ground comprise pinions that are hardened and necessarily ground and wheels that are not hardened but that have been ground because grinding facilitities were the only means available for accurate finishing.

Comparisons and tabular statements of the effects introduced by the employment of typical high duty steels are given in Appendix 4.

IMPROVEMENTS IN TOOTH FORM

This is a factor which has made a useful contribution to the increase in permissible tooth loading. It is fairly well established that the tip relieved involute type is the most suitable, but many of the particular involute forms that have been in common use have failed to take the best advantage of the possibilities available. The matter is one of complexity but the essential features can be summarized concisely as below.

Flank Angle

Within reasonable limits the flank angle can vary without in itself affecting the suitability of the tooth. A large flank angle increases the radius of curvature of the pinion tooth and enables a greater load to be carried at the line of contact, but the effect is closely compensated for by the fact that the line of contact is itself shortened. The real importance of flank angle is to be associated with tooth depth, a small flank angle being a necessary adjunct of a deep tooth.

Tooth Depth

The greater the depth of the tooth the longer the line of contact and the greater the load that the tooth will carry from the point of view of surface stress. Considerations of gear cutting limit the depth of the teeth in relation to the pitch to about 45 per cent in excess of the British Standard tooth and for which the most suitable flank angle is about 14½ deg. Another consideration which arises is that of the ratio of sliding to rolling, which can become excessive in some circumstances when an excessively deep tooth is employed, but every satisfaction has so far been shown by limitation in depth to about 20 per cent greater than the British Standard form, and for which the most suitable flank angle is about 16 deg. The optimum tooth forms are, in the author's opinion, always to be found between these limits and in marine reduction gearing there is no justification for so shallow a tooth as the British Standard indicates.

Pitch

The pitch of the teeth should always be the least that is permissible in relation to the root stress for the loading employed. With the materials presently employed all normal requirements for marine gears are met between the limits of 0·4 and 0·8 inch, but as suggested by the last line of Table 4, medium high tensile materials may lead to the adoption of lower pitches for small gears, while case hardened gears may necessitate the employment of greater pitches for large gears to provide a reasonable balance between surface and bending stresses, while at the same time ensuring that bending fatigue is not the criterion of failure.

Addendum distribution

The aim in assessing the distribution of addendum between pinion and wheel is to equalize the ratios of sliding to rolling at the beginning and the termination of engagement. Balance is usually sufficiently achieved by the adoption of a pinion addendum approximating to 60 per cent of the active depth of the teeth.

Helical Angle

When the cutting of gears was indifferent in quality a large helical angle was justified to reduce the shuttling action on the pinions and to increase the component of force providing acceleration for such movement. With the development of high accuracy gears, these considerations have become irrelevant and the principal objects of adopting a helical tooth become those of establishing axial location and providing against shock loading at entry to the contact zone. These requirements are adequately met by a helical angle of only a few degrees but, in view of the small increment of axial loading introduced by angles of up to 12 or 15 deg., there is little point in dropping below these limits. For this latter angle the increment of loading due to the axial component is 3½ per cent but with a helical angle of 30 deg. this rises to 13½ per cent and, accordingly, the faces of gears so designed require 10 per cent additional width. The adoption of helical angles like 40 and 45 deg. is completely unjustified today.

When considered in comparison with pinion diameters, the relatively narrow faced gears now commonly employed do not warrant the thinning of the ends of the pinion teeth and the requirement for a reduction of load at the ends of the teeth is adequately met by the chamfer which it is customary to provide.

EFFECT ON GEAR PROPORTIONS

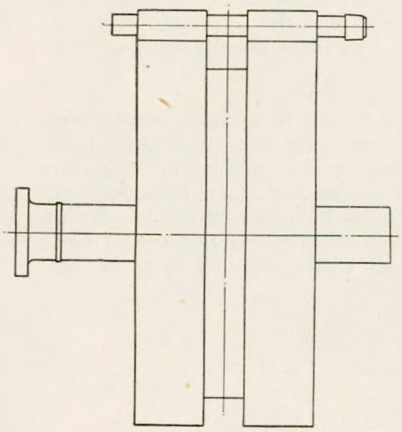
It is well known that double reduction gears suffered in this country a long period of banishment because of the tooth failures which became so commonplace as a result of the gross inaccuracies introduced in their formation. With the development of accurate gear cutting the double reduction gear now takes its rightful place in the marine installation, enabling smaller turbines to be adopted with lesser risks of distortion and the use of higher pressures and temperatures without a multiplicity of turbine units.

It is instructive to examine the relative sizes of a single train of gears such as illustrated in Fig. 1, where the quality of gear cutting and the loading that were standard say 20 years ago are compared with modern practice, employing similar materials. The increase in loading naturally permits a smaller gear unit, but generally the effect is not so much to reduce main wheel diameter as to introduce a reduction in the ratio of the length to the diameter of the pinions and this tends to reduce their torsional and bending deflexions and minimizes the need for centre bearings. The reduction of distortion leads to better distribution of load along the length of the teeth and this permits a further rise in average tooth loading; the point serves however to stress the great importance to be attached to accurate bearing alignment on relatively highly loaded gears.

ADJUSTMENT OF HELICAL ANGLE

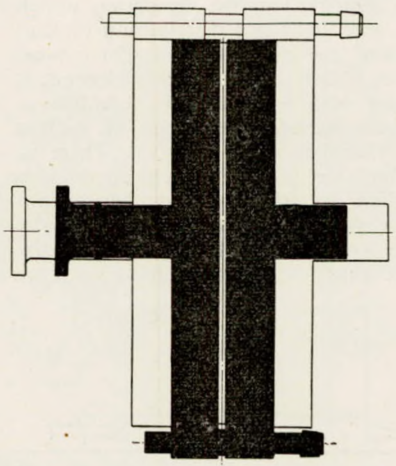
In cases where the length of the pinion helices exceeds about $2{\cdot}5\times$ the diameter of the pinion, consideration should be given, if reduction in the size of the gears is of paramount importance, to the introduction of a varying helical angle on the pinion teeth to compensate for distortion under load. Until recently this was a theoretical rather than a practical consideration, because of the difficulty experienced in obtaining an even

GEAR AS DESIGNED AND PRODUCED IN 1928



Item		Pinion	Wheel rim
Material (see appendix 4)		II	I
p.c.d., inch		13.0	162
Face width, inch		60	
Face width/ pinion p.c.d.		4.61	
Loading coefficients	P	575	
	$P/De^{2/3}$	110	

GEAR FOR TRANSMITTING SIMILAR TORQUE AS DESIGNED TO TAKE ADVANTAGE OF MODERN HIGH QUALITY FINISH



Item		Pinion	Wheel rim
Material (see appendix 4		II	I
p.c.d., inch		13·0	160
Face width, inch		40	
Face width/ pinion p.c.d.		3·08	
Loading coefficients	P	865	
	$P/De^{2/3}$	165	

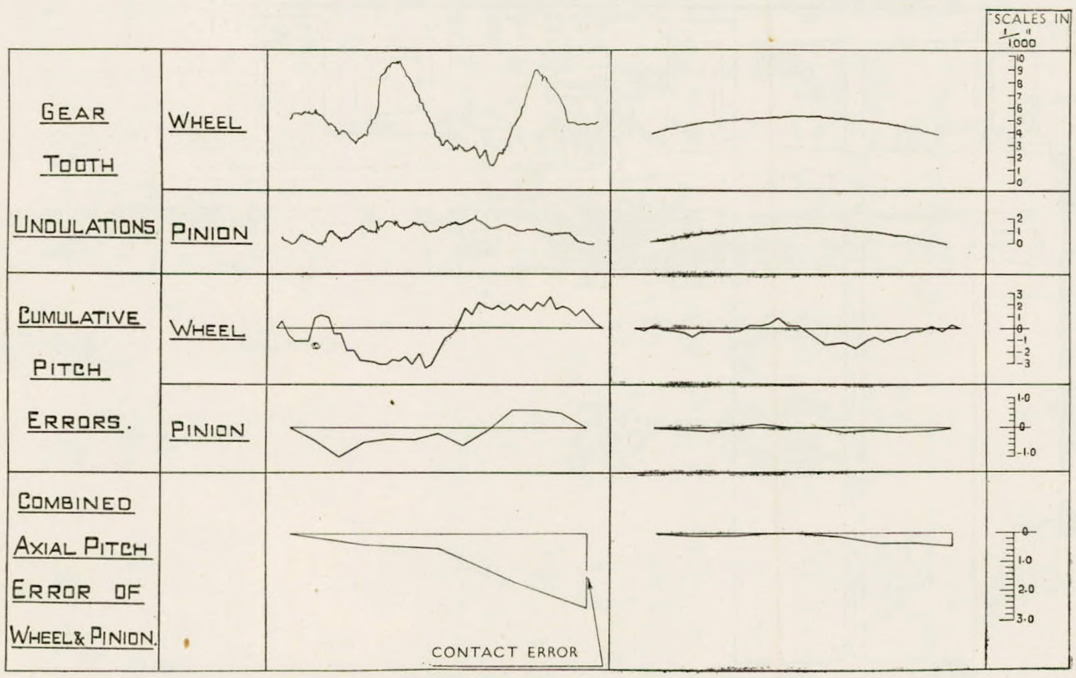


FIG. 1—*Comparative sizes and error characteristics of a single gear train of twenty years ago and today*

marking across the face width of the gears, even under static loading conditions. To have attempted further adjustment would have been to gild a somewhat wind-blown lily and the most that could be done under the conditions that prevailed was to ensure that no heavy marking occurred at the ends of the pinion teeth.

With the availability of modern equipment for gear finishing, it is however reasonably simple, after a uniform bearing has been obtained along the teeth, to modify the helical angle of the pinion teeth by a selective shaving process so as to obtain a proper distribution under full power load. In this connexion two points are to be emphasized—firstly, there is no advantage in attempting such adjustment on the gear before a correct static meshing has been obtained, because the errors in helical angle which cannot be dissociated from even the highest class of hobbing can be as great as the further degree of adjustment required to absorb distortion. It is therefore of no use to attempt any adjustment in the hobbing stage, even if facilities such as sine bar correctors are available on the pinion hobbing machine. Secondly, there is nothing problematical regarding the helix adjustment required and, once the manufacturing technique is mastered, to contemplate research in this field with test gears would be wholly out of place. Perhaps this point requires amplification. In the case of a pinion unadjusted for load distribution, the distortion which occurs under load and the consequent mal-distribution of load is a matter of great complexity, the calculation of which takes the form of an infinite series and is fundamentally dependent upon the flexibility of the gear teeth under dynamic conditions. The maximum loading which occurs in the case of such a gear running under load is therefore indeterminate. There is, however, no dubiety regarding the loading which exists on a gear having the pinion helices modified to provide uniform loading, the reason being that if an equal load is initially assumed to occur across the gear face, the form of pinion distortion can be calculated accurately and if the pinion teeth are finished to compensate for this distorted form an even distribution of load will be achieved effectively and the subject has no relation whatever to the unknown factors affecting load distribution on a gear not so adjusted. The subject does not therefore, in the author's opinion, lend itself to intelligent research.

SINGLE HELICAL GEARS

Something has been heard in recent years of an advantage to be found in the adoption of single helical gear units with thrust blocks to carry the out-of-balance load. This is indeed a case of the cart being put before the horse. The reason for the claim cannot be dissociated entirely from the production of ground gears, wherein the diameter of the grinding wheels is such as to require a gap of about 8 inch between adjacent helices, giving an obviously undesirable design when space and weight happen to be considerations of major importance. An aspect of this position which is not generally realized, lies in the fact that if only the pinion unit is to be profile ground on the tooth flanks, very much smaller grinding wheels may be employed because of the lesser wear which occurs in the grinding of a small gear. Consequently the gap between helices with gears of this type can be reduced to about 4 inch, enabling the advantages of a double helical gear to be retained in conjunction with the benefits of ground pinions where high loading renders the adoption of such a gear desirable. Reference to an earlier paragraph will show that the limitation of grinding to pinions is also supported by other considerations.

FLEXIBLE COUPLINGS

It is appropriate to put on record in a paper dealing with the development of marine gears that the large claw tooth sliding couplings, so widely employed until three or four years ago, are now to be regarded as out of date; their place has been taken by a small tooth coupling which can be produced with considerable accuracy and which requires a minimum of hand fitting. In operation the small tooth coupling is to be pre-

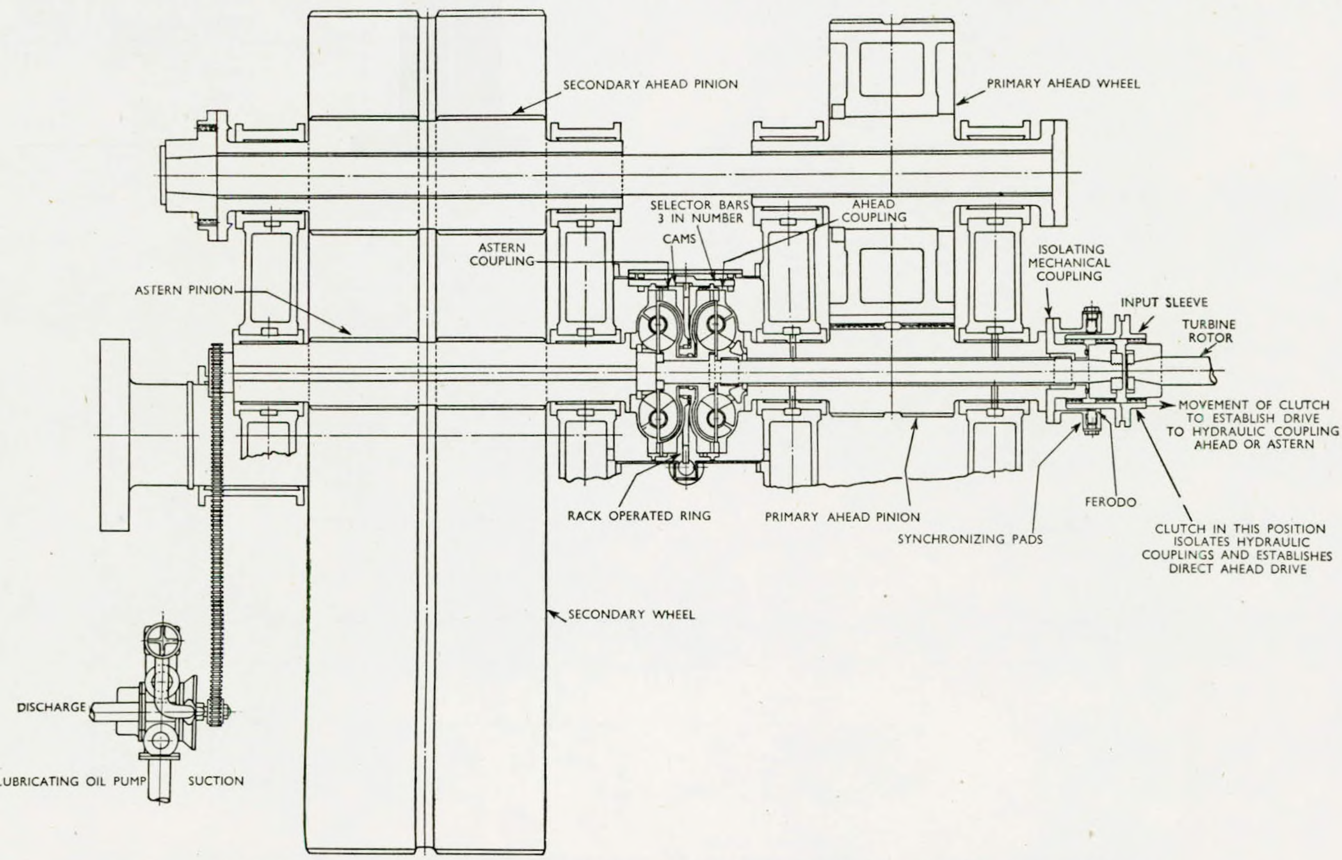


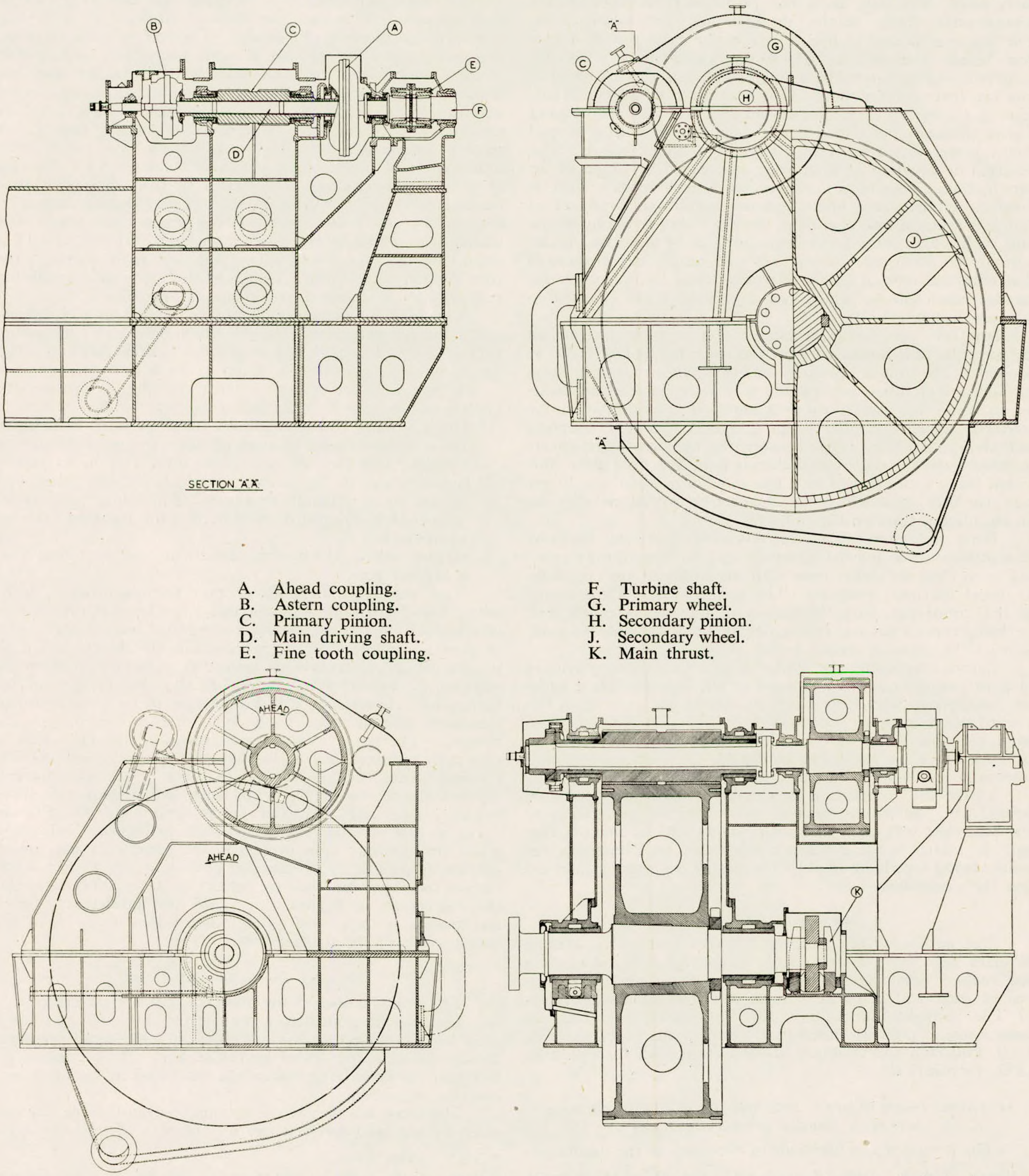
FIG. 2—*Gear with double reduction ahead drive and single reduction astern drive*

ferred, having particular regard to a number of vibrational complaints which have been attributed to couplings of the large tooth type. The members of the small tooth couplings are conveniently produced on a Fellowes type machine.

REVERSIBLE GEARS

A broad field of development has been opened up in the quest for means of obtaining reversal of propeller thrust without interruption to the normal flow of fluid in turbines. This has been necessitated primarily by gas turbine designs wherein it is not feasible to introduce an astern turbine and the matter is also one of significance in steam turbine propulsion when very high steam temperatures are employed.

The problem can, of course, be met by the adoption of propellers of the variable-pitch type but while this may have its attractions for relatively small craft, the system is unlikely to



A. Ahead coupling.
B. Astern coupling.
C. Primary pinion.
D. Main driving shaft.
E. Fine tooth coupling.
F. Turbine shaft.
G. Primary wheel.
H. Secondary pinion.
J. Secondary wheel.
K. Main thrust.

FIG. 3—*Arrangement of reversible gear train with hydraulic coupling and torque converter (Pametrada design)*

prove widely popular with owners operating larger tonnage.

The problem can also be met by electrical transmission but at the cost of at least 8 per cent loss in efficiency as compared with gearing and the introduction of complication that is not inconsiderable either in respect of initial outlay or upkeep.

Reversibility in gearing mechanism is being successfully achieved in small units by clutches of the airflex type or of the S.L.M. oil-operated type, working in conjunction with lay shaft gears. For large units two principal types are receiving consideration, both employing hydraulic reversal clutches. The first is indicated in Fig. 2, which illustrates a type of gear now being developed in which the ahead drive is double reduction and the astern drive single reduction, thereby eliminating lay shaft construction. Under astern running conditions there is a lowering of turbine revolutions and some consequent loss in efficiency but which is not incompatible with a normal astern power requirement of 70 per cent ahead power. The selection of direction of rotation is determined by which of the two hydraulic couplings associated with each turbine shaft is flooded. Under steady ahead steaming conditions a mechanical clutch is brought into operation, thereby isolating the hydraulic unit with which some 2 per cent slip is to be associated under normal conditions of driving. It will readily be appreciated that with this type of gear a relatively heavy loading is carried on the astern pinion which has its helical angle adjusted to take up distortion.

Another type being developed and incorporated in an experimental unit is referred to in a recent paper by Dr. T. W. F. Brown*. The unit is indicated on Fig. 3, ahead running being through a hydraulic coupling which is not capable of mechanical isolation and astern running through a hydraulic torque converter which is similar to an elongated hydraulic coupling embodying stationary vanes between the two rotating elements to reverse the circumferential direction of the fluid flow, this action being accompanied by a loss in efficiency of some 30 per cent, the heat equivalent of which must be carried away by the oil circulating through the coupling.

Both the above types of reversible gear are relatively straightforward in normal operation and any uncertainty existing as to their suitability rests with the ability of the couplings to dispel the heat generated when manœuvring. Experiments in this connexion, with the former of the two types of gear, are being carried out and it is anticipated that valuable information will be obtained on the matter at an early date.

Before concluding the author would take the opportunity of anticipating criticism in respect of the fact that for a paper on development very little has been said of research. Is it not a word that is perhaps overstressed today? Is it not sometimes employed when what is really implied is the investigation of troubles which are known to be capable of rectification with a reasonable application of plain commonsense?

In the field of gearing development, research is certainly needed into the employment of new materials in relation to the load they will carry and their adaptability in manufacture and the matter is being pursued vigorously by Pametrada, the results being eagerly awaited by the marine engineering industry and the steelmakers.

ACKNOWLEDGMENTS

The author wishes to make acknowledgments to Messrs. William Beardmore and Co., Ltd., whose metallurgical research has enabled developments referred to in the paper to be proceeded with, and in conclusion he would thank the Directors of The Fairfield Shipbuilding and Engineering Co., Ltd. for permission to publish information given in the paper and Mr. R. P. Fullerton and members of his Design Staff for assistance in its preparation.

APPENDIX 1—EQUIPMENT FOR THE PRODUCTION OF HIGH ACCURACY MARINE REDUCTION GEARS

The principal combinations of processes in the production of large high accuracy gears are:—

(1) Hobbing and shaving.
(2) Hobbing and lapping.
(3) Hobbing and grinding.
(4) Planing and grinding.

Except where resort is made to surface hardening the most exacting requirements can be met by the first two processes listed and to which the following notes refer.

The basic requirements of hobbing machines in respect of their accuracy of motion are clearly indicated by a British Standard Specification just issued*. The specification, however, gives no guide as to whether a creep mechanism is preferable in the drive between the main worm and the master gear; the disadvantage of such a mechanism is the obvious increase in the number of working parts calling for extreme accuracy, but its advantage, particularly for large gears, lies in the fact that it renders unnecessary the supreme accuracy in the worm drive unit that it is necessary to obtain and maintain in the case of a solid table machine where error in this component produces a periodic pitch error in the cut gear, which cannot be removed by any known post hobbing process and which will introduce a whine in the finished product. In the case of a creep table machine this objection does not apply although the very greatest importance attaches to the creep ratio employed and attention is drawn to this point in Appendix 2.

The hobbing machines should be housed in temperature controlled compartments, whereby the variation in temperature is limited to ±1 deg. F. and a generous temperature controlled supply of cutting oil should be fed to the hobs.

A post hobbing process is most desirable to procure the finest quality of finish in the following respects:—

(1) Tooth profile, which is always liable to some inaccuracy on a hobbed gear because of the extreme difficulty of ensuring that the hob runs at all times true to its axis.
(2) Helical angle, the uniformity of which, between wheel and pinion, it is difficult to ensure in hobbing even under temperature controlled conditions with the best class of equipment.
(3) Surface finish, which is fundamentally below standard on a hobbed gear.

Two alternative methods of post hobbing present themselves, namely, lapping and shaving. The lapping process can be effective if it is most carefully controlled and an abundance of time is available for the process, but the danger is always present of lapping compound being left embedded in the tooth surfaces, or indeed of its finding its way into crevices of the gear units. As an economic proposition in the production of merchant gearing there is in fact little to recommend the process. The other alternative is that of shaving, which is now perfected to such a degree that it is on an economic footing for merchant as well as naval gears. The process is rapid and consists first of shaving the wheel units and subsequently shaving the pinion units selectively, i.e. by varying the infeed of the cutter so that variations of helical angle as between pinion and wheel are rectified. The process is also completely effective in correcting profile errors and in providing a polished tooth surface free from undulations, subject to the qualifications that shaving cannot be regarded as a cure for indifferent hobbing and that it is impossible to eliminate waves longer than the active width of the shaving cutter.

APPENDIX 2—CREEP FRACTIONS

The principle underlying the choice of a good creep fraction has been developed from the late Dr. Tomlinson's pioneer work on the subject and is explained in some detail on pages 256-260 of an earlier paper by the author†. It is unnecessary to repeat the underlying theory but the following essential conclusions are relevant.

The creep fraction is the decimal portion of the fraction given by the ratio wm/c where w = number of teeth in worm

* Brown, T. W. F. 1948. Trans.N.E.C., Vol. 65, p. 117, "British Marine Gas Turbines"

* B.S.S. 1498:1948

† Davis, A. W. 1945. Trans.I.E.S., Vol. 88, p. 179, "Current Practice in Marine Gearcutting"

Table I—Loading factors on gears having 45 tons per sq. in. U.T.S. pinions 35 tons per sq. in. U.T.S. rims and 30 deg. helical angle

Type of gear	Year of production	Method of finish	Loading factor $P/D_e^{2/3}$ Active tooth depth relative to British standard value $\frac{2 \times \text{pitch}}{\pi}$ $\delta=1\cdot0$	$\delta=1\cdot2$	$\delta=1\cdot44$
Cross channel single reduction	1926	Hobbed	122		
	1936	Hobbed	128		
	1949	Selectively shaved		150	
Ocean going single reduction	1927	Hobbed	100		
	1931	Hobbed	120		
Ocean going double reduction primary	1922	Hobbed	115		
	1943	Hobbed	140*		
	1948	Selectively shaved		135	
Ocean going double reduction secondary	1922	Hobbed	85		
	1943	Hobbed			170*†
	1948	Selectively shaved			145
Destroyers single reduction					
British	1932	Hobbed	180		
British	1945	Hobbed	240		
German	1945	Hobbed	240		

* Designed abroad. † 18 deg. helical angle.

wheel; m = number of teeth in master wheel; c = number of teeth in creep ring. In effect the creep fraction represents the fractional difference in rotational position of the main driving worm after one complete revolution of the master wheel, the fraction representing the proportion of one revolution of the worm.

In the case of a solid drive machine the value of this fraction is zero (which may also be expressed as 0/1) and a cyclic error in the worm motion is reproduced in axial bands on the cut gear.

If a creep mechanism is introduced the value of the creep fraction should be chosen, having regard to the following features:—

(a) A creep fraction of $\frac{1}{2}$ introduces the same characteristics as a solid drive but in lesser degree, because of the phasing effect of alternate feeds, and this applies in similar fashion to any other value which can be expressed as a simple fraction, the effect becoming of decreasing importance as the value of the denominator increases.

(b) A long wave undulation occurs when the creep fraction approximates closely to a simple fraction and in this connexion it is important to note that the lower the denominator of the simple fraction the wider berth it should be given.

APPENDIX 3—LOADING FACTORS

The derivation of a true criterion of loading is a matter of complexity but it is nevertheless essential for purposes of design and general comparison to have a simple and reliable criterion with a knowledge of its limitations.

American practice is to employ the factor P/D_e where D_e is the pinion pitch circle diameter (p.c.d.) corrected for the finite diameter of the wheel with which it is meshing, by the expression $D_e = \frac{(\text{p.c.d.})\, R}{R+1}$ where R = gear ratio.

For many years it was the practice of Messrs. Parsons Marine Steam Turbine Co. to use this expression for pinion diameters up to 10 inch and to use the expression $P/D_e^{0\cdot5}$ for pinions above 10-inch diameter. It is known that this latter value errs on the low side in respect of the index of D_e and also that a value for this index as high as unity is not justified. When this latter value is employed the factor obtained can only be considered in relation to the size of gear referred to, that is to say, that the smaller the gear the higher the factor that may be permitted. When this consideration is overlooked, as it has been on notable occasions, the results obtained are dangerously misleading.

One authority claims than an index of 0·8 is to be desired, another that a value of $\frac{2}{3}$ is to be preferred. A marine engine designer usually works with a slide rule having a cubic scale and the value of employing an index of $\frac{2}{3}$ is therefore not inconsiderable as a matter of convenience and its employment is, in the author's opinion, to be encouraged, thereby giving a loading criterion of $P/D_e^{\frac{2}{3}}$.

The principal limitations of this factor are as follows:—

(1) There must be a limiting value of P for every tooth form in relation to the type of material employed, this being from the point of view of bending stress on the teeth, it being recognized that the factor is in itself really a function of surface stress.

(2) For purposes of comparison the factor must be regarded as a function that varies proportionally to the ratio of tooth depth to tooth pitch.

(3) If the pinion helices have not been adjusted to correct for distortion under load and the ratio of gearface width to pinion diameter exceeds about 2·5, no centre pinion bearing being fitted, then the allowable value of the loading criterion must be limited to take proper account of con-

Table 2—Typical steels suitable for development as gear elements

Reference	C	Ni	Cr	Mo	Va	Heat treatment	Process required for finishing teeth	Present usage	U.T.S., tons per sq. in.	Yield, tons per sq. in.	Elongation, per cent in 2 inch	Izod	Brinell
	Approximate analysis												
I	0·3	—	—	—	—	Normalized	Selective shaving	Rim-standard	35	18	26	20	150
II	0·3	3·5	0·25	—	—	Normalized and tempered	Selective shaving	Pinion-standard merchant	45	33	23	40	185
III	0·3	3·5	0·25	—	—	Oil hardened and tempered	Selective shaving	Pinion-standard naval	45	35	25	45	200
IV	0·3	3·0	0·8	0·4	0·1	Oil hardened and tempered	Selective shaving	Proposal	60	53	20	40	300
V	0·35	4·0	1·75	0·45	—	Air hardened and tempered	Selective shaving	Proposal	80	72	14	15	380
VI	0·15	3·0	0·6	—	—	Case hardened and refined	Profile grinding	Proposal	55* (root)	40* (root)	20 (root)	30 (root)	500

* There is some uncertainty as to whether these relatively high values can be obtained with large forgings

Table 3—Designs employing high duty steels in comparison with present standard practice

Scheme	Steels employed (see Table 2)		Remarks in respect of materials	Suggested permissible limit of loading for high quality finish with 15 deg. helix $P/De^{2/3}$					Suggested permissible relative maximum value of P for any one tooth form and pitch and any one class of gear	Based on single reduction gear design			
	Pinion	Rim		Single reduction $\delta=1{\cdot}2$	Ocean-going double reduction: Primary $\delta=1{\cdot}2$	Ocean-going double reduction: Secondary $\delta=1{\cdot}44$	Naval $\delta=1{\cdot}2$	Values relative to Scheme B		Relative weight of complete gear unit	Relative cost per ton: Pinion and rim forgings with heat treatment	Relative cost per ton: Complete gears	Relative total cost of complete gears
From Fig. 1 for reference 1928 design	II	I		(110)				(0·49)		155	95	95	147
1949 design	II	I		(165)				(0·73)		115	95	100	115
A	II	I	Standard merchant practice	200	180	200	270	0·90	1·00	110	95	100	110
B	III	I	Previously restricted to naval practice	225	200	225	300	1·00	1·00	100	100	100	100
C	IV	III	Advanced design	270	240	270	360	1·20	1·50	85	160	115	98
D	V	IV	Limit proposal avoiding profile grinding of teeth	360	320	360	480	1·60	1·50	70	190	125	87
E	VI	V	Proposal involving profile grinding of pinion teeth	450	400	450	600	2·00	1·25	60	200	140	84

centration of loading due to distortion.

Table 1 gives an indication of the limited increase that has been adopted in the loading factor over the last twenty-five years, employing wheel rims of 31/35 tons per sq. in. U.T.S. and pinions of 45 tons per sq. in. U.T.S. nickel steel, normalized or oil hardened and tempered.

APPENDIX 4—MATERIALS

Typical steels suitable for development as gear elements, together with their physical properties, are given in Table 2, in which are also included materials commonly employed today.

It will be appreciated that when adopting higher duty steels for increased gear loading, two principal features have to be borne in mind, namely, the resistance of the teeth to bending fatigue, which calls for a high yield strength in conjunction with a high Izod value, and the resistance to wear which calls for a high Brinell number.

Table 3 gives suitable combinations of steels for use as gear elements, together with provisional loading factors for high quality gears. Research now in progress will result in more

Table 4—Comparison of gears designed for the same torque

Scheme	Repeated from Fig. 1: Year 1928		Repeated from Fig. 1: Year 1949		To limit merchant designs referred to in Table 3: A		B		C		D		E	
Element	Pinion	Wheel rim	Pinion	Wheel rim	Pinion	Wheel rim	Pinion	Wheel rim	Pinion	Wheel rim	Pinion	Wheel rim	Pinion	Wheel rim
Material (see Table 2)	II	I	II	I	II	I	III	I	IV	III	V	IV	VI	V
P.C.D., inch	13·0	162	13·0	162	13·0	162	12·4	154	11·6	144	10·4	130	9·55	119
Face width, inch	60		40		33		31·5		29·5		26·5		24	
Face width/pinion p.c.d.	4·61		3·08		2·54									
Loading coefficients P	575		865		1,050		1,145		1,310		1,630		1,920	
Loading coefficients $P/De^{2/3}$	110		165		200		225		270		360		450	
Pitch of teeth to illustrate effect of working to a uniform factor of safety in root stress as would be defined by adoption of relative maximum values of P given in Table 3, inch					0·55		0·60		0·45		0·55		0·80	

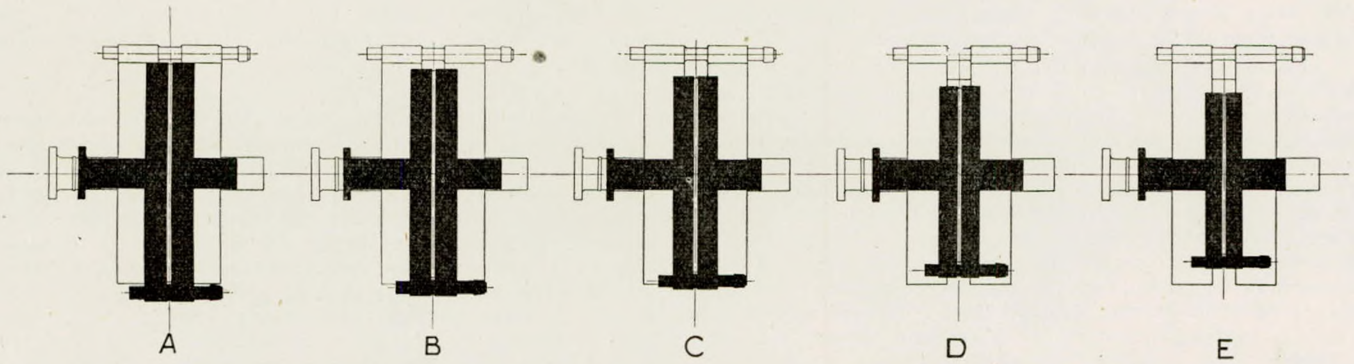


FIG. 4—*Advanced gear design compared with 1928 design for same torque*

reliable values of these factors becoming available. Relative weights and costs of units designed on the basis of these factors for transmitting the same torque as the gears referred to in Fig. 1 are given in the last four columns of the table, and Fig. 4 illustrates the resulting variation in size. To facilitate comparison relevant particulars of the gears referred to in Fig. 1 have been entered at the head of Table 3 and the outline of the 1928 design has been superimposed on each of the silhouettes appearing in Fig. 4. The comparison is further extended in Table 4 in which particulars of the gears in question are listed, the last line of the table giving an indication of tooth pitch variation as referred to in the body of the paper.

It is to be recognized that the difference between the "1949 design" and "Scheme A" represents what is today regarded as a desirable margin of safety on the suggested permissible limit.

MINUTES OF PROCEEDINGS OF THE ORDINARY MEETING HELD AT THE INSTITUTE ON 8TH MARCH 1949

An ordinary meeting was held at the Institute on Tuesday, 8th March 1949 at 5.30 p.m. R. K. Craig (Chairman of Council) was in the Chair. A paper entitled "Trends in the Development of Marine Reduction Gearing" by A. W. Davis, B.Sc. (Member) was read and discussed. Sixty members and visitors were present and six speakers took part in the discussion.

A. D. Timpson (Member) proposed a vote of thanks to the author which was accorded with acclamation.

The meeting terminated at 7.30 p.m.

Discussion

MR. C. TIMMS (Visitor) said that there were, no doubt, many points in the paper which would give rise to considerable discussion, but he would like to confine his attention to one or two points which were well worthy of amplification and further consideration. In the first paragraph of the paper the author referred to the problems of gear production as being "historical in the manner of their solution". It was generally agreed that considerable progress had been made in the production of accurate gears during the past few years and evidence of the progress made was clearly shown by the curves in Fig. 1 of the paper.

The design and development of new methods of measurement in the early war years had played a very important part in the progress recorded, yet on the other hand if full advantage was to be taken of finishing processes such as shaving, lapping and grinding there was a need for further refinement in methods of measurement to meet the more exacting requirements. The advantages which could arise from refinements in measuring technique were illustrated by the records in Fig. 5. The upper record was typical of one taken with the undulation recorder. Its traverse length was approximately 2 inch and it indicated that the gear tooth surface was relatively smooth. This result was comparable with measurements obtained from lapped or shaved gears and it would appear that the contact between the mating tooth surfaces was very satisfactory. With the aid of more sensitive measuring equipment a better appreciation of the actual contact conditions could be obtained. For example the lower record in Fig. 5, representing the same gear tooth surface, was obtained with a Talysurf instrument which had been set-up to provide a traverse of 2¾ inch. This record was clearly more informative than its predecessor since it revealed both the actual nature of the tooth surface irregularities and possible sources of error in the gear grinding machine. The shorter pitch wave was presumably associated with lack of truth of the grinding wheel and the longer undula-

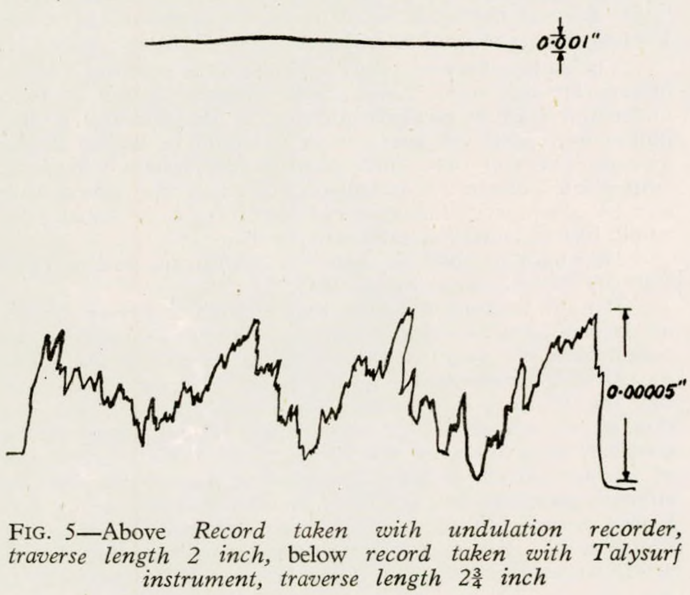


FIG. 5—Above *Record taken with undulation recorder, traverse length 2 inch,* below *record taken with Talysurf instrument, traverse length 2¾ inch*

tion which approximated to about 1 inch in length indicated lack of truth of the machine guiding surfaces.

He was not suggesting that it was essential for the measurement of marine gears to use such high scales of magnification which in the latter example amounted to 40,000×; but there was ample scope for applied research on the development of measuring equipment to provide effective control of machining processes. There was also a definite need for a unified system of measurement so that results obtained by different manufacturers could be compared on the same basis. The need for the latter was clearly evident by the numerous types of pitch comparator which had been designed in recent years. No doubt each type of instrument satisfied local requirements but some degree of standardization on instrument design was very desirable.

In Appendix 1 the author stated that the need for a high standard of accuracy for the main indexing drive of a gear hobbing machine was not so critical in a creep table machine as in a solid table machine. No doubt it would be possible to find support for this argument when considering the beneficial effects of the overlapping cuts of the hob, but if full advantage was to be taken of any benefits which might arise from a creep machine in comparison with a solid table machine, it seemed to him that the standard of accuracy of the indexing worm should be the same for both types of drive.

A further point which had some bearing on this problem was the fact that the creep ring was normally cut in its own cradle in the gear hobbing machine and the periodic errors of the indexing worm might be reflected in the teeth of the creep gear. In the case of straight spur teeth the worm error would give rise to a variation in tooth form and the resulting irregularity in the table movement would be similar to that of a solid table machine. For this reason alone a high standard of accuracy for the worm drive was very essential.

ENG. CAPT. T. W. ROSS, O.B.E., M.Sc., B.M.E., R.A.N. (ret) (Visitor) referring to single helical gears, said that it appeared to be impossible, with the present state of machine tools, to abolish cumulative pitch errors, although they might be reduced to a degree where they would be of no nuisance with single helical gears. However, any error in cumulative pitch with double helical gears must produce a tendency to axial hunting of the pinion, with consequent excessive tooth loadings and perhaps noise. It might be theoretically possible that the errors in each and every portion of pinion and wheel would cancel each other out, but this was not a practical solution. He would much appreciate the author's comments on this aspect of the matter, and also on any details which the author might have of the losses which might be expected due to the Michell thrusts with an installation of single helical gears.

The author described what appeared to be a perfect method of securing maximum contact and uniform loading of teeth under full load by selective shaving of the pinion. If this pinion were used for astern work it would be taking all the load on one end only, and therefore the system which the author later illustrated and described, where the astern drive was by a separate pinion, would appear to be essential. He would like the author to comment on this.

He would also like the author to confirm the date in Table 1 for the German destroyer as 1945.

If tooth loading was to be high enough to impose stresses on the root sections which might cause fatigue failure, the great benefit to be obtained from shot peening the roots could not be too strongly emphasized. With a finish by grinding, the surface was left in a state of very considerable tension, and shot peening to induce surface compression would appear to be essential; when the work was all machined, the dangerous state of surface tension induced by grinding was absent, but the strength against fatigue failure of the root section could be very largely increased by the state of surface compression induced by shot peening. It was of historical interest that the then Engineer Lieutenant-Commander Barry Hocken, R.N., proposed to the Admiralty in December 1915, to shot peen by projecting balls either from a centrifugal projector or by steam, air or magnetically.

CAPT.(E) L. A. B. PEILE, R.N. (Visitor) said that briefly, the author had indicated the advantages which were now available to everybody who demanded them. The first advantage was the assurance of reliability and long life, and the second a reduction in first cost, where full advantage was taken of the design of improved gear-cutting technique. The third advantage was a saving in space and weight, and he thought that the author might rightly have claimed a further advantage in the greatly increased quietness of operation. Accurately cut gears were surprisingly quiet, and every operator would agree with him when he said that, whatever the standard of the engine-room crew, whether good, bad or indifferent, they would keep watch better in quieter conditions.

These advantages could be obtained only by the highest standards of gear-cutting, and in Appendix 1 the author had laid down the essential items, with which they in the Admiralty most fully concurred. They were, first, that the gear hobbing machine should be up to the highest grade laid down in the British Standard Specification; secondly, that the machine should be housed in temperature-controlled conditions; and thirdly, that a post-hobbing process should be employed. Nothing less would suffice. So far as he knew, there were at present only about four gear hobbing machines in the country, or at any rate large ones, which were fully up to the British Standard Specification Grade A. There were, however, quite a number of other machines on which considerable overhaul and refinement work was being done, and which they believed would come up to B.S.I. standards in a reasonable period of time; but there were many more machines which were a long way below standard, and which must be overhauled if they were to be capable of producing good gears. He knew that full order books made it very difficult to arrange to lay off a machine for overhaul, but he hoped that no one would advance the argument that a sub-standard machine was perfectly satisfactory for the class of work which it was called upon to do.

The author had shown most clearly that it was a bad business proposition to order a gear of poor accuracy the performance of which was safeguarded by a large factor of safety. Anyone adopting such a policy was paying more for a less reliable article than the engineer whose specification required the highest gear-cutting standards. He hoped that there would be continuous pressure to bring more and more hobbing machines up to full British Standard Specification.

There was one point on which he was prepared to argue with the author, namely the remarks in the paper about post-hobbing processes. The author showed such a marked preference for gear shaving as to give the impression that gear lapping was hardly in the picture. That might be a natural point of view for anyone who was an enthusiast for shaving to adopt, but as yet there was no evidence to suggest that the results of one process were greatly superior to those of the other. His personal opinion was in favour of shaving, as for comparable results it was quicker; but lapping was not ruled out, and was, he believed, capable of giving extremely good results.

COM'R(E) J. H. JOUGHIN, R.N. (Visitor) said that he would first comment on three points of detail. The first concerned surface-hardened gears, which were now being successfully case-hardened and ground up to a diameter of 5 feet, which made their employment for the wheels of primary gears quite possible.

The second was the adjustment of helical angle where he would prefer to think that there were appreciable gains to be made when the length of the pinion helix was even less than 2½ times its diameter. While he fully agreed with the author about the value of doing this, he felt that there might be a great deal of value in testing out the results of doing it. He quite agreed that the mathematics could be worked out, but they were starting from an at present unknown position with regard to tooth load concentration, and therefore the advance was unknown until they had tried it. As an engineer, he suggested that if they tried it out on shore they would adopt the advantages to be gained from it with much greater confidence and much sooner.

With regard to single helical gears, he thought that in the past they had regarded them as being the natural corollary of grinding, because they fitted in with that process; but they had recently been examining closely the problem of boring some of their gear cases, both double helical and single helical, and they could not help being struck by the greater ease of adjustment with single helical gears as regarded matching their helical angles. The mere fact that they had always thought in terms of double helical gears should not prejudice them from adopting ground gears simply because they were single helical.

He felt that the author had done a great service by giving a glimpse of what was now available and of which advantage could be taken, but, while taking a survey of what might be possible in the future, he would suggest that they should also look more closely at the possibility of grinding gears. He had already indicated that case-hardened and ground gears might be used for the primary wheels of double-reduction gears, and he suggested that the author might give in his Appendix 4 another scheme, E(2), where the primary train, both pinion and wheel, would be case-hardened and ground, and the limit of loading of this train would be, say, 600 for merchant ships.

He would suggest to those who were thinking of bringing their gear production plant up to date by some post-hobbing process that they should look at the possibility of the grinding process. The author pointed out that ground gears were sometimes referred to as gears which need not necessarily be finished by grinding, namely air-hardened or rim-hardened at the steelmakers. He himself would like to look at it from the other point of view, that with hard material such as the author's No. V in Table 2 grinding might be a process which made it possible to keep closer control of the accuracy and of what one was doing with that material; the simplicity of the grinding machine, as opposed to the hobbing machine, might make it possible to achieve and maintain the desired accuracy with these hard materials at less cost and trouble. He would greatly appreciate the author's comments on the future use of grinding.

DR. W. A. TUPLIN, M.I.Mech.E. (Visitor) said the author had rendered a service by presenting in the paper a number of important facts in a nicely concentrated form. Many of the things which he said were obviously true, and a number of them were, one might say, not quite so obviously true. For instance, on the second page of the paper, in the paragraph headed "Tooth Depth", the author said that a depth about 20 per cent greater than the British Standard was possible. Personally, he thought that that was true. It must not be assumed that there was any implied criticism of the British Standard Specification in question, which had to cover gears of all sorts, whereas the field of marine propulsion gearing was somewhat more restricted and might permit a greater tooth depth than suggested by the British Standard to be adopted without serious detriment. If one did adopt that greater depth, the question of what pressure angle one should use was not technically a very critical one. The author suggested 16 deg. instead of 20 deg. In the past, he had used 17½ deg. It was just a matter of adopting a compromise, and even the British Standard of 20 deg. might suffice with this greater tooth depth.

The author also made some remarks about helix angle. It was a curious fact that for gears of given diameters and widths and materials the load capacity was practically independent of pressure angle or helix angle; one could wangle these as much as one liked and get no change in load capacity. In fact, the permissible tooth load on a gear of given diameter mating with another of given ratio, with given materials, was proportional to the area of the tooth projected tangentially; i.e., if one took an axial plane through the centre of the gear, looking at the teeth they appeared as a narrow rectangle, and the permissible load was proportional to that area multiplied by the number of teeth in the gear.

The author gave in Fig. 1 the outline of a gear designed and produced in 1928, and on the right-hand side, superimposed on it in black, a gear for transmitting similar torque designed to take advantage of modern high-quality finish. The author need not have many qualms about the possibility of transmitting the load by gears of those sizes, because the stresses there were about the same as had been used for land turbine gears for about twenty years past.

On page 92 the author dealt with the adjustment of pinion helix in order to procure uniformity of tooth loading. The idea was that one assumed a load for uniform distribution on the pinion and calculated the deflection of the pinion shaft, and then if one made the helix in such a form as to be the converse of that one would get uniform load distribution when one put the gears in service. That was the sort of thing which might be decided by a student fresh from the university, full of methods of calculating deflexions under given loadings. He could calculate that easily, and he would think that it would be all right; but the author, in the course of his experience, must, like other people, have had many bitter moments when the thing which seemed bound to happen just had not happened. Personally, therefore, he would not have the author's confidence in assuming that if a pinion were made in that way the result was bound to be right.

On the subject of single helical gears the question of thrust was raised, and he wondered why more use had not been made of what he thought of as the Brown-Boveri system, in which one placed a cone on each shaft in such a way that the axial thrust on the pinion was transmitted through the cone on the pinion shaft to a corresponding cone on the wheel shaft. The system was self-locked in respect of end thrust. That had been used to a certain extent by Brown-Boveri, and seemed to have considerable possibilities even in large gears. In a particular test of the load capacity of such cones they had found that it was well up to any likely requirements. The load capacity was very great, because the surfaces were almost exactly rolling together; there was very little sliding, and it was surprising how much load could be taken between such surfaces.

On page 94, the author had something to say on the subject of research, and asked whether the word was not perhaps overstressed today. Personally, he thought that it was. The author went on "Is it not sometimes employed when what is really implied is the investigation of troubles which are known to be capable of rectification with a reasonable application of plain common sense?" It was, of course; it would be the common experience that when one had struggled with a problem and at length obtained an answer, the answer was always the obvious thing, and one felt inclined to say "Of course, I should have known that that would happen". It could be said that research was nothing more than common sense, because when one had the answer it became common sense.

It was true, as the author said, that the word "research" was used in all sorts of connexions, but it was always part of the work of a research establishment to test what one knew to be right, to be quite sure about it, and the stuff in a research establishment might be called junk if one thought that research was bunk, or one might call the research establishment a test house if one thought that research could make useful contributions to engineering progress, or one might call it an imperial organization for the conception and exploitation of scientific ideas in engineering subjects if one's main idea was to impress on the people who had the money that the control of research was a highly skilled operation and that one needed the money for it.

In Appendix 1 reference was made to the housing of hobbing machines in temperature controlled compartments, so that the variation in temperature was limited to ±1 deg. F. If that were done one could be reasonably sure that one was controlling things all right, but it would be, to use the author's phrase, "gilding the lily", because it was not necessary to control the temperature as closely as that. What was really necessary was to maintain the temperature nearly uniform throughout the machine during the period of cutting the job. It did not matter whether the temperature was 69 deg. F. or 30 deg. F. The machine was constructed entirely of ferrous substances, the coefficient of thermal expansion being about the same, and so long as there were no marked changes in temperature during the cutting of a job, or variations in temperature between different parts of the machine, the object would be achieved. It was not necessary to say that the temperature must be held within narrow limits. It would not do any harm if that were done, but there was no reason to insist on it.

On the subject of creep fractions, the author had followed the pioneer work of the late Dr. Tomlinson, and the only criticism which could be made was that the analysis as demonstrated in Dr. Tomlinson's paper was not so broad as one would like it now. Dr. Tomlinson wrote in terms of his undulation measuring instrument which traced a line parallel to the tip of the tooth, whereas the part of the tooth with which they were concerned was the flank, which was an area, and one line could give only a limited impression of the area. The particular line used was the one which the instrument liked, and actually it was not a single line, so that there was no reason for surprise if the conclusions reached from that analysis were not quite what they ought to be. Any fraction at all was better than nothing or 1, and, while the creep fraction of ½ was not the best which might be selected, it was not bad by any means, and the general conclusion now was that the best creep fraction in general was just a little off ½. To be quite rigorous, the best creep fraction of any job depended on the rate of feed used in cutting that job.

In Appendix 3 the author referred to the question of the index of the power by which one indicated the load capacity of the gear. The load capacity of gears, according to the British Standard formula, was proportional to the 0·8 power of the linear dimensions. The author agreed with that, but said that it would be much easier to work at ⅘. He himself, a good many years ago, saw a remark of the same general character in an engineering journal, and, being younger then and not so afraid of portentous phraseology, he said that this indicated "a departure from scientific rectitude".

In Table 2 the author referred to a number of materials, and his reference I, the material with 0·3 per cent carbon and apparently nothing else, left him guessing. It was the sort of thing which he would never use for gear material at all, because it was possible to get something a good deal better at no higher cost. What he would prefer was the 0·4 per cent carbon steel. It did not cost any more, it had 30 per cent greater load capacity, and it cut better, in the sense that a low carbon steel was rather apt to tear and leave a rather pock-marked surface, whereas with a 0·4 per cent carbon steel there was no need to worry about that; there was no difficulty in cutting, and quite a good load capacity was obtained.

In Table 3 the author referred to Scheme C, called "Advanced Design", in which he used a steel containing nickel, chromium, molybdenum and vanadium for the pinion in conjunction with a nickel steel wheel. He had done something similar to that for thirty years, using a nickel-chromium steel pinion with a 0·4 per cent or 0·5 per cent carbon steel, and that gave the same load capacity as was now proposed as advanced practice. He did not know whether one should introduce land practice into the discussion, but what was now suggested as being advanced practice had been common in land practice for at least thirty years, and, as he had said before, the stresses on a proposed design in marine service were nothing out of the way in land surface.

In Appendix 4 the author referred to the fact that research now in progress would result in more reliable values of these factors becoming available. Actually the stress factors were already available, at least as far as gears of small or moderate dimensions were concerned, and the only doubt was whether these factors would be realised in large sizes.

MR. R. M. MACARTHUR (Visitor) thought the paper was rather startling but was particularly valuable because it was the first, so far as he knew, which gave a definite assessment of the value of the crossed-axes process applied to marine reduction gears. A first reading of the paper tended to convey the impression—or had done so in his own case—that the author's message was that until very recently the errors associated with the marine reduction gears were exceedingly large, but that something in the nature of a revolution had taken place almost over night, and that it now behoved gear designers to bestir themselves from a comfortable sleep of some twenty years' duration and realize that the millenium, or something closely approaching it, had arrived.

He did not think that this was entirely the case. From the introduction of reduction gearing the need for the highest possible degree of accuracy was clearly recognized, and progress since then had been fairly steady, if not spectacularly rapid, particularly if it was related not to the passage of years but to the volume of manufacturing activity in this field, which had received considerable impetus at certain periods.

In this pursuit of accuracy, a new factor entered with the introduction of the crossed-axes shaving process, first in America seven or eight years ago and more recently in this country. He was under the impression that experience so far had shown that some of the claims for this new process had been in some measure substantiated, but that the final assessment of its true value had not yet been made. It was therefore of particular interest at this stage in its development to have an expression of opinion from the author, who had obviously had considerable experience of the new process. Personally, he thought that the paper would have gained considerably in value if the author had given a more detailed account of the necessary technique and the results attained. The only precise indication which the author gave of results was contained in the statement that 80 to 90 per cent of the full theoretical contact could be achieved with complete certainty, provided that certain stringent but practicable precautions were taken.

Personally, that seemed to him to be rather extreme. It should be appreciated that it meant that for the type of gears under consideration the elastic compression of the teeth under load was of the order of 0·0002 or 0·0003 inch, and that errors of much less than this were therefore significant and capable of influencing the distribution of load to a considerable extent. The author's claim implied that the total effect of all errors present, expressed as an excess or deficiency of metal at any point of a tooth on the line of contact, was well within 0·0001 inch, and personally he found that a little hard to believe. It was, of course, purely a matter of opinion, as it could not be demonstrated by any combination of measurements with existing instruments.

This result was claimed to be achieved by selective shaving, and he would like to ask two questions about that process. It was generally agreed that crossed-axes shaving would not remove undulations the wavelength of which exceeded about ½ inch. How did the author overcome this difficulty, in view of the fact that in the type of gears under consideration the wavelength would normally exceed that amount? Secondly, in selective shaving he believed that the wheel was first shaved in a straightforward manner and the pinion was then selectively shaved. The major errors were normally in the gear, not in the pinion, and it was difficult to see what correction could be applied to the pinion only to neutralize all the errors in the gear, unless the teeth of the latter were perfectly uniform and any errors were repeated exactly round the circumference at any particular section. How could the type of cyclic helix angle error generally known as "wind" be corrected?

He would like to express his own opinion of the results from crossed-axes shaving, on which he would welcome the author's comments. First, however, he would like to refer very briefly to some of the theoretical considerations involved. One might assume that no error existed in the shaving cutter, and it was claimed that, since there was no extraneous gearing between cutter and gear, no error was introduced from this source, and the accuracy of the cutter would be transferred to the gear.

That, however, required some qualification. The cutter, assuming a uniform cutting action to take place, would generate conjugate profiles in the gear teeth, but they would only be correct if the velocity ratio between cutter and gear was perfectly uniform. In general this would not be the case because of errors in the gear, and the degree of correction obtained would depend largely on the extent of the initial errors. It would also depend partly on their nature; for example, certain types of profile error might be completely removed, whereas another type might be practically unaffected. In general, the results to be expected could be summarised in the following seven points:

(1) A considerable improvement in surface finish. About this there was no question.

(2) An averaging out of pitch errors between flanks simultaneously in contact with the cutter.

(3) A considerable but not necessarily complete correction of profiles.

(4) No appreciable correction of accumulated pitch error over spans of any considerable number of teeth.

(5) Complete correction of eccentricity in hobbing as regards depth of cut, assuming the gear to be truly mounted for shaving, but no improvement in the resulting flank-to-flank pitch errors.

(6) Little, if any, reduction in undulation magnitude where the wavelength exceeded $\frac{1}{2}$ inch.

(7) No improvement in helix angle unless selective shaving was employed, which could be partially effective in certain cases.

In connexion with this helix angle correction, the author said that pinions exceeding a certain length/diameter ratio should be corrected for the effects of deflection in torsion and bending, and in a typical case described later it might be estimated that the maximum correction necessary, measured circumferentially, might be of the order of three or four tenths for torsion and perhaps three or four times that amount for bending. It might be true that theoretically the correction at any section of the pinion could be calculated with considerable accuracy, assuming any deflexion in the wheel to be uniform. The author indicated that they could be applied with complete certainty and that there was no room for speculation or research on this matter. Since the correction to be applied to any section of the pinion was a function of various powers, up to the fourth, of the distance of the section from the ends of centre of the pinion, or some other datum point, it would be interesting to know in more detail how the result was achieved on the shaving machine with such certainty. One might be excused for a little speculation about that.

In the latter part of the paper the author indicated how, in view of the increased accuracy available, the trend of development must be towards higher pitch loading and higher duty material. There followed certain proposals which merited serious consideration, but on which he would not like to comment without further study. He would, however, like to ask one or two questions about them.

The author stated that improvement in tooth form had made a useful contribution to increasing the permissible tooth loading, and outlined a range of possibilities, mentioning that considerations of gear cutting limited the depth of the teeth in relation to the pitch to about 45 per cent in excess of the British Standard form, which, it should be remembered in passing, was not intended for use in turbine gears. As the optimum tooth form was still a matter of considerable speculation, it would be interesting if the author could make more specific recommendations.

In Table 2 reference V was to an 80-ton material. The highest duty steel normally hobbed was in the 60-70 ton range, and he would like to ask whether this 80-ton material had been successfully hobbed and shaved, as one obstacle to its use was the possibility of some inaccuracy in cutting due to hardness, which might have results which more than offset the gain in other directions.

Correspondence

MR. A. HOARE (Member) wrote that it was, perhaps, worthy of note that manufacturers of large marine gears were spurred to seek greater accuracy by pressure from the Engineer-in-Chief's Department quite early on in the war, but only as hostilities drew to a close did the way through the besetting troubles become clear.

The author mentioned the reasons for large helical angle and explained the difficulty of matching pinion and wheel helices, but treated the case for single helical gears a little casually.

Sir Charles Parsons, first with the *Vespasian,* after with the *Badger* and *Beaver,* and then the *Leonidas* and *Lucifer,* established a fashion for double helical gears, but it should be remembered that the development of the Michell thrust was in hand about the same time as the gearing for the aforementioned vessels, and it was hardly likely that it would have been favoured as a means of balancing the axial loads of single helical gears.

After all, gears were then undergoing teething troubles, recognized as arising from inaccuracies in cutting, and it was reasonable to suppose that prudence was not without influence in any step taken to give continuity to development.

The fashion, once established, had remained until today, and they were still faced with the manufacturing difficulties associated with mating two rigidly connected helical wheels with their pinions. Irregularities along the length of the teeth overshadowed to some extent the equally serious problem of matching the helical angle of pinion and wheel but with more accurate cutting, which had given teeth practically free from undulations along their length, the differences in helical angle between pinion and wheel were emphasized.

A single helical gear with a small angle of teeth, sufficient only to give continuous contact and avoid any possible engagement shock (although as teeth became more perfect this engagement shock diminished), and having, in consequence, relatively small end thrust, seemed to be the next and logical step.

Matching of helical angle between wheel and pinion would, in the case of the single helix, involve something less than half the work required for a similar operation on a double helical set, with the added benefit that the pinion of the single helix set had sufficient freedom in its journals to adjust itself to any slight remaining error, be it tooth deflexion, torsion, or helical angle. This, of course, was denied the double helical pinion.

It therefore seemed that they should not turn too easily away from the benefits of single helical gears with Michell thrust to carry the out-of-balance load.

MR. S. A. COULING (Visitor) wrote that he was in agreement with the author except for the fact that instead of using a hobbing machine with "creep" mechanism, he advocated the solid table machine. Further he was unable to support the idea that shaving should be advocated as a desirable post-hobbing process without more evidence of qualification.

However, he agreed with the author that when using a "creep" machine for first cutting a marine gear it was essential to shave or lap, since the best tooth surface cut by a "creep" machine was relatively rough compared with the solid table cutting. Shaving had its place as a help to mass production but he had not been able to show improvement in accuracy of a gearing train shaved compared with the project of a properly hobbed gear on a solid table machine. He thought it was necessary to give a word of warning herewith regarding shaving. Some experiments had been carried out by the United States Navy and as reported by them in Paper No. 48A50, of the A.S.M.E., a hobbed gear was able to outlast in loading and time, three shaved gears the last of which was hobbed and shaved. The possible explanation lay in the fact that the shaving tool burnished as well as cut and it was probable that the surface was fatigued by the shaving cutter operation. It would be interesting to learn further of these experiments.

Cutting helical angles non-differentially he was able to cut correctly within 0·0001 to 0·0002 inch in a length up to 2 × 10 inch face-width and where the same machine was used for pinions and wheels and the same portion of the lead screw there was no difficulty in matching each element perfectly.

With regard to the loading coefficient he would welcome agreement upon this point. If he had any fault to find with the author's suggestion it was that he made the loading too low. There was no doubt that in the past, due to poor gear cutting, gearbox design had been far from economical and where improved technique had been practised, higher velocity and higher coefficients than in general use, had, in fact, been used quite satisfactorily. The material reference III in Table 2 had proved a very good servant for many years and with accurate cutting and matching of mating gears the surface would work-harden equivalent to case-hardening and with its tough core he was of the opinion that it would be difficult to beat.

MR. H. A. WILSON (Member) wrote that the paper epitomized the great advance which had been made in the design and manufacture of reduction gearing in the past twenty to thirty years, and it was only by such critical analyses as the author had undertaken that the further improvement he envisaged became possible. Although his firm had no reduction gearing in their fleet, he recently had made several passages on vessels so equipped and the very silent runing of modern gears was a direct reflection of the improvements already made. Even the man in the street sometimes wondered at the almost inaudible gear-boxes in the modern motor car compared with the rasping, tearing concert which was almost accepted as part and parcel of such a mechanism several decades ago.

MR. W. OWEN (Visitor) wrote that as one who is interested primarily in the gear hobbing machines themselves, it was very refreshing to note that the author emphasized the fact that "the difficulties of accurate production (of gearing) may be regarded as historical". In a paper of this nature, the gear hobbing machine maker was very often pilloried, and the change of tone therefore was very welcome. He did not lose sight of the fact, however, that the "accurate production" had been due partly to the experience and perseverance of the users of the machines, amongst whom could be numbered the author and his firm; but the struggle for yet further accuracy was still the paramount aim of the machine maker.

Concerning this point he would refer to the author's reference to sine bar corrections on the machines. The object of the correcter mechanism on a gear hobbing machine was not to correct a faulty spiral angle after the gear was cut, but to correct the motion of the hob saddle before the machine was put into production. In other words it was an attempt to carry out a precept well understood and known by the author: "obtain as high a degree of accuracy in the individual parts as is possible, after which 'correct' for the remaining inaccuracy in any manner which is available". Some of these methods would appear to violate the principles of good design, but what did that matter if better results were achieved?

The author touched briefly upon the difference between the creep and the non-creep drive, and made the case for the creep clearly and succinctly on p. 94, but in dealing with creep fractions he embarked upon a voyage with about six other authorities each with an oar and all pulling in different directions.

This question of creep ratios and creep fractions had been going on for about ten years and, with the exception of the dictum laid down in the recently published B.S.I. specification on this point, they were no further advanced. He had before him, as he wrote, a chart upon which he had marked down some of the figures which had been advocated over the last ten years by various people. There were sixteen, ranging from 0·185 to 0·74. In his own view, adhering broadly to the B.S.I. specification it mattered little what the creep fraction was so long as the creep gearing and the members leading up to it were as accurate as they could now be made.

One very large machine of which he heard had a creep ratio of 0·93 and a creep fraction of 1·0 (or 0) and produced "very good" gears, and he knew of no instance in which trouble in a gear had been attributed to an unsuitable creep fraction or creep ratio in the machine which produced it.

COM'R(E) L. BAKER, D.S.C., R.N.(ret) (Member) wrote that the opinions expressed by the author were very timely and indeed it was to be hoped that designers would take every advantage of utilizing the information given for the benefit of the completed ship.

It was perhaps not sufficiently appreciated that weight-saving was important in the steam ships of the Mercantile Marine, almost to the same extent as in the Royal Navy. This was not to suggest that naval machinery would not and should not always be more highly loaded than that for a merchant ship, because the life of a ship in the Royal Navy included only a relatively small proportion of high power steaming. It was unfortunate, from the shipowner's viewpoint, that so few of the shipbuilders took as much interest in the development of gears as the author's firm. Progress might well have been more rapid had the interest been wider.

On the figures shown by the author, the case for grinding pinions appeared to be complete when one observed that the weight was reduced by 50 per cent and the cost by 26 per cent. Unfortunately, the number of gear grinding machines of adequate performance in this country was small and it was therefore apparent that a wholesale re-equipping of the gear cutting facilities of the marine industry was essential before shipowners could expect to be able to purchase the gears they required.

He agreed with the author that the large claw tooth sliding couplings were to be regarded as out of date but unfortunately many engine building firms still preferred to produce them.

Without wishing to disagree with the author on the validity of the index of $\frac{2}{3}$ for the expression D_e, he felt that the convenience in using a slide rule was hardly sufficient to justify adopting $\frac{2}{3}$. The purchase of a log-log slide rule would make the choice of index independent of artifices. Surely the choice of index, if of serious importance, was one of those items on which research should be carried out.

The author somewhat decried the use of the word "research" and it must be admitted that "development" would frequently be the more appropriate word. It was however a fact that development could only cease to be empirical when it was fully backed by fundamental research and it had been largely the lack of fundamental research that had compelled shipowners to be conservative in their designs, for only the very few could afford to take the risk of step-by-step progress into the unknown.

With reference to the development of reverse gears, he felt that more should have been said, and indeed more work should be in progress on the development of reverse epicyclic gears. A considerable amount of information was available from German sources on the applications which they had developed, and this form of gear was particularly suitable for making real advances in both scantlings and production techniques since one was less hampered by the age-old prejudices of conventional form.

One of the advantages of the epicyclic form of gear was that it enabled the designer to utilize the airflex type of clutch so that reversing could be achieved without the cost of the inefficiency of the reversing mechanism. The airflex type of clutch had been fully tested and was capable of considerable increases in loading without prejudicing its safety.

A. H. IJSSELMUIDEN (Visitor) wrote that in Fig. 1 the finish of a gear hobbed in 1928 was compared with a modern one produced by hobbing and shaving and illustrated the valuable results of the modern methods. Would it be possible for the author to show the improvement due to modern hobbing separately from that due to shaving in order to see what the result of the modern hobbing machine was in comparison with that of the old one, and what part of the improvement was due to shaving?

A comparison of the accuracy of modern gears with gears of 1933 gave the following result. The latter gears had since that time, been in regular service in the single screw M.S. *Madoera* and M.S. *Manoeran.* In each ship two Diesel motors drove through the gear one propeller running maximum with 86 r.p.m. and absorbing 6,500 s.h.p. The gears had been made by Demag and the accuracy had been controlled with the Maag-apparatus. The pitch error was put into a diagram and then the curve of cumulative pitch error was drawn. Of one of the wheels, with a p.c.d. of 90 inch, the cumulative pitch error was 8/10,000 inch, which seemed favourable compared with the value of the cumulative error shown in Fig. 1 for the shaved wheel. It should be admitted that the last third cut of the teeth was a very fine one. According to the author's conclusion regarding the loading coefficient on accurate gears, the loading coefficient on this gear was high, P being 1,700lb. per inch width, the total face width being 31½ inch.

Mr. J. M. Newton (Visitor) wrote that he was in general agreement with the author's conclusions and his comments dealt with tooth errors in circumferential and axial directions respectively and the use of hard materials for gear wheels. Distinction should be made between errors of tooth form and pitch which acted as a circumferential direction on the one hand, and axial pitch errors and torsion and bending of the pinion on the other hand. The former added to the total useful load carried by the gear teeth further parasitic loads caused by acceleration of the gears which resulted from tooth errors. These parasitic loads, as Dorey and Forsyth* had shown might be much greater than the useful load. The latter distributed the total load (parasitic plus useful) unequally along the length of the teeth. If the teeth were rigid the inertia forces resulting from the accelerations were proportional to V^2M where V was the pitch line speed and M the mass to be accelerated. If the teeth were sufficiently flexible to absorb the acceleration elastically as a spring the accelerating forces varied as $V\sqrt{M}$. These relationships suggested that for very high speed gearing it was desirable that the deflexion of the teeth at full load should be approximately equal the circumferential tooth errors. In the case of the best modern gearing this flexibility would be small—not exceeding 0·001 inch. But even in the case of the best gearing having tooth to tooth errors of a few tenths of one thousandth of an inch, these errors were much greater than the deflexion of the teeth of British Standard Form, and therefore he thought the author was wise in recommending tooth depths 20 to 45 per cent deeper than British Standard. It would probably also be an advantage to increase the flexibility still further by making the gear irons relatively thin combined with some form of damping, and the teeth would have increased tip relief to ensure smooth engagement. It was important to secure as uniform as possible a distribution of the tooth load along the helices and the author showed how this could be done by selective shaving. This led at once to the question of the accuracy of alignment of the various shafts in the gear case. Errors in boring the gear case were likely to be greater than the hobbing errors. How were these allowed for? One alternative was to measure the boring errors and allow for them in selective shaving but this would not be advisable if an interchangeable series of gear wheels and pinions were required. Or the errors might be corrected by fitting special or adjustable bearings or by correcting the gear case itself. Whichever method was adopted it was obviously necessary to support the whole gear case in a way which eliminated deflexion of the gear case in service. Such deflexions were bound to occur if gear cases were directly supported on flexible structures such as ships' hulls—and this naturally suggested a three point support for the gear case, or mounting the gear case on a structure which did not follow twisting motion of the hull. It would be interesting to have the author's view on this question.

In all transport applications small size and small weight of the propulsion machinery were of great value. On merchant ships it allowed a greater useful load and in warships a greater fighting power. From the point of view of the user, the smaller the gear was for a given duty, the more valuable it was. Great interest therefore attached to the data given in Table III where it was shown that a hard reduction gear unit weighed only 60 per cent and cost only 84 per cent of a gear unit of the standard type for merchant practice. Hard gears having ground tooth profiles (as the author pointed out) required a large gap if they were double helical. On the other hand, they might be single helical with no gap. In all cases the gap represented excess weight, a slice through shafts, gear rims and gear case required for manufacture, but useless in performance. Elimination of the gap was therefore valuable. This consideration suggested that there was a very strong case for single helical hardened and ground gears having a small helical angle so that the thrust to be taken by the thrust block (or which might be partially balanced by the thrust of the turbine) was moderate in relation to the power transmitted.

Mr. A. Sykes (Visitor) wrote that he was in agreement with the author's view that improved accuracy in gear cutting and finishing not only made possible increased loading by virtue of the improved distribution of load but also enabled harder materials to be used which demanded accurate bedding in the first instance, as the fact that they did not bed in, or wear in, readily, would otherwise cause concentration of loading with consequent tooth breakage at a relatively early stage.

There was every justification for a harder and higher tensile material for the pinions than for the wheels, as the number of contacts which any one tooth made in a given time was greater in the member which made the greater number of revolutions per minute and, therefore, its fatigue life was reached at an earlier stage if the stress imposed was within the fatigue range.

There was ample experience of the use of high tensile steel pinions in land turbine gears where it had been regular practice for over 30 years to employ nickel chromium steel pinions of 60 tons tensile, with medium or high carbon steel wheel rims, allowing increased loading with entirely satisfactory results. The author suggested that casehardened gears might call for larger pitches; this was, he believed, beyond doubt as, with hardened materials, the allowable bending stress at the roots of the teeth increased approximately in proportion to increased hardness, whilst the allowable surface loading was more than in direct proportion to increased hardness. This was due to the fact that, in addition to the allowable pressure per unit area being increased, the width of the band of contact on each tooth was increased by the greater pressure which could be applied before the elastic limit in compression was reached, or perhaps it would be better to say the elastic limit in shear, as pitting, which was one of the commonest forms of surface failure, was generally recognized to be a shear stress failure.

The author expressed some doubt as to the validity of the figures in the last line of Table 2, but he thought in this connexion the author was tending to be over-conservative.

The figure of 55 tons U.T.S. and also the Brinell number of 500 would apply to the core only, whereas the condition of the material at the surface of a case-hardened gear had an important bearing not only on the allowable surface pressure but also on the allowable bending stress. The surface layers on quenching expanded and left the material in a pre-compressed state which tended to offset the tensile stress caused by bending, thereby increasing the effective root strength to a value higher than that indicated by the core strength.

In the case of naval gears there was a further factor which enabled smaller dimensions to be employed, namely the shorter

*Dorey, S. F. and Forsyth, G. H. 1947. Trans.N.E.C.Inst. Vol. 63, p. 267, "Some Gear Cutting Inaccuracies and their Effect on Gear Loads and Gear Noises" period of time for which full load was applied. In a merchant vessel the gears might be working on full load for longer periods but warships were on full power for short periods only and higher stresses might be imposed without reaching the fatigue limit.

Addendum correction was not very important where the pinions had large numbers of teeth, as in marine gears. It was generally felt that the all-addendum gear was over-corrected and, in his view, the author's figure of 60 per cent was a suitable one; there was, however, no particular virtue in equalizing the ratios of sliding to rolling at the commencement and termination of engagement; this had often been advocated but it was based on rather arbitrary assumptions.

It was stated in the paper that the increment of loading increased with increasing helix angle but there was a compensating factor which was not mentioned, that of increased radius of curvature on a section normal to the tooth. The net result was that helix angle did not affect the surface stress. There might be a benefit in using low angles in the case of single helical gears to reduce the unbalanced load which had to be taken by a thrust bearing.

It was an open question as to whether tip relief on helical gears was necessary. The case of straight spur gears, where the whole tooth passed instantaneously from a position where it was undeflected and carried no load to one in which it was loaded and deflected, undoubtedly called for tip relief, but the corresponding condition in helical gears seemed to be met by end relief rather than tip relief; there was, however, some doubt as to the load distribution along the line of contact in a helical gear and this subject was worthy of further research.

The virtue of grinding lay in the fact that hardened materials could be used which could not be finished by any other means but, for materials within the range which could be cut by a hob, the hobbing process gave accuracy of pitch and uniformity of helix at least equal to that of grinding, without the procedure of measuring and eliminating of errors by selective grinding, which had usually to be carried out in the grinding process.

The author stated: "... the lower the denominator of the simple fraction the wider berth it should be given". Whilst the principle which the author intended to express was, no doubt, correct, his wording might be liable to be misunderstood. The point was not that the value should be widely removed from a simple fraction, but that exact values of simple fractions should be studiously avoided. As a matter of fact a value only slightly removed from a half was probably the best.

The author had dealt with the aspect of load carrying capacity rather than noise reduction, but the same steps which had tended to make greater loads possible had helped to reduce noise. He thought the stage had now been reached when, from a noise point of view, accuracy of gear cutting had achieved a great deal and it was becoming more necessary to consider the resonance of gear blanks and casings and to carry out experimental work with a view to modifying designs, having this aspect in view.

Mr. T. A. Crowe (Visitor) wrote that there was no doubt that in the past ten years there had been a great improvement on the quality of the gears leaving the hobbing machine but he noted that the author had suggested that the high quality gears were obtained at a greater cost per ton weight than their less accurate predecessors. He felt that this increased cost per ton, was separate entirely from increased labour charges and could be possibly attributed to the use of finer feeds when cutting, but more definitely to the cost of stripping down, repairing and rebuilding of the hobbing machines, so that they could meet the tolerances as laid down for the production of hobbed gears. He wondered if the author would state how long in the past five years had the gear wheel machines, which he had been associated with, been under repair. From the money spent in improving the standard of gear hobbing machines at Clydebank he felt that the price of the original machine would soon be equalled in maintenance costs.

He was interested in the author's opinion on the principal lines of development and their influence on design. The intensity of loading per inch width had been gradually building up, and he felt that coupling this with the higher speeds of rotation in use today, gear designers and cutters would be forced to give serious thought to the materials to be used for mating gears and which led him to discuss the adoption of higher duty materials for gears.

Mr. Crowe thought that the author should have made clear that the fundamental reason for adopting materials of greater surface hardness was not only the increasing of the intensity of loading per inch width, but more so the increase in rotational speed of the turbines.

The author would agree that the use of higher steam pressures and temperatures for main machinery had forced on turbine designers a small h.p. turbine, a consequential effect of which was the marked increase of rotational speed and a greater gear speed reduction.

If harder materials were not used especially in the h.p. turbine line, failure due to surface pitting might in time appear. He felt, however, apart from the use of harder materials to restrict pitting some method of de-aerating the lubricating oil should be evolved, together with control of the lubricating oil supply pressure and spray so that no splashing back of sprayed oil on to the advancing pinion could take place.

The author in his remarks on the single helical gear, showed, he considered, some inconsistency. Surely the labour and money spent in producing gears, of accurate pitch, surface finish and tooth form was to increase the loading rate on the gear teeth. This, to his mind, suggested reduction of the face width, with the trend to the single helical gear. The single helical gear disposed of the "shuttling" of the double helical at the cost of a small thrust bearing. This effect was shown, he thought, by the single helical and ground gears with the small helical angle.

He could understand the author arranging the addendum to be a certain percentage of the actual depth of the teeth, as by this method he could adjust the ratio of sliding to rolling of the gears. This would depend, of course, on what he was taking as his criterion, viz., hertzian compressive stress or "scuffing". For the former the all addendum gear was considered the best. It was interesting to have the author's remarks on the helical angle of 12-15 deg. Modern hardened and ground gears had helical angle of about 12-13 deg.

In his reference to reversible gears the author dismissed the variable-pitch propeller as being suitable mainly for small craft and unlikely to prove widely popular for larger tonnage. Recently a pair of Kamewa reversible propellers were fitted to the motorship *Los Angeles* which developed 7,000 h.p. on each shaft. These propellers were 17 feet in diameter and were the largest fitted to any ship so far. The experience obtained with the two 3,500 h.p. Kamewa reversible propellers on the Johnston liner *Suecia* had shown during the last five years that these propellers gave satisfactory service on ocean going vessels of considerable size and there was no factor in the design of such propellers which would prevent the manufacture of the largest size required for ocean going vessels.

With regard to the hydraulic reversing gears which the author described, it was interesting to note that a reversing mechanism incorporating an hydraulic reversing coupling for astern drive and a solid mechanical clutch for forward drive was designed by the Swedish firm of Allmanna Svenka Elektriska Aktiebolaget of Vasteras, Sweden, in 1922. This design had the advantage in the forward drive that there was no loss of power due to slip so that virtually 100 per cent transmission efficiency was obtained.

The Author's reply to the Discussion

MR. DAVIS in reply wrote that in the matter of precision Mr. Timms showed most clearly the variety of interpretation that could be applied to a term such as "high accuracy" even by those following similar lines of thought. He referred to two stages in the development of accuracy, the second consisting of tracking down elusive 10,000ths of an inch but it was suggested that the process of development was more clearly seen in the form of a curve which became assymptotic to a line parallel with the time axis in respect of ordinates representing production errors or alternatively the advantages accruing from high precision; the only two divisions into which such a conception could be divided were those representing what was commercially practicable and what was commercially impracticable and it was suggested that in some respects the boundary had already been crossed. On the other hand, Mr. McArthur tended rather to confuse the issue by his statement that from the introduction of reduction gearing the need for the highest possible degree of accuracy was clearly recognized. Such need was in fact not recognized for very many years and in some quarters there did not yet appear to be full realization.

In reply to Mr. Crowe's question as to the time required for the reconditioning of hobbing machines to bring them up to the limits of accuracy now required, he could state that his firm spent two years on the reconditioning of their pinion hobbing machine, three years on one of the intermediate wheel machines, and what would amount to four years on their large wheel machine during which time the machines had been completely off production.

In respect of his remarks on temperature control, it seemed that Dr. Tuplin did not perhaps face up to the true issue. It did not matter at what temperature level stability was achieved, and it might be convenient to have different levels for summer and winter respectively, but the fact that the machine was constructed almost entirely of ferrous substance did not mean that the machine would not distort to a significant extent during changes of temperature even although these changes occurred relatively slowly. If the equipment was unable to maintain the temperature level within 1 deg. F. it might be regarded as unsuitable for hobbing practice particularly where the hobbing of wheels was involved.

Mr. Ijsselmuiden asked for records to be presented showing typical marine hobbed gears before shaving and these were indicated in Fig. 6 to the same scale and in respect of the same wheel and pinion to which the second portion of Fig. 1 relates.

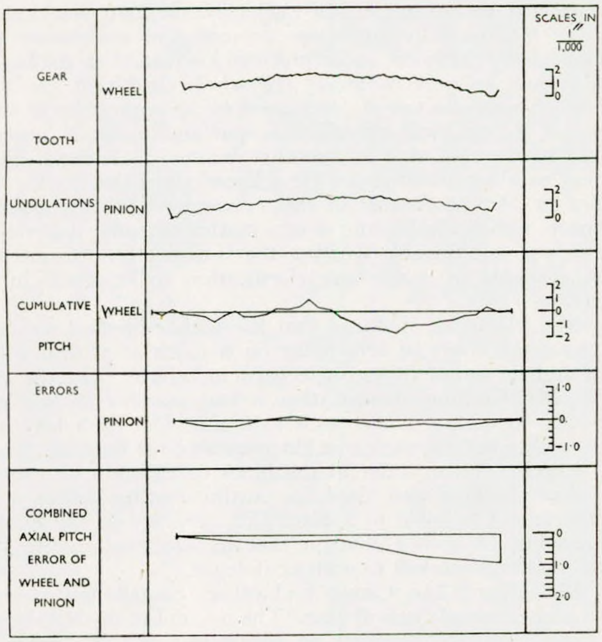


FIG. 6—*Errors recorded before shaving from gears as designed to take advantage of modern high quality finish*

Mr. Timms in referring to the accuracy required of a driving worm for creep as compared with the non-creep type of machine introduced an argument which might be carried *ad infinitum.* The non-creep machine had a characteristic which called for a worm of the highest accuracy attainable, and when this accuracy was attained it seemed unreasonable to justify lesser accuracy in the worm for a creep drive machine because for such an essential component only the best was good enough. But this did not dispose of the fact that the errors still existing in this high class product would produce a more serious effect when employed in the non-creep machine thereby provoking the quest for ever greater accuracy and so ultimately into the realms of commercial impracticability.

Dr. Tuplin, Mr. Owen and Mr. Sykes all commented on the choice of creep fraction. It would be clear that a creep fraction of 1/1 represented a non-creep drive in which the cyclic variations of the worm produced axial lines of pitch variation in the cut gear. A similar but lesser effect occurred for every creep fraction that could be expressed as a simple fraction, the effect becoming progressively less important as the denominator increased numerically. If on the other hand a value was chosen with only a slight deviation from the aforementioned particular values the resulting effect was to introduce a slight inclination in the waves of pitch variation on the cut gear, so that in any one axial plane of contact the effect of a long wave was achieved, giving banded markings which could not be corrected by shaving. Thus the effect of just avoiding a simple fraction was quite different to the effect produced by a creep having the exact fraction itself but was no less undesirable, and he would repeat that, the lower the denominator of the simple fraction, the wider the berth that should be given to it in the choice of a good creep fraction. For example, any value between 0·80 and 1·00 was bad because of its approximation to 1/1; 0·45 to 0·55 was to be avoided on account of the approximation to 1/2; 0·30 to 0·36 on account of 1/3; 0·38 to 0·42 on account of 2/5. It could be properly contended that the field of choice for a good creep fraction was severely limited but the very simplicity of the principle involved appeared to obscure its application.

In reply to the question Mr. McArthur raised regarding elimination of hobbing errors by shaving, it was again emphasized that shaving was not to be regarded as a cure for indifferent hobbing but only as a means of perfecting the teeth produced by high quality hobbing. A cyclic helical angle error was not a characteristic to be associated with a well-hobbed gear wheel and it would certainly not be possible to effect correction by selective shaving of the pinions.

The correction of tooth profile by shaving was entirely dependent on the profile of the shaving cutter. It was essential to keep control on these profiles for which purpose a simple and completely reliable type of profile measuring machine was marketed.

Mr. McArthur was correct in assuming that no appreciable correction of accumulated pitch error could be achieved by shaving but that there was almost complete correction of any eccentricity introduced with hobbing, assuming the gear to be truly mounted for shaving. In this latter connexion a gear must perforce be truly mounted when running on its own journals and this was the only correct set up for shaving.

It would appear to be the case theoretically that no undulation amplitude in excess of about ½ inch could be reduced by the shaving process and yet experience had recently been gained on a wheel hobbed by outside contractors with an undulation of 1½-inch pitch which had been completely eliminated. The degree of success achieved in this connexion was to be associated with phasing of the waves on adjacent teeth and it would not be true to say that all waves of 1½-inch pitch could be eliminated.

Mr. McArthur was correct in assuming that the helix angle was not affected by shaving unless the selective process was employed.

In reply to Captain Ross, it was to be emphasized that by using single sided cutters when shaving selectively, the astern faces of the pinion could be corrected independently of the ahead faces and consequently if any correction was to be made for pinion distortion this might be applied to suit the astern direction of loading as well as the ahead direction.

Mr. Couling cited an American publication in support of a statement that a hobbed gear was able to outlast, in loading and time, three shaved gears the last of which was hobbed and shaved. He himself had examined the reference quoted but was unable to find any mention of three shaved gears. The experiments described indicated that there were some incongruities. The photographs showed that the hobbed gear took serious punishment apparently uniformly along the helix while the one shaved gear referred to, by the incidence of blemish, gave serious evidence of banded marking. With marking of this nature it was impossible to assess the loading under which failure occurred. Furthermore, reference was made to the rolling of the surface over the tips of the teeth giving a razor edge; this was indicative of incorrect profile and increased load concentration. Again the hobbed gear was run for several hundred hours at fractional loads which would inevitably improve the quality of contact and provide a work hardened skin on the teeth. On the other hand the shaved gear was immediately subjected to overload. As a final point of criticism no mention whatever was made in the paper of the mode of finishing or the quality of finish of the mating pinions and it was always dangerous to derive any positive conclusion from such an incomplete report.

Commander Joughin asked for comments on the future use of grinding. Pending experience in this direction expressions of opinion bear too great a proportion of surmise and he would restrict his comments to the statement that while experiments that had proceeded in the hobbing and shaving of high tensile steels would indicate that grinding would only be necessitated where case hardened gear elements were employed, it might be that with the general adoption of double reduction single train and locked train gears, hobbing and shaving machines of the smaller capacities were liable to be overloaded from time to time and with the availability of grinding equipment it might prove expedient in emergency to use the process for primary trains whether or not case hardened gears be employed.

Mr. Timms, Dr. Tuplin and Commander Baker all expressed doubts as to the validity of the index $\frac{2}{3}$ employed for the expression D_e. No simple index could properly represent the various involved factors which determined the true loading of a gear tooth and the employment of any such simple index inevitably represented a "departure from scientific rectitude". Reference showed that Dr. Tuplin was not quite accurate when he stated that the author had expressed agreement with the index 0·8 which he notes was employed in the British Standard formula, and the author would remark that the authority to which he referred in this connexion was Dr. Tuplin.

In reply to the question raised by Captain Ross, he regretted that in giving the date in respect of the German Narvik Class Destroyers, an error was made. The first of this Class of Destroyers went into service in 1940.

Dr. Tuplin and Mr. McArthur both drew attention to the fact that the British Standard tooth form was designed to cover a wide variety of purposes. He recognized this point and would emphasize that his remarks were not an implied criticism of the B.S. tooth, but were intended to emphasize the undesirability of adopting this form for turbine gears. Mr. McArthur asked him to make more specific recommendation in respect of tooth form and Mr. McArthur was referred to p. 191 of the author's 1945 paper to the Institute of Engineers and Shipbuilders in Scotland, reference to which has already been quoted. Two of the forms visualized at that time have since been adopted and put into service, viz., the form having a tooth depth in relation to pitch 20 per cent in excess of the British Standard, and pitch 0·40 and 0·60 inch respectively, with flank angle of 16 deg. in lieu of 18 deg. previously advocated. These teeth have proved to be a good form for cutting, there having been no trouble with hobs such as had been experienced with deeper teeth. Service experience extending now to a period of two years had been completely satisfactory. He would associate himself with Mr. Newton's view that one of the advantages to be derived from greater tooth depth was increased tooth flexibility to absorb local errors in helical angle.

With regard to the effect of helical angle on loading, Dr. Tuplin and Mr. Sykes both commented upon the author having ignored the increase in radius of curvature which applied with increase of helical angle. It was agreed that from the point of view of tooth surface stress the increase in radius of curvature had the approximate effect of balancing the increase in load but from the aspect of tooth bending stress, increase in helical angle produced an increase in stress in the proportion indicated in the paper. It was acknowledged that compensation for this extra stress could be made by increasing tooth pitch and as it was usual to employ the minimum pitch consistent with an agreed margin of safety the position could properly be thought of as originally stated, but he agreed that it was important that the characteristic of a constant tooth surface stress with change of helical angle be kept clearly in mind.

Mr. Newton very properly emphasized the need for accurate boring out of the gear cases and that further thought be given to the method of support in which connexion he suggests a three-point mounting so that the structure of the gear case would not twist with the motion of the hull. If this principle were to be adopted it would involve certain constructional problems on account of the alteration which would occur in the deflexion of the gear case as it was progressively built up, particularly when the main wheel was loaded on its bearings, so that the final shape of the gear case might differ materially from its shape when being bored out even although the position of the supports was unaltered.

Dr. Tuplin noted that only a student fresh from the University would be likely to share the author's assumption in assuming that if a pinion were formed to counteract its calculated deflexion under uniform load the result would be bound to be right. Commander Joughin remarked on the same point and it was suggested that there was some misunderstanding of the position. When running with a broad faced gear and uncorrected helical angle, the load concentration was indeterminate. If the helix angle was corrected in accordance with the calculated deflexion under uniform loading, then the loading was known to be even along the whole length of the tooth subject to irregularities of production being appropriately small. It was if the practical irregularities were significant, as probably would be the case, that any results obtained in research on the subject would be worthless. He acknowledged the specific view raised by Mr. McArthur in this connexion that the point of support within the bearing was a matter of some uncertainty, but it was questionable whether the facilities for measurement were available to enable any clarification to be made in this direction.

Mr. McArthur deduced that the author implied an ability to shave selectively at any point on a tooth to within 0·0001 inch and he found this a little hard to believe. On the other hand, Mr. Couling claimed that it was possible to hob gears correctly to within 0·0001 inch to 0·0002 inch in a length up to 10 inch when the same machine was used for hobbing pinions and wheels. When different machines or different elements of the same machine were used for cutting mating helices it was of course not possible to achieve this accuracy in the hobbing process but he would confirm that in selective shaving these limits could be worked to with confidence.

Mr. Hoare, Mr. Crowe and others contributed views in favour of the single helical gear. The possibility of development in this direction must not be overlooked but the views the author had expressed in favour of the double helical gear were to be attributed to his objection to the added complication represented by the introduction of thrust bearings rather than the loss of power represented by their friction and which would not amount to more than about ¼ per cent of the power transmitted. Even so it was to be noted that in the present era when such stress was sometimes put upon economical consumption rates, ¼ per cent could be of appreciable significance.

In reply to the point raised by Captain Ross the shuttling characteristic of a double helical pinion could be reduced to an absolute minimum if the two helices of the gears were cut with the same direction of rotation of the table or chuck of the hobbing machine so as to introduce the same master wheel errors into the two helices and in the same phase.

Captain Peile and Mr. Wilson supplied material to compensate for an omission in the author's remarks when they commented upon the advantages of silent running which were achieved with the advent of high quality gear finishing.

INSTITUTE ACTIVITIES

JUNIOR SECTION

Lecture at Northampton Polytechnic

A lecture was delivered at Northampton Polytechnic on the 11th March, entitled "Precision Measurement" by Mr. J. Loxham. It was well attended and great interest was shown in the N.P.L. film "Precision Measurement" with which Mr. Loxham opened his talk. He followed the film with a discussion of the various types of precision measuring devices, comparing and contrasting the British instruments with those made in other countries. Some actual instruments were exhibited and demonstrated by the lecturer. At the close of the lecture some members of the audience availed themselves of the opportunity to ask questions and an interesting discussion developed. This lecture, by an acknowledged expert on the subject, was most interesting and instructive.

Mr. C. W. Tonkin (Associate Member of Council) represented the Council.

MEMBERSHIP ELECTIONS

Elected 2nd May 1949

MEMBERS

Charles Bell
Ronald Stewart Cadenhead
William Corteen Cowin
William Embleton
John Douglas Ferguson
John William Footitt
William Harold Gash
David Halkett
William Blyth Lang
John Finlay McCutcheon
Edward Perry
John Phillips
Herbert George Saunders
Wladimir Sozonoff
Stanley Charles Wiltshire, Lieut.(E), R.N.

ASSOCIATES

Eldred Lewis John Frost
Thomas William Fyson, Comm'd Engr., R.N.
Owen Charles Haines, Lieut.(E), R.N.(ret)
Clifford Angus Hardy
James William Heeson
Alexander Fraser McDonald
Douglas Royston Matthews, Comm'd Engr., R.N.
Cecil Patrick Mitchell
George Ian Quick
Vellore Ramanatha Rajagopalan
Arthur Ross
Kenneth Ross
William John Tizard, E.R.A., R.N.
Donald Wagg

GRADUATES

Hugh Grills Beck
William Cameron Lang

TRANSFER FROM ASSOCIATE TO MEMBER

William Cooper Boyd
David Brown
John McCaig
Henry McLaren
Kenneth Henry Marsh
Matthew Sampson Newton
Eric Douglas Parsons
Richard Joell Pescod
William Ronald
Ian Wans
Robert Alexander Watson

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Reginald William Parsons
Ernest George Webb

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Mostafa Nayer El-Mamoun, B.Sc., Lieut.(E), R.E.N.

TRANSFER FROM GRADUATE TO ASSOCIATE

Harold John Coles

OBITUARY

JOHN WILSON HENRY (Member 2560) was born in 1879 at Porthcawl and was educated at Clifton College. He served his apprenticeship with The Taff Vale Railway, Cardiff. He commenced his sea-going career with Messrs. Forster and Hain Co., Ltd., later joining the staffs of Messrs. Elders and Fyffes (Shipping) Ltd., and Messrs. Hallett, Patterson and Co., Ltd., obtaining his Extra First Class certificate in 1908. In 1910 he obtained an appointment as assistant to his uncle, Mr. Charles Jones, a Cardiff ship surveyor and consultant. In the 1914-18 war he was transferred from the army to the R.N.R., and served as a surveyor with the rank of Engineer Lieutenant. In 1919 he returned to Mr. C. Jones until the latter's death in 1925. After a period with Messrs. Charles Ratcliffe and Co., as Superintendent, he obtained an appointment with Messrs. Sir William Seager and Co., Ltd., as Superintendent, which post he held until his retirement in 1945. He was elected a Member in 1911 and was a Fellow of the Society of Consulting Marine Engineers and Ship Surveyors, which Society he took an active part in forming. He leaves a widow and two sons.

MARCEL PORN (Member 8514) was born in Roumania in 1884 and received the earlier part of his education at the Evangelical Realschule, Brasso, Hungary. He then took a course at the Technical College, Mittweida, Germany, where he obtained a Final Certificate in Electrical and Mechanical Engineering. Upon the completion of an apprenticeship with Julius Teutsch of Brasso, Hungary, he took up an engineering appointment in the works of Metalurgica Romana, Bucharest in 1903, and a few years afterwards became engineer to Credit Petrolifier, also of Bucharest. In that position he was closely concerned with the building and operation of Roumanian oil refineries. In 1909 he started business on his own account in Roumania as an engineering contractor and consulting engineer, which continued until 1916, when he took a commission in the Engineer Corps of the Roumanian Army. After the 1914-18 war, Mr. Porn came to this country, became a naturalized British subject and started an engineering business in London, which specialized mainly in lift installation work. The business was reorganized in 1927, under the style of Porn and Dunwoody Ltd., and, with Marcel Porn as governing director, took over the agency for Stigler lifts and for Deutz oil engines. In connexion with the latter agency, he will be particularly well remembered for his work on marine installations as well as on many other applications of the Deutz engine. At the outbreak of the last war the company concentrated on oil engine interests and on the production in this country of Deutz spares. In addition to his active association with the firm which he established, he was a director of the West Cambrian Power Company. He was elected a Member in 1937 and was also a Member of the Institution of Mechanical Engineers. He was also greatly interested in the traditions of the City of London, being a Past Master of the Worshipful Company of Gardeners. He died suddenly at his home, Little Kingshill Grange, Great Missenden, on the 21st April 1949.

ANDRÉ SÉE (Member 11867) was born in 1900 and was connected for more than twenty years with the Ateliers et Chantiers de St. Nazaire-Penhoët Co., which he joined shortly after he graduated as a naval construction engineer. For many years as chief engineer of the hull department, he was stationed at St. Nazaire, in charge of the design, construction and trials of naval and mercantile vessels constructed at Penhoët. In this capacity he was responsible for the design, construction and fitting out of the liner *Normandie*. He was appointed assistant manager of the Paris head office some years ago and on becoming general manager of the company he was associated with M. René Fould, chairman of the company. As a token of the great part he had taken in the rehabilitation of the Penhoët works, completely devastated by the Germans before the fall of St. Nazaire in May 1945, M. André Sée had recently been promoted by the French Government an officer in the Legion of Honour. He was the son-in-law of M. André Lévy, former general manager of Penhoët, who was deported during the war and died in a German concentration camp. He was elected a Member in 1948 and was also a Member of the American Society of Naval Architects and Marine Engineers.

WILLIAM WHYTE (Member 4930) was born in Dundee in 1878 and was educated at Dundee High School and apprenticed with his father's business, Messrs. J. H. Whyte and Cooper, engineers and boiler makers, Dundee. He joined the Merchant Service as a sea-going engineer and served on many of the vessels of Messrs. Furness Withy and Co. His sea-going career continued up to and including the 1914-18 war and he rose to chief engineer. After the war he accepted a shore appointment on the superintendent staff of Messrs. Furness Withy and Co., and became Superintendent Engineer in the London area. He also served in this capacity in the north of England and in Germany until his retirement on a pension in 1938. He died on the 11th April 1949 at the age of 71. He leaves a widow.