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FACTORS AFFECTING THE PERFORMANCE OF TANKER PROPULSION GEARING – AN OPERATOR'S VIEW

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read at 17.30 on Tuesday, 26th February, 1980

ISSN 0309-3948
Trans I Mar E (TM)
Vol. 92 1980 Paper 9

Factors Affecting the Performance of Tanker Propulsion Gearing—An Operator's View

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SYNOPSIS

The authors discuss the design and performance of marine reduction gears in turbine and medium speed diesel installations from the operator's standpoint. The various criteria used by manufacturers in the design of marine gearing are examined and the relationship between these and Classifications Societies' rules is discussed. Consideration is given to the various operating factors which can affect performance and comment is offered on why gearing sometimes falls short of expectations; examples from service experience are used for illustration. The part played by the operator in achieving and maintaining trouble-free operation and the more practical aspects of gear design and installation are examined. Special mention is made of investigational work and the techniques available. The importance of research laboratories in providing a link between design and practice is considered and it is emphasised how research and development can reduce uncertainties in gear design, thus increasing margins of safety. Finally, the authors look briefly into the future and attempt to define what the operator will expect from the manufacturer in terms of reliability and maintenance.

1. INTRODUCTION

Whilst several excellent papers and articles have been written on the subject of marine gearing, there has been little feedback of service experience from operators. It is the purpose of this paper to help fill the gap and to promote discussion of certain aspects upon which special emphasis has been placed.

Catastrophic failures of marine gearing are rare nowadays. Nevertheless, gradual deterioration, necessitating expensive and time-consuming replacement during the design lifetime, does occur. Such failure in service can result from design errors, mishaps during manufacture or installation, and sometimes from faulty operation. Frequently it will be due to a combination of circumstances rather than to a single cause; sometimes the responsibility must be shared.

Considerations of design, manufacture, installation and operation show that, for reliable performance, all these areas are interdependent. Particular aspects, such as surface finish, alignment and lubrication are believed to be critical and the incidents described as case histories are intended to emphasise this.

Table I lists some of the ships operated by Shell Companies. They comprise a total of nearly one million h.p. for geared turbine VLCC and about 70,000 h.p. for the general purpose ships. Most of the comments relating to design and performance are drawn from experience with these ships but it is believed that many apply also to the experience of other operators.

2. DESIGN, MATERIALS AND MANUFACTURE

2.1 Design

The basic requirement of a marine gear set is that it shall operate for up to 20 years with complete reliability and minimum maintenance. To achieve this the designer has to satisfy operating conditions which are subject to the influence of several factors peculiar to the marine environment and which are not easy to quantify. This is not, perhaps, always fully appreciated, consequently safety margins in the design can be eroded to a level at which gear wear, in one form or another, occurs.

With rationalisation of the turbine industry, including the marine sector, the user now has a very limited choice of geared turbine plant. On the medium speed diesel side the choice of gearing is wider but operating experience is more limited. It is, therefore, in the user's interest to acquaint himself as closely as possible with the design of the plant he will be operating. His specifications for main gearing are necessarily brief as it is not possible greatly to influence the basic gear design. Nevertheless, the design safety margin predicted by the manufacturer should be verified, and here the guidance provided by the various classification societies and others, such as ISO, ASME and AGMA, is of great value.

Calculations based on information from Classification Societies will show that surface finish and percentage tooth contact under dynamic conditions are of fundamental importance in gear rating.

The various Rules will permit a substantial increase in load factor, provided certain standards are met. In the case of Lloyd's Rules, for example, the increase of surface loading can be as much as 33 per cent. It is in deciding whether the standards are, in fact, being met that uncertainties can arise. Whilst the required percentage of tooth contact is stated precisely, there is no equivalent qualification for surface texture. This will be covered by a valuable addition in BS 1807, currently under revision. Doubtless the various Rules also will include similar guidance in future.

In the case of gearing for diesel engine applications, Rule may limit the load factor, depending on the type of engine/gear coupling adopted. Indeed, experience suggests that such a restriction is justified. Toms[1] includes valuable discussions on further points of design criteria which are well worth noting.

2.2 Materials

Although certain manufacturers now favour hard-on-soft pinion/wheel material combinations, through-hardened gears for merchant ships continue to be the preferred choice of most turbine/gear makers, even for the highest powers. For diesel gear sets the choice generally favours hard-on-soft combinations,

Table I. List of "L" class ships

SHIP "L" CLASS			DESIGN		
Name	Summer dwt	Entered Service	Ship Builder	Main Engine	Output kW
Lotorium	268,450	1975	H. & W.	S.L.	26,900
Lampas	318,000	1975	H. & W.	S.L.	26,900
Lepeta	318,000	1976	H. & W.	S.L.	26,900
Leonia	318,000	1976	H. & W.	S.L.	26,900
Lima	318,000	1977	H. & W.	S.L.	26,900
Lagena	317,207	1974	B.V.	S.L.	26,900
Liotina	317,588	1974	B.V.	S.L.	26,900
Lottia	317,212	1975	B.V.	S.L.	26,900
Limatula	315,700	1974	O.S.	S.L.	26,900
Limnea	315,700	1975	O.S.	S.L.	26,900
Linga	315,700	1975	O.S.	S.L.	26,900
Liparus	315,700	1975	O.S.	S.L.	26,900
Lyria	315,700	1976	O.S.	S.L.	26,900
Limopsis	315,700	1976	O.S.	S.L.	26,900
Latona	278,220	1973	C. de A.	S.L.	24,200
Leda	278,220	1973	C. de A.	S.L.	24,200
Latia	278,220	1974	C. de A.	S.L.	24,200
Latirus	278,220	1974	C. de A.	S.L.	24,200
Lucina	278,220	1974	C. de A.	S.L.	24,200
Labiosa	278,220	1975	C. de A.	S.L.	24,200
Litiopa	310,991	1977	C. de A.	S.L.	24,200
Laconica	311,861	1975	M.E.S.	K.H.I.	26,900
Lanistes	311,881	1975	M.E.S.	K.H.I.	26,900
Lembulus	254,146	1974	V.D.S.	G.E.	23,900
Lepton	318,006	1975	V.D.S.	G.E.	27,000

"F" CLASS GENERAL PURPOSE SHIPS—8 in number

Summer dwt	—	32,230	Entered service	—	1974/1976
Shipbuilder	—	H.M.V.	Main Engine	—	M.A.N./A.G.W.
			Output	—	7500 kW

H. & W.	Harland & Wolff	S.L.	Stal-Laval
B.V.	Bremer Vulkan	K.H.I.	Kawasaki—Heavy Industries
O.S.	Odense Staalskibsvaerft	G.E.	General Electric Company U.S.A.
C. de A.	Chantiers de l'Atlantique	M.A.N.	Maschinenfabrik—Augsburg
M.E.S.	Mitsui Engineering & Shipbuilding Co. Ltd.	A.G.W.	A. G. Weser
V.D.S.	Verolme Dock & Shipbuilding Co. B.V.	H.M.V.	Haugesund Mekaniske Verksted A/S

and sometimes hard-on-hard, depending on gear configuration. For diesel drive gears, there is evidence that surface-hardened pinions are needed; operating conditions can prove critical for through-hardened material combinations where the diesel drive is not through hydraulic or highly elastic couplings.

Since no single gear steel can provide all desirable properties, some compromise is necessary; particularly as identical materials for pinion and wheel should not be used. Among the many factors to be considered in selecting gear steels, the following are believed to be of fundamental importance:

i) strength;
ii) bending and surface fatigue resistance;
iii) wear resistance;
iv) compatibility with manufacturing processes.

Clearly adequate strength must be maintained over the appropriate portion of the tooth section. For low-alloy steels, the ultimate tensile strength (U.T.S.) bears a simple relationship with hardness, since the maximum hardness is essentially a function of carbon content. The section thickness over which the desired hardness is to be maintained relates to the hardenability of the steel, this being conferred by specific alloying elements and influenced by grain size.

Fig. 1 illustrates the relationship between hardenability and the tensile strength of some quenched and tempered gear steels; their chemical composition is given in Table II. For such steels, a 50 point difference in Brinell hardness between the pinion and wheel usually results in satisfactory operation, provided that steels of adequate strength have been chosen. For the relatively thin wheel rim sections a material of lower hardenability than that necessary for the pinions can usually be selected.

Modern forged steel pinions and forged or rolled steel wheel rims should have a high integrity and small grain size; the former is achieved by vacuum re-melting and the latter by close control of forging and heat treatment operations, together with the presence of grain-refining elements. Grain size can influence many properties, including toughness, hardenability, ease of heat

Table II. Chemical composition of some typical British through-hardening gear steels taken from BS 970

ALLOY	EN NO.	COMPOSITION (%w)					
		C	Si	Mn	Ni	Cr	Mo
070M55	9	0.50-0.60	-	0.50-0.90	-	-	-
530M40	18	0.36-0.44	0.10-0.35	0.60-0.90	0.90-1.20	-	-
709M40	19	0.36-0.44	0.10-0.35	0.70-1.00	-	0.90-1.20	0.25-0.35
817M40	24	0.36-0.44	0.10-0.35	0.45-0.70	1.30-1.70	1.00-1.40	0.20-0.35
826M31	25	0.27-0.35	0.10-0.35	0.45-0.70	2.30-2.80	0.50-0.80	0.45-0.65
826M40	26	0.36-0.44	0.10-0.35	0.45-0.70	2.30-2.80	0.50-0.80	0.45-0.65

treatment and fabrication. A small austenite grain size makes steels tougher and more suitable for heat treatment, being less susceptible to warping and cracking, with less retained austenite on quenching.

For these reasons, a moderately fine grain size of between 6 and 8 on the ASTM scale is desirable for through-hardened gear steels. Typical composition ranges of through-hardened gear steels used in Shell ships are shown in Table III which also summarizes the major effects of the various elements.

The Societies' Rules relating to gear design are devised to ensure a safe limit for both bending stress at the tooth root and for tooth surface stresses. The maximum permissible bending load is related to tooth geometry, whereas that for contact stress is proportional to a factor based on the maximum Hertzian contact stress. The latter takes into account gear geometry as well as the tensile strength of the lower strength material, i.e. that of the gear wheel.

With safe limits imposed, a properly constructed gear set should operate without excessive bending or surface stresses. However, if the limits are exceeded due to dynamic misalignment, with perhaps other effects such as poor surface finish, vibration, or marginal lubrication, the gears become susceptible to surface contact fatigue (pitting) or, in extreme circumstances, bending fatigue sufficient to cause tooth breakage.

With low-alloy steels, the highest bending fatigue strength in gear teeth is obtained from a steel of about 1000 to 1150 N/mm^2 (65 to 75 tonnes/in^2) U.T.S. This is due to the notch sensitivity of the high strength steel and the stress concentration effect at the tooth root. Rolling contact fatigue strength, on the other hand, increases with U.T.S. For these reasons, if high contact stresses are

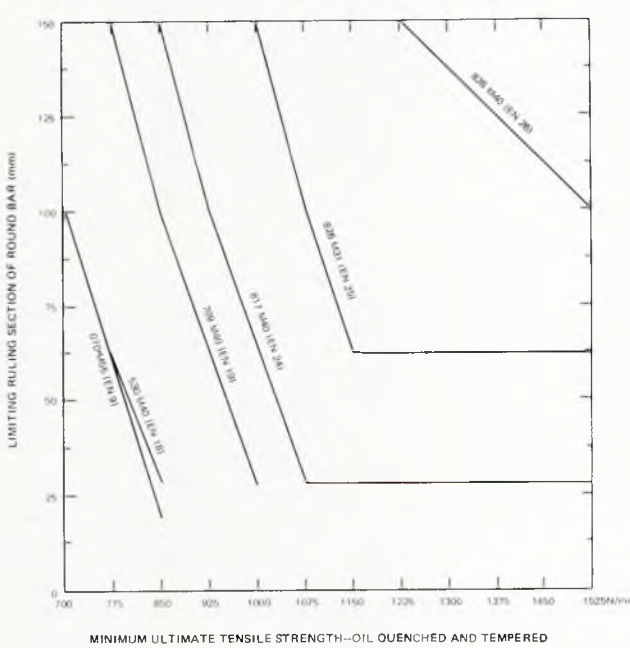


FIG. 1. Hardenability of typical gear steel (through-hardened from BS 970)

present, the gears should be surface-hardened. Processes such as carburising and nitriding also result in the generation of residual compressive stresses which improve bending fatigue strength.

Another surface treatment which looks promising in laboratory disc tests is Tufftriding, a salt-bath nitriding process which increases the resistance of carbon and alloy steels to wear, scuffing, fatigue and corrosion, also reducing their notch sensitivity[2,3]. In some Unisteel rolling contact fatigue tests, Tufftrided specimens outperformed gas-nitrided specimens and also showed a very high resistance to scuffing.

2.3 Scuffing

In normal operation, gears should be separated by an oil film bridging the asperities. Should marginal contact occur, a certain amount of protection is provided by the naturally occurring oxide films. If these are disrupted, however, metal-to-metal contact occurs with the possibility of high wear and scuffing. The latter is defined as gross damage of the working surfaces caused by the formation, and tearing of local welds.

It is a complex problem still not fully understood despite much investigation[4]. Scuffing occurs abruptly, accompanied by frictional heat and sometimes noise. The surfaces are roughened and white-etched transformed layers are produced in the steel. In severe cases, such as one to be described later in the paper, the whole load-carrying surface of the gear is destroyed.

During the 1950s scuffing of marine gears was something of a problem[5] but advances in gear technology have made it a rare occurrence nowadays. At one time EP turbine oils were used as a running-in aid but with most formulations there was a decided risk of side effects; such oils are totally unnecessary for running-in modern gearing but on occasions can be of help in healing damaged surfaces.

Through-hardened gears can be designed with a healthy margin against scuffing by keeping applied loads at a modest level; designing for as high a rolling velocity as practicable to encourage the formation of an oil film; selecting an oil of the highest viscosity consistent with the requirements of the high speed bearings; using low sliding speeds to discourage the generation of heat; and ensuring the best possible surface finish.

Teeth of small module would be advantageous for some of the above requirements but care is necessary to avoid lack of impact and bending fatigue strengths.

On the materials side, retained austenite can lead to scuffing, and its presence in the microstructure should be kept to a minimum. Alloying elements such as chromium, tungsten, molybdenum, vanadium and silicon, which do not promote austenite formation, are acceptable. However, other alloying elements, principally carbon, but also nickel, manganese and nitrogen, increase the content of retained austenite. For this reason, a sub-zero quenching treatment is necessary for case-carburised steels. However, retained austenite is not the only consideration when assessing the role of alloying elements and other factors have to be taken into account. Furthermore, the concentration of alloying element can be significant. Although not a gearing problem, this was highlighted by the "wire wooling" or "machining" failures of high chromium steels sliding at high speeds in white metal bearings. To prevent such failures it was necessary to reduce the chromium content of the steel from 3 per cent Cr to less than 1.5 per cent Cr.

2.4 Pitting

The roughness, hardness, thickness and chemical composition of surface layers, the presence of tensile or compressive stresses in them, and of inclusions or defects in or near the surface, all affect the resistance to pitting of the gear teeth.

Fatigue pitting can take a variety of forms, but all belong to one or other of the two general categories, incipient or progressive. Most incipient, or initial, pitting is attributed to the presence of high spots on the tooth flank. The pits are small and shallow, randomly distributed, usually developing just below the pitch line. They may join to form larger pits, particularly if some misalignment is present. Normally this type of pitting is arrested once the gears are run in.

Similar in category, root pitting has occurred on the final reduction wheels of some Shell ships, both diesel- and turbine-driven. This was believed to be due either to inaccurate profile or to lack of tip relief. In any event, no harmful effect on the

Table III. Typical composition ranges of specified alloying elements for some through-hardening gear steels used in Shell ships

ALLOYING ELEMENT	TYPICAL WHEEL COMPOSITIONS % WT.			TYPICAL PINION COMPOSITIONS % WT.			MAJOR ROLE OF ALLOYING ELEMENT
	W1	W2	W3	P1	P2	P3	
Carbon	0.38—0.45	0.35—0.45	0.22—0.29	0.30—0.38	0.35—0.45	0.30—0.35	as quenched hardness, hardenability and strength.
Silicon	0.15—0.35	0.15—0.35	0.15—0.40	0.15—0.35	0.15—0.35	0.20—0.40	strength and hardenability.
Sulphur	< 0.035	0.015 max.	0.020 max.	< 0.035	0.015 max.	0.020 max.	kept as low as possible to minimize embrittlement, segregation and the presence of sulphides.
Phosphorus	< 0.035	0.015 max.	0.025 max.	< 0.035	0.015 max.	0.025 max.	
Manganese	0.50—0.80	0.75—1.00	0.50—0.80	0.30—0.50	0.60—0.90	0.35—0.50	strength, high hardenability, ties up sulphur.
Nickel	< 0.6	—	< 0.30	1.4—1.7	1.65—2.00	0.40 max.	strength and toughness without brittleness, promotes fine grain size, hardenability.
Chromium	0.90—1.20	0.80—1.10	0.90—1.20	1.3—1.6	0.70—1.00	2.5—3.3	high hardenability, hardness and strength.
Molybdenum	0.15—0.25	0.15—0.25	0.15—0.25	0.20—0.30	0.40—0.60	0.40—0.70	strength, high hardenability, minimizes temper brittleness.
Vanadium	—	0.03 max.	—	—	0.05—0.10	—	grain refinement.
Specification (1)	42 Cr Mo 4	AISI 4140	SIS 2225E	34 Cr Ni Mo 6	AISI 4340	SIS 2238E	(1) there may be minor adjustments in composition according to requirements of gear manufacture. Typical material combinations are W1 P1, W2 P2, W3 P3.
Country	West Germany	U.S.A.	Sweden	West Germany	U.S.A.	Sweden	

pinions was observed and the pitting did not continue beyond a certain stage reached in the early life of the gears.

Cases of progressive pitting are almost always due to some form of overload, commonly misalignment. Unless the cause is removed, pitting will continue if the area lost by pitting is too great and is unlikely to heal, even when corrective action has been taken. Most progressive pitting is larger and deeper than initial pitting, with branching fatigue cracks extending deep into the metal. It can, however, take the form of a type of spalling or attrition of the dedendum surface, frequently leading to a distinct wear step at the pitch line.

The pitting in such cases is much finer, but large, shallow secondary pits result if the condition cannot be corrected. In the few cases of spalling within Shell Companies' experience there has been evidence that some form of superimposed vibration was at least partly responsible for the damage.

Generally speaking, contact fatigue pitting is the result of a combination of several of the following features:

a) high surface loading due to dynamic overload misalignment;
b) rough surface finish;
c) profile with high spots;
d) susceptible surface layers or inclusions;
e) inadequate or unsuitable lubrication.

Another form of damage, associated with surface-hardened gears, is ex-foliation of the hardened case. The cause may be inadequate case thickness or the presence of high residual stresses resulting from too abrupt a demarcation between case and core. This has not occurred within Shell Companies' experience of operating nitrided gears.

2.5 *Manufacture*

The performance of a gear set, assuming correct tooth design, depends greatly on the accuracy of manufacture, both of the gear elements and of the gear case. The necessary high standard of surface finish cannot be attained if the initial machining of the gear is indifferent. There is little that can be done to correct excessive undulations or inaccurate profile resulting from poor hobbing. Shaving is a finishing process and cannot adequately correct such faults. The through-hardened gears in Shell ships are hobbed and shaved to a surface finish of about 0.8 μm (30 μ in).

Even with a reasonably fine finish the shaving process can have a significant effect on the nature of the surface. Slow cutting speeds can give rise to chattering and edge build-up, causing significant work-hardening of the surface layers. A high cutting speed, on the other hand, whilst producing a cleaner cut, can generate local flash temperatures sufficient to transform the surface, sometimes producing white layers of high hardness and extreme brittleness. An example of "chatter marking" is shown in Fig. 2a whilst 2b shows the associated cracks.

Satisfactory surface finish is highly dependent on the material's hardness; for shaving the upper limit is in the region of 375 Vickers hardness which corresponds approximately to 1250 N/mm^2 (81 t/in^2) U.T.S. Consequently, from the standpoints of machining and bending fatigue strength, the effective upper limit for through-hardened steels is of this order.

In the final stages of manufacture, accurate meshing and good surface finish depend on selective shaving. The ways in which this is carried out by gear makers vary from the application of shop floor "feel" and know-how to computer control utilizing error feedback in a closed loop.

The final performance of the gear set depends as much upon the accuracy of the gear case as on the gears; errors in gear case machining which do not come to light during shop checks will almost certainly cause problems during installation in the ship.

Therefore, one of the most important factors in producing reliable gears is quality control, applied both to the gear elements

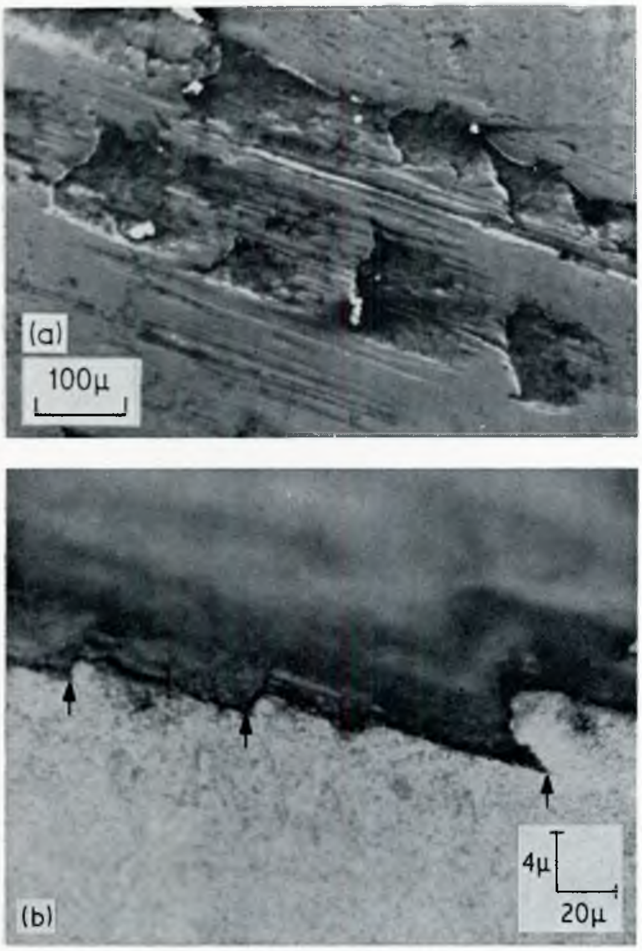


FIG. 2. a) Chatter markings on tooth flank, b) Taper section through longitudinal axis of a shaving groove, showing cracks associated with chatter effect

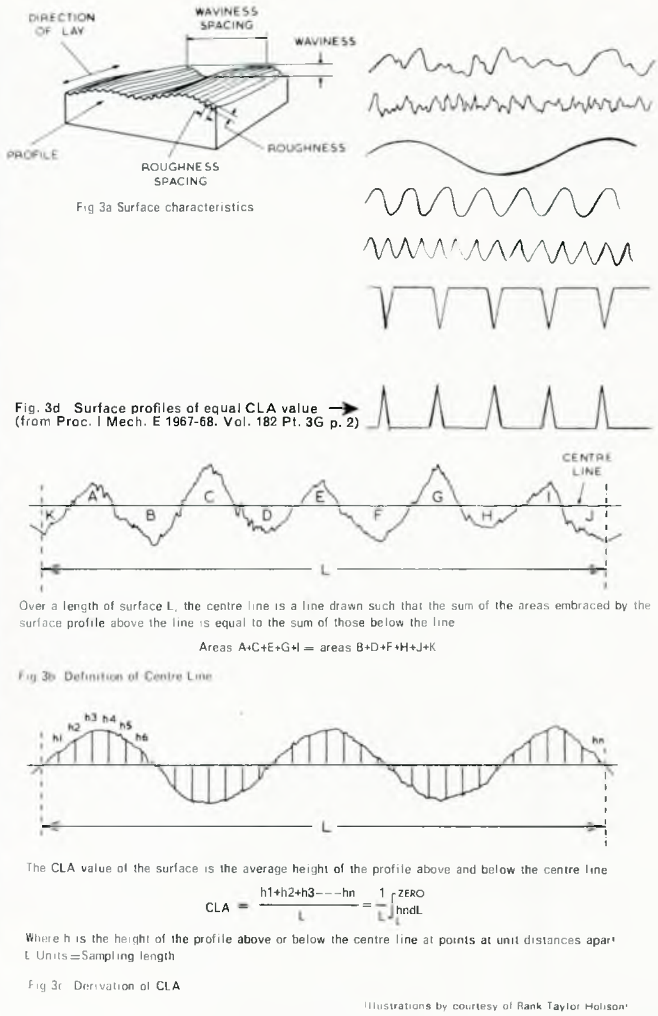


FIG. 3. Surface texture

and to the gear case. There have been one or two instances within Shell Companies' experience which suggest that this has been less than adequate. Even if they are not actually provided, the user should at least have access to the results of quality control records in terms of surface finish, undulations, profile and pitch errors, and lead angle, all of which should be within tolerance. This kind of information is not always made available, complete, to users.

2.6 Surface texture

The measurement of surface texture is of fundamental importance in the production of high quality gears. The two variables considered are roughness and waviness as illustrated in Fig. 3a. Roughness comprises the irregularities resulting from a machining process Waviness is a component upon which roughness is superimposed and is due to various factors associated with manufacture, such as chatter, vibration, machine cyclic errors, warping and so forth.

Surface texture is practically assessed by drawing a fine, pointed diamond stylus across the surface. Vertical movements of the stylus, highly magnified, are recorded graphically. At the same time, the movements are computed electronically to produce a numerical assessment of the surface, this being defined as R_a or Centre Line Average, the most commonly quoted numerical rating. The illustrations in Figs. 3b and 3c describe how CLA is defined for a prescribed sampling length. Surface texture values are normally assessed as the mean results of several sampling lengths. The cut-off length should be chosen so as to take into account both the waviness and the fine scale roughness. It should be borne in mind that the roughness will probably vary according to the direction in which it is measured and to the position on the tooth where it is measured. Where possible, traces of the surface should be obtained, as many different surface forms can have the same CLA or R_a value. All the surfaces shown in Fig. 3d have the same CLA value but they obviously would behave quite differently in practice.

In respect of gearing, criteria for surface roughness should be related to the oil film thickness. To avoid wear, scuffing and pitting, the sum of asperity heights on the working flanks of the gear teeth must be less than the oil film thickness which is normally a few microns. High quality gear specifications will, therefore, call for a surface roughness of about one micron R_a or CLA.

As mentioned earlier, surface texture requirements will be included in the revised BS 1807.

3. INSTALLATION

Two most important requirements during installation are the attainment of good static gear alignment with the least possible constraint of the gear case; and the optimum alignment of engines and propulsion shafting to the gear set.

Methods used by manufacturers to verify shop alignment differ widely but the object is to ensure correct gear-meshing with the gear case fastened down. The use of transfer marking, preferably red and blue on pinion and wheel respectively, is still the only practical method of checking no-load gear contact and the permanent tape records obtained are valuable for reference on any future occasion when gear alignment is in question.

Adjustment after the gears have entered service is most undesirable, being expensive and time-consuming; one or two GP ships had to have such attention after commissioning. It has now become clear after some years in service that the best aligned gears are those which required the least adjustment of the gear case in the ship and which most closely matched the shop readings.

Thermal effects have the biggest influence on engine/gear alignment which should of course be optimum for the hot, full-load running condition, Coupling flexibility can only take a limited account of installation errors.

Propulsion shaft alignment is still commonly checked by the jacking method although increasing use is made of strain gauges. Whilst not as repeatable as the latter, the jacking method is probably easier to apply when a check is required but experience has shown that certain elementary points should be watched if best results are to be obtained. They are summarised below.

Jacking should only be carried out in a truly hot or truly cold engine condition. Temperatures inside and outside the ship should be recorded, together with details of ballast condition which should be as near as possible to that obtaining for the original alignment.

The effect of influence numbers should be verified, if necessary, by shimming selected bearings. Before commencing the actual jacking, the main shaft should be rotated to ensure no hang-up on the thrust.

Only use jacking points at properly machined pads, which should be protected when not in use.

A spacer between the jack and shaft should be used, the top side of which is shaped to the radius of the shaft.

At least two sets of readings should be taken, the shaft being rotated through 90° or 180° between each set.

3.1 Dynamic alignment

Static gear alignment at the time of installation in the ship can be assessed readily and with fair accuracy, albeit in a qualitative sense. It is, however, the accuracy of alignment under dynamic conditions which really counts and this is much more difficult to measure. Normal practice for many years, has been to judge the percentage of tooth contact along, and over, the flanks by observing selected teeth which have been painted with a hard drying lacquer, such as Talbot Blue or Dykem. Removal of the lacquer indicates the load-carrying area of the tooth. This is, again, a qualitative assessment which can be the subject of energetic debate on board during trials.

A more modern approach is the now fully developed technique of strain-gauging selected teeth of the main wheel so that a quantitative measurement of root strain can be made as the different pinions enter and leave mesh with the wheel. This latter approach is more precise and has the advantage that load distribution across the tooth flanks can be recorded at any time during the running-in of the gears from light to full load. However, it does not provide all the information required to judge the quality of tooth contact.

Assessment of alignment quality is even more important for modern gears than for the older, more conservatively rated, units. The percentage contact over and along the teeth is precisely defined in Lloyd's Rules as a condition for permitted increase in gear load factor, together with an acceptable surface roughness which, however, is not defined quantitatively. This latter can also become the subject of debate, and will be dealt with in a later section.

The permissible load increase of 33 per cent is considerable, and if the prescribed conditions for alignment and surface roughness are not met, the effective safety margin of the gear set can be reduced below an acceptable level, bearing in mind other environmental factors which may also influence alignment. These include gear case deflection; temperature distribution inside and outside the case; shafting alignment; and hull deflections.

No matter how accurately the gear elements are made, they will function correctly only in a gear case where good alignment is retained at full-power. In essence, gear case design is a matter of achieving this, either by sufficient rigidity to resist torque effects or by deliberately allowing deflection to occur in such a way that the symmetry of the gears is not affected[6, 7].

Generally, and especially for fabricated cases, there is increased awareness of the importance of structure design. Gear case and bearing support deflections can now be predicted more closely with the aid of finite-element analysis. In addition, full-scale deflection tests of a gear case under simulated load can be carried out to confirm predictions. Yet, although detailed attention is paid to gear element design, only two of the Classification Societies mention gear case design and then only as a general reference.

Uneven temperature distribution inside a gear case can result in asymmetric expansions of pinions and/or wheel, both radially and axially, resulting in effective misalignment[8]. Sometimes observation of oil film discolouration along the tips of pinion teeth can be a clue that load distribution is not even. This is most marked in cases where an EP oil has been in use.

Incorrectly aligned propulsion shafting can result in damage to the final reduction gears. Main wheel bearing sensitivities are high and if there is sufficient disparity between the loads carried by the forward and after bull gear bearings, there could be a tendency to skew, resulting in "opposite end" overload with possible fatigue pitting[9].

Again, the hogging and sagging of the aft-end hull between loaded and ballast conditions have been the subject of much discussion in relation to gearing performance. Absolute deflections of the hull girder can be quite large. However, the machinery is sensitive only to deflections relative to a base line between the forward and after bulkheads. These are in the order of 1.0 to 2.0 mm, and should not significantly affect the load distribution. There are cases where serious problems arose but it is believed that these were the exception rather than the rule.

Over recent years, a number of comprehensive measurement programmes have been undertaken by Shell Companies with the aid of the leading Classification Societies, the object on each occasion being to check the sensitivity of the main machinery, particularly gearing, to changes in the operating condition of the ship.

All the factors mentioned above were considered in these tests and the conclusion reached was that, provided the installation of main plant is carried out to a high standard, well designed main machinery is generally more tolerant of variables than has hitherto been supposed.

It is to the credit of the Classification Societies and others concerned with measurement that both simple and sophisticated techniques have now been refined so that reliable data can be acquired and recorded in forms which lend themselves readily to simple or in-depth analysis.

4. LUBRICATION AND MAINTENANCE

The lubricant should be treated as an integral part of a geared turbine system, its physical and chemical properties being considered in the same way as those of a gear steel. It is disappointing that, apart from general reference to lubrication systems, amongst the Classification Societies only Norske Veritas make specific mention of lubricant characteristics.

The oil both lubricates and cools the gears, in addition protecting them from corrosion[10, 11]. Viscosity is a prime characteristic and should be high enough to give hydrodynamic film lubrication but not so high as to cause excessive churning losses. The full oil film gives the lowest friction and wear losses, at the same time providing the greatest protection against scuffing and pitting.

Laboratory tests have shown that pitting life increases with oil film thickness[12, 13] and this is borne out by the practical experience discussed later. Wear, scuffing and pitting can all result from inadequate or badly distributed oil supply. Initial cleanliness, both of the oil and the gear system, is also important, together with adequate cleaning of the oil in service. The presence of abrasive particles in the system can give rise to serious troubles, particularly if the gears are operating under marginal conditions of loading.

Oil viscosity for geared turbine systems has to be a compromise to satisfy the requirements of the high-speed bearings as well as those of the final reduction. Over many years this has not given rise to serious problems but, with increased gear loadings, it is believed that there is now less margin of safety in respect of protection against pitting, as opposed to scuffing.

The gear oil for diesel gear sets can be selected on its own merits. There is less demand for oxidation resistance and anti-rust performance, and the viscosity best suited to gear operating conditions can be used. For preference, a mineral oil without EP additives should be used as there is little risk of scuffing in modern gear designs.

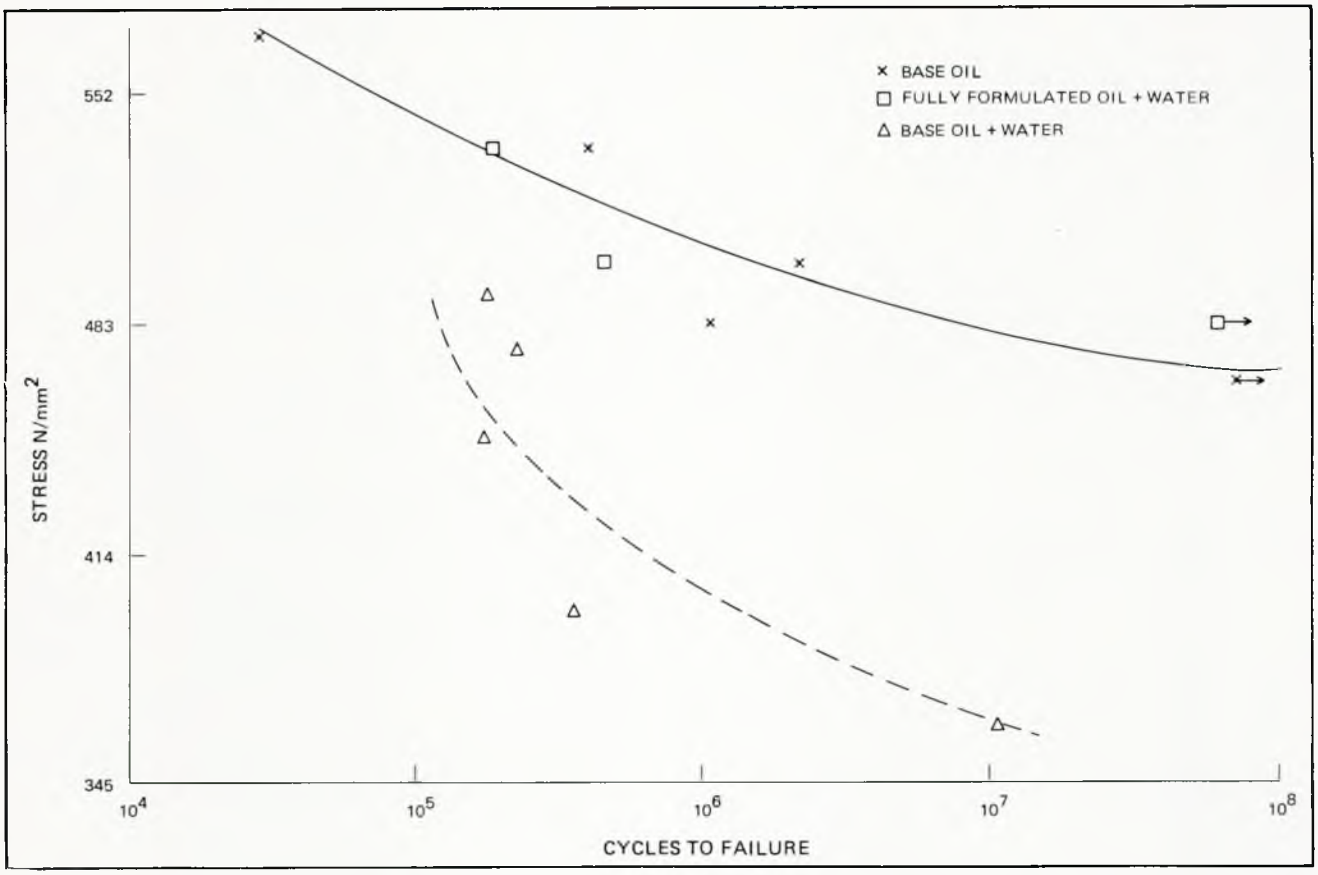


FIG. 4. Wöhler fatigue tests on En 25 steel in oil and water

4.1 Corrosion

The main corrosion problem in marine gear lubrication occurs in steam turbine systems in which some water is always present, albeit in very small quantities. Steps must be taken to keep the water content as low as possible because the results of corrosion can be serious.

This has been demonstrated in the laboratory by using the Wöhler rotating bending fatigue test. Steel specimens were subjected to fatigue at various stresses in oil or oil-in-water mixtures. Results, illustrated in Fig. 4, showed that no corrosion occurred with a fully formulated turbine oil, irrespective of the presence of water. In these tests a sharp fatigue limit was obtained, indicating no corrosion fatigue.

However, omission of the anti-rust additive from the oil formulation resulted in severe rusting in the presence of water and no fatigue limit was measurable. In practice, this would mean the danger of heavy, rapid pitting of the gear teeth and eventual failure due to corrosion fatigue.

A well maintained geared-turbine lubrication system will have a low top-up rate of around 12 per cent of the total oil charge per year. With the oxidation inhibitor present in the initial oil charge, this freshening of the oil is sufficient to give it an indefinite anti-oxidation life.

However, the anti-rust additive sometimes reaches an unacceptably low level owing to depletion from various causes. This will be apparent in the severe ASTM D 665 Procedure B Rusting Test in which polished steel specimens are immersed in a stirred oil-salt water mixture for up to 48 hours. Deterioration in performance in this test indicates the need for addition of anti-rust additive to the oil charge. This has been done successfully on occasions in Shell ships, and without side effects, thus obviating the need for a new oil charge.

4.2 System maintenance

Cleanliness is the watchword which must be used from the outset if the plant is to be really trouble-free.

At every stage of installation, cleanliness of components is of paramount importance. The time and attention paid to meticulous cleaning at the commissioning stage yields handsome dividends. Flushing procedures are now defined in considerable detail, they should be followed to the letter. In particular, cleanliness of new gear case should receive special attention; experience suggests that this is not always the case.

Once the system is commissioned and in normal operation, straightforward good housekeeping will ensure a low maintenance load.

Some water and other contaminants will inevitably be present in the system. Regular attention to filters and to centrifuging at the correct temperature, 70° to 80°C, should be part of good housekeeping. Most of the Shell VLCC are fitted with full-flow coalescing filters and dehumidifiers in addition to 5μm filters for removing solids; results have been very satisfactory.

Any components which are removed for examination should be scrupulously cleaned before reassembly and a double check made to ensure that they have been reassembled correctly. Lack of vigilance in this direction during refits can result in serious damage. Within the experience of the authors' company there have been two expensive incidents during refits, one due to incorrect assembly of the lubricating oil supply and one in which a component was actually omitted from reassembly.

Routine checks of oil condition are a good general guide to system cleanliness but any abnormalities in smell or appearance of the oil should be followed up immediately with more comprehensive tests.

4.3 Inspection

Gear inspections are valuable only when carried out thoroughly at intervals of not less than six months. More frequent inspections are not usually necessary and invite contamination. The first signs of trouble can easily be missed if the inspection is too infrequent or not systematic. This is particularly true of the main wheel teeth to which access is sometimes limited.

In cases of fatigue damage, it is often desirable to monitor the progress of pitting. This is best achieved by using blue tape records from one or more marked teeth. From carefully prepared tapes, it is possible to plot the area lost through pitting against running hours.

For detailed investigation, silicone rubber replicas have considerable advantages over resin materials. Silicone rubber produces highly accurate negatives of machining marks, surface flow and micro-pits, and unlike resin can reproduce re-entrant pits; exceptionally fine detail of the surface can be obtained. A positive can be cast from the original and aluminized to provide an almost exact duplicate of the actual gear surface.

Another useful technique is the use of acetobutyrate plastic film (Triafol) which is applied to the gear surface after flooding with dimethyl ketone and allowing the film to dry before removal. The main difficulty with Triafol is its propensity to curl. Surface details, however, can be observed if a slide is prepared and the plastic film projected directly on to a screen, or by aluminizing the surface and examining it under a microscope. The replica is, of course, a negative of the gear surface.

For examining debris associated with damage to the gears, magnetic plugs correctly positioned in way of the lubricant flow are a simple means of obtaining samples. Examination of such debris can often identify the wear process e.g. abrasive, pitting, etc. Where the gears (or tooth surfaces) differ substantially in chemical composition, it can be used to identify which gear is suffering damage. The major benefit of magnetic plugs is that samples can be taken while the gearing is in operation.

The scanning electron microscope has in recent years proved an extremely useful instrument for gear wear investigation. Because of its considerable depth of field and great range of magnification when compared with optical microscopes, the instrument is ideal for topographical studies. Moreover, with its quantitative analytical facilities, system debris and reaction products can often be identified.

4.4 Lubricants testing

Regular examination of representative oil samples, drawn whilst the system is in operation, is useful as a general check on oil condition. Service hours and top-up quantities should always be supplied with the test sample which, in the case of Shell ships, is submitted to the laboratories for routine checking of viscosity, acidity and water content at six-monthly intervals.

4.5 Spectrographic oil analysis programme (SOAP)

The spectrographic analysis programme was developed to monitor the development of wear in critical applications such as aero engines, the aim being to predict failure before it occurred so that remedial action could be taken in good time.

The test itself is carried out by burning a small representative sample of oil in an a.c. arc struck between a graphite rod and a graphite disc, wetted by the oil. The radiation emitted is analysed in a computer-controlled spectrometer and the amount of trace metals in the oil determined by comparison with standard samples. Attention is drawn to the presence of any metal in amounts greater than previously agreed limits, for example, 10 p.p.m. for iron, which generally indicates the extent of wear of oil-wetted surfaces.

The method is precise, cheap and rapid and could be useful for marine applications.

4.6 Ferrography

This is a comparatively new monitoring technique presently being evaluated by Shell Research Ltd. It is a means of measuring the severity, and identifying the nature, of wear processes. The technique is based on the extraction of magnetically susceptible wear particles from a lubricant under the influence of a magnetic field.

In the ferrographic analyser a metered sample of diluted oil is passed down an inclined slide, placed between the poles of a permanent magnet. This develops a high magnetic gradient along the length of the slide so that wear particles in the oil samples are subjected to an increasing magnetic field as they move down the slide. The particles deposit out according to volume and magnetic susceptibility, the smaller and less susceptible being deposited at the end of the slide.

Non-magnetic materials which have been subjected to wear are also deposited at the exit region because small particles of high magnetic susceptibility get attached to them during the rubbing process.

The resulting ferrogram is examined for particle type, size and concentration. By examination of its optical density the severity of wear can also be determined. A "wear severity index" can be calculated by using a direct-reading ferrograph.

The main constraint of the technique is the upper limit of particle size which can be dealt with, so that it may have to be used in conjunction with magnetic plugs.

The foregoing techniques provide in the laboratory as comprehensive a picture as possible of what is happening to the plant in service. This is a valuable part of a general research programme aimed at improving gear performance.

5. CASE HISTORIES

Experience over many years of operating turbine plant ranging between 5600 kW (7500 shp) and 16,400 kW (22,000 shp) has demonstrated a high reliability of the gearing. However, during the tanker building bonanzas which were to prove so costly in later years, sizes and powers increased considerably. It was in these ships, built between 1966 and 1974, that hitherto unsuspected design weaknesses appeared, both in the hull and in the engine room.

Design deficiencies in gearing are rare but those which occur can prove exceedingly costly to the operator as well as to the builder. The now well documented failures of epicyclic units[14] at that time form a good example. However, the use of nitrided materials for the epicyclics proved perfectly satisfactory.

The load carrying capacity of nitrided gears is believed to be well within requirements and no sign of distress due to overload has ever been observed on the very many gears examined. In one or two cases, removal of the white layer occurred but this did not, with one exception, prove troublesome. Generally, careful honing of the affected area was sufficient to restore the finish of the working surface, after which the gear was reassembled for normal service.

In one instance, the sun wheel illustrated in Fig. 5, damage had penetrated the nitrided case; the gear had to be replaced. On other occasions, small pits appeared on the working surfaces of nitrided teeth; after careful honing there was no further deterioration. It should be remembered that nitrided gear elements are more vulnerable to handling damage, being very easily chipped if handled roughly or knocked in transit.

The use of through-hardened materials for turbine gears has given no cause for serious concern. Two cases of slow progressive pitting occurred and these are discussed below. A number of the largest gear sets incorporate fully chocked gear case designs but no problems occurred which could be attributed to inadequate stiffness, either of the gear case or of the hull.

Nevertheless, the fully chocked design is believed to be more vulnerable to hull effects such as distortion of the aft end structure in heavy seas and deflection of the ship's double bottom between ballast and loaded conditions. Successful operation demands adequate alignment of both gears and propulsion shafting in addition to a conservative rating of the gears.

FIG. 5. Pitting and white layer removal on sun wheel

A large number of the Shell ships are equipped with Stal-Laval AP engines of varying powers up to 27,000 kW (36,000 shp). These engines incorporate a four-point support for the final reduction gears. As a result of using epicyclic units for the first, and in the case of the HP train, second, reductions, the final-reduction gear case is, of course, narrower than conventional double-reduction, articulated designs. Gear case/hull interactions are therefore, minimized. An additional feature is the provision for on-site correction of alignment by manipulation of adjustable spring-boxes at each corner of the gear case. This has proved to be a quick and effective means of optimising alignment in practice.

In presenting a general account of experience with gearing, reference is made below to various problems, each of which can be related to one or more of the following areas:

a) basic design;
b) quality of manufacture;
c) installation;
d) operating environment.

5.1 Manufacturing accuracy

The two cases of final reduction pitting mentioned above occurred early in the life of the gear sets and each was carefully monitored to establish the rate of progress. Neither gave cause for real concern, particularly as both ships were slow-steaming for substantial periods. Some months' operation at full power showed that the progress of pitting was containable.

The units had been supplied by a manufacturer whose gears were suffering an unusually high casualty rate, above 25 per cent had significant damage. In a few cases replacement of the final reduction gears was necessary.

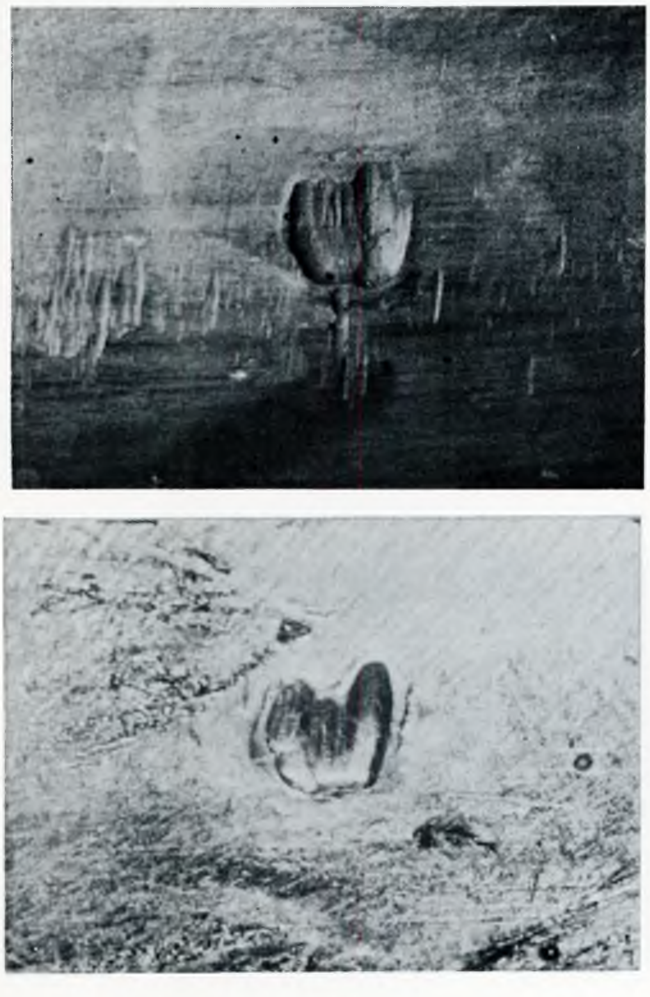


FIG. 6. Identical forms of surface blemish, different ships, different shipyards

A curious feature of these gear sets, which were all of the same frame size, proved to be heavy and extensive tracking in the very early stages of running, of the final reduction teeth, due to some form of contaminant. Although the surface finish of the affected gears appeared to be seriously damaged, no direct connection with fatigue pitting could be established. Despite the fact that the sets were commissioned by different shipyards, following strict flushing procedures, the marks appearing on the tooth flanks were strikingly similar, as can be seen in Fig. 6.

Attempts to identify the contaminants have been unsuccessful but it is known that similar so-called dirt tracking was experienced in several ships other than those of Shell.

The pitting problem has been investigated in depth by the manufacturer and others, leading to conclusions that the basic cause was lack of accuracy in gear cutting and finishing. In addition, there was evidence that the temperature distribution through the pinions and main wheel was not even, causing a tendency for heavy base loading of the double helix.

5.2 Metallurgy and surface finish

The main plant of the eight general purpose ships comprises two medium-speed diesel engines driving single-reduction gears that transmit 7500 kW (10,000 shp) to a variable pitch propeller. The original choice of oil was a leaded EP type of 48 cSt viscosity at 50°C.

In the first ship, after four months in operation at full power, rapid progressive pitting of pinions and main wheel occurred, and the oil was changed to a full EP type 80 cSt viscosity at 50°C. The change in EP formulation was only for reasons of rationalisation. The main object was to increase the viscosity and hence the film thickness. Whilst the pitting slowed down considerably, it did not cease; it was necessary, finally, to renew all the gear elements.

Operating from new with the oil of higher viscosity, the replacement gears also suffered progressive fatigue, shown in Fig. 7, Fig. 8 shows the load-carrying area lost per 1000 hours of running at full power. Of the eight ships in service, the first suffered progressive pitting in two gear sets. Three others showed fatigue damage which stabilized and the remainder were satisfactory. As mentioned earlier, the best performers were those which required the least adjustment when being installed in the ship.

The considerable investigation into these failures is dealt with here at some length because there were several factors contributing to the problem, including design, surface finish and alignment.

The design K value was 144 but, when calculated against Lloyd's 1973 Rules the allowable K value (assuming good finish) was 125, indicating a marginal gear design in respect of surface fatigue resistance.

The gears were hobbed and finished by shaving to a design surface finish of 0.8 μm CLA (32 μin). Examination of undamaged areas of the failed gears showed that surface roughness varied between 0.8 and 2.5 μm CLA. In the latter regions, peak to valley depths of 5 to 7 μm were measured. Close examination showed significantly large areas of poor surface finish, often in the region of the pitch line.

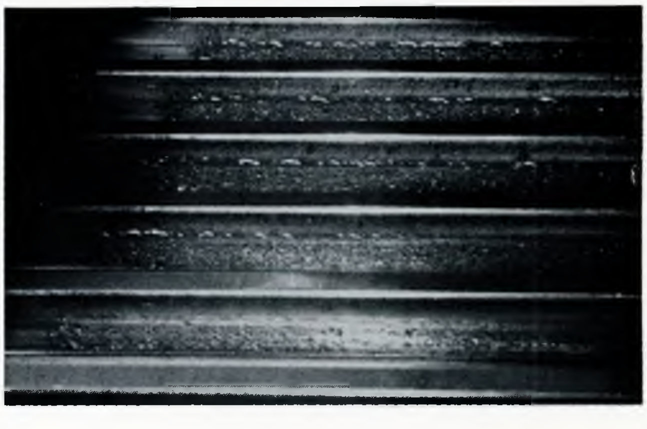


FIG. 7. Progressive pitting on replacement gears

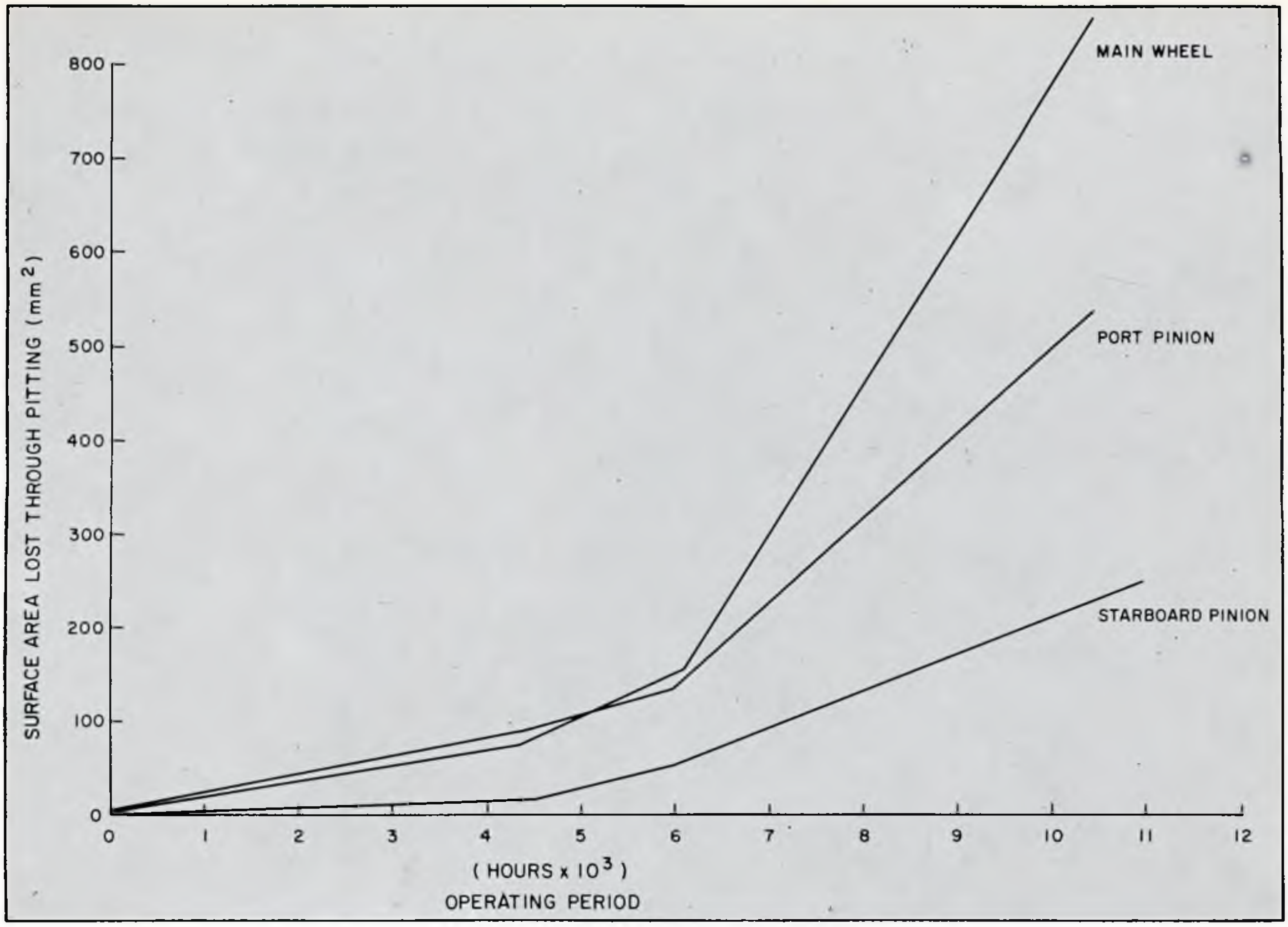


FIG. 8. Pitting rate of gears suffering progressive pitting

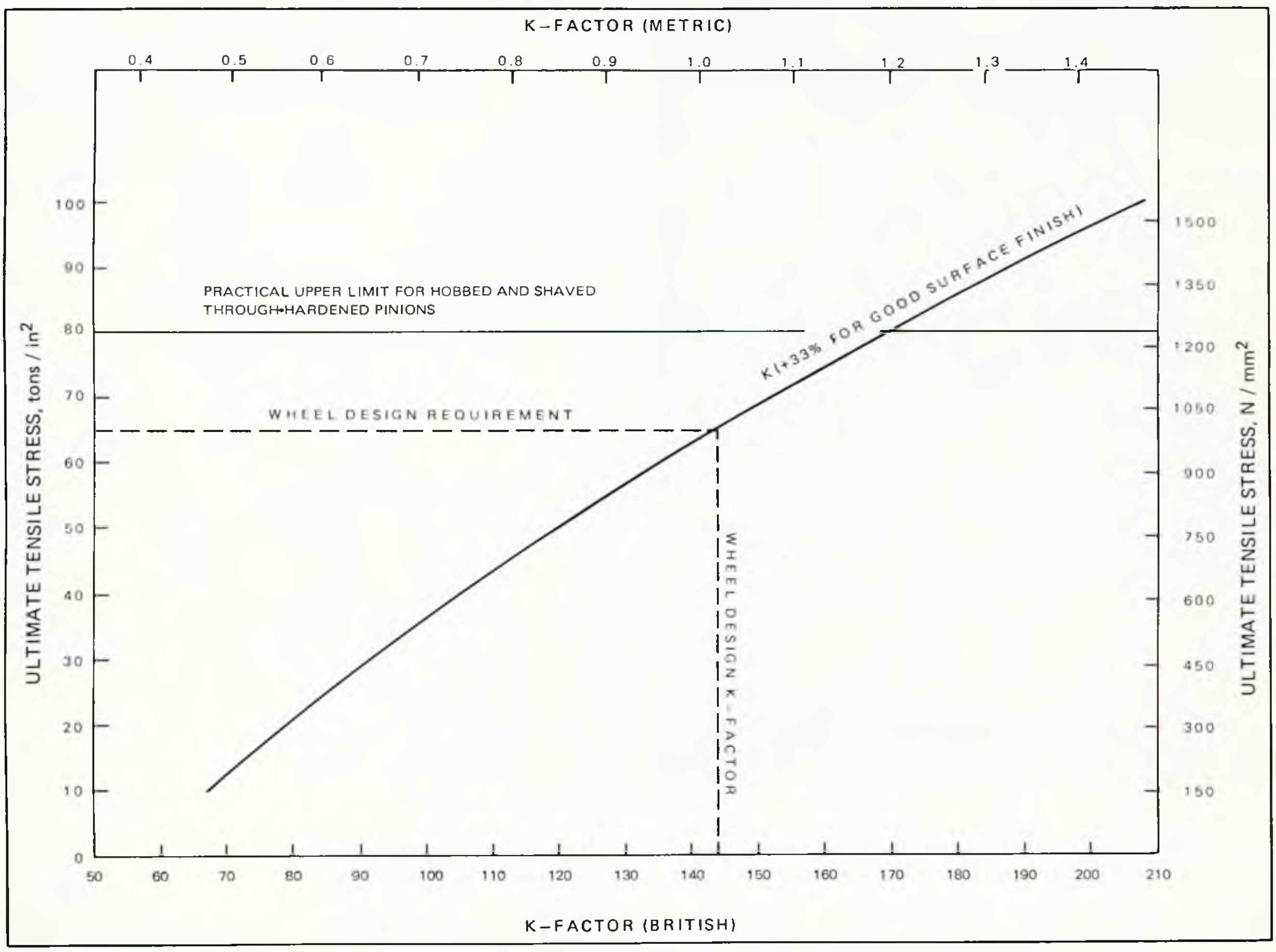


FIG. 9. Ultimate tensile stress vs K factor (Lloyd's Rules)

Since the K value depends on gear geometry it is possible, for a particular gear design, to relate ultimate tensile stress to the K value as shown in Fig. 9, which indicates that, in this case, the wheel required a steel of 1004 N/mm² UTS or 65 t/in² (293 H_B) and the pinions (based on criteria already mentioned) a steel of 1158 N/mm² UTS or 75 t/in² (340 H_B). It is worth noting that the

FIG. 10. Surface of gear tooth close to end of progressive pitting marks, showing association of pitting with deep machining marks

P – Pitch point

S – Represents sliding direction. In the addendum, sliding in the same direction as rolling ; in the dedendum, it is in the opposite direction to rolling.

↘ Represents direction of crack propagation
The distorted surface layers flow in the same direction as that of sliding

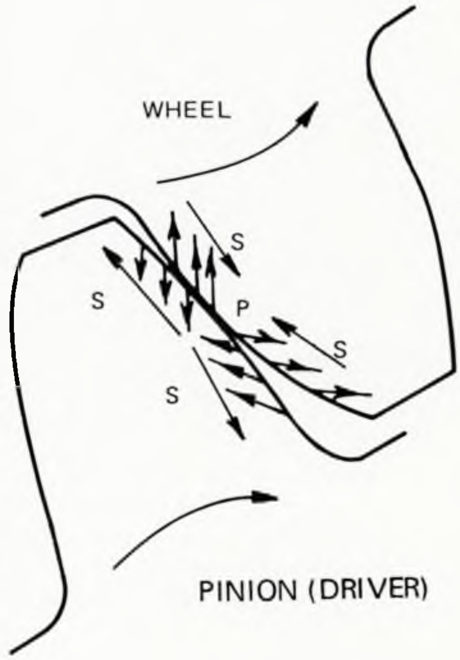


FIG. 11. Schematic representation of sliding and crack propagation directions

required minimum strength or, rather, hardness, of the pinion was such that high quality shaving would be difficult to achieve.

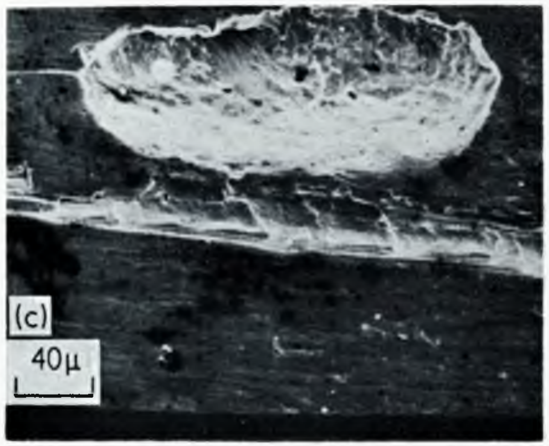
In respect of alignment, all the gear sets showed a persistent tendency to mark "heavy aft" despite deliberate attempts to bias the loading forward when carrying out static alignment. Scraping of the bearing to correct matters proved largely ineffective and it was finally concluded that dynamic effects were responsible. The exact explanation of these is proving difficult to establish.

Examination of the gears showed that pitting started in the region of the pitch line in the form of micro-pits, often associated with regions of poor machining and considerable surface plastic flow. A typical section from one of the pinions at the end of the pits, spreading aft to forward, is shown in Fig. 10. The pits had spread for more than half the face width of the gears.

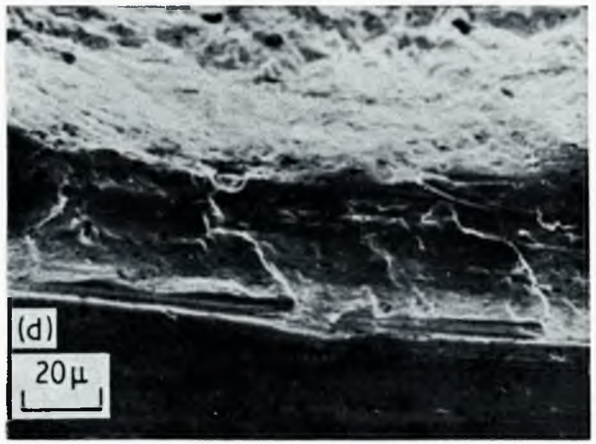
Associated with the pits were cracks penetrating in the directions shown schematically in Fig. 11, though there were isolated exceptions. Taper sections and micro-hardness measurements showed that the original machining marks were associated with regions of increased hardness, particularly at the base of the machining grooves; similar effects caused by in-service scoring were also observed.

Investigations of the surface by scanning electron-microscopy (SEM) showed fully developed pits initiated by micro-pits as illustrated in Fig. 12 and the association of micro-pits with surface flow over machining marks in Fig. 13. Transverse sections through such developing micro-pits are shown in Figs. 14 and 15 leading to the likely sequence of micro-pit formation depicted in Figs. 16 and 17.

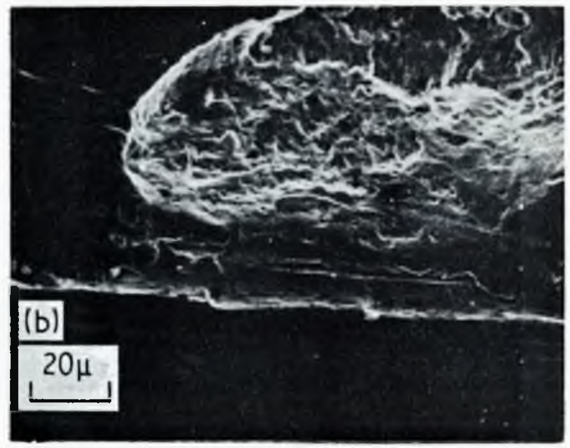
Initiation of fully developed pit by micro-pit (1)



Arrow 2 in above showing micro-pit associated with machining mark



As (c) but showing micro-pit initiation region associated with chatter marks



Micro-pit associated with machining mark (3) showing surface flow across machining mark adjacent to pit

FIG. 12. Association of micro-pits with machining marks and fully developed pits—starboard pinion

Empirical relationships have been developed for contact fatigue limits relating the stress limit to hardness and also to surface finish. Fig. 18 shows the effect of surface finish according to Takeda *et al*[15], for the gears under discussion. Since these were clearly suffering from dynamic misalignment which would effectively raise the value of S max (maximum Hertzian contact stress) in Fig. 18, pitting would not be unexpected considering the poor surface finish observed.

It is perhaps significant that this diagram also suggests that the gears might well have shown an improved performance if the surface finish had met design requirements of 0.8 μm CLA which is approximately equivalent to 2 to 2.5 μm peak-to-valley, since there would have been a larger safety margin.

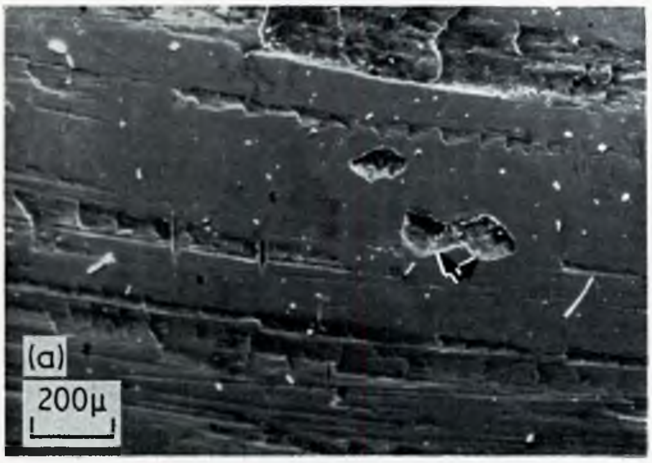
This rather full investigation showed that, in cases of severe overload, surface finish is crucial but, perhaps of equal importance, are the metallurgical changes that can occur on the gear tooth surface such as work-hardening and the formation of cracks.

5.3 Marginal lubrication

The importance of ensuring that gears and bearings are operated under normal lubrication conditions cannot be over-stressed; the penalty for neglect can be prohibitive. A good example is the heavy pitting and general surface disturbance in both the addendum and dedendum of a final reduction wheel from a 20,000 kW (28,000 shp) set shown in Fig. 19. The pinions were also damaged but not to the same extent.

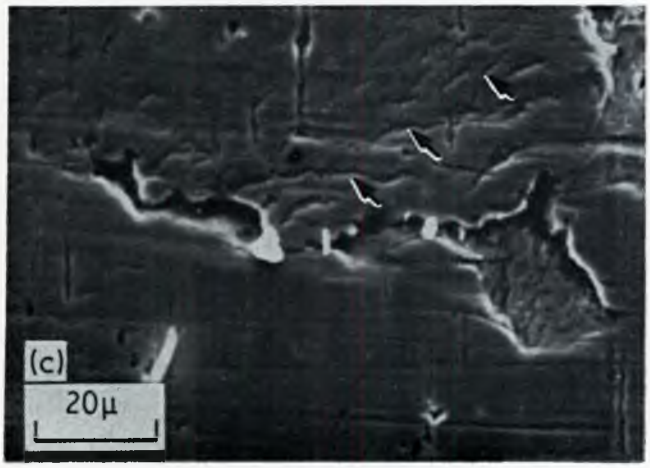
The pitting occurred due to gross oil supply restriction over a long period of operation The cause of oil deficiency is obvious from the photograph in Fig. 20 which shows the state of the oil filter fitted before the sprayers. Mutton cloth had been wrapped around the filters for flushing purposes and subsequently over-looked. It became so badly contaminated that the gauze burst, and this probably saved the gears.

Two interesting points emerge from this classic example of an operating fault. Despite the restricted oil supply, the gear had not, as might have been expected, suffered from scuffing. This illustrates the conservative nature of the gear design in this respect.



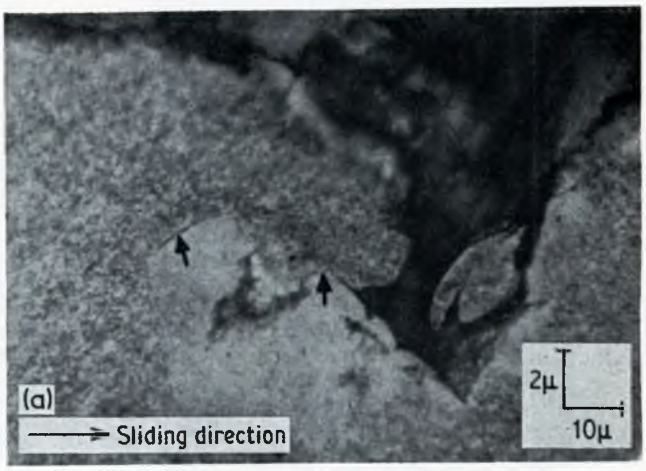
Association of micro-pits with machining marks

Higher magnification view of (a) showing surface flow across machining mark

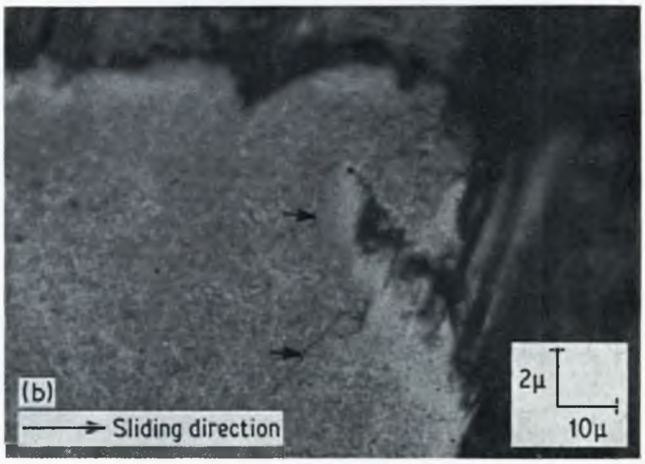


Higher magnification view of (b) showing fatigue cracks at surface of flowed layer

FIG. 13. Association of micro-pits and surface flow with machining marks in starboard pinion, forward of continuous pitting

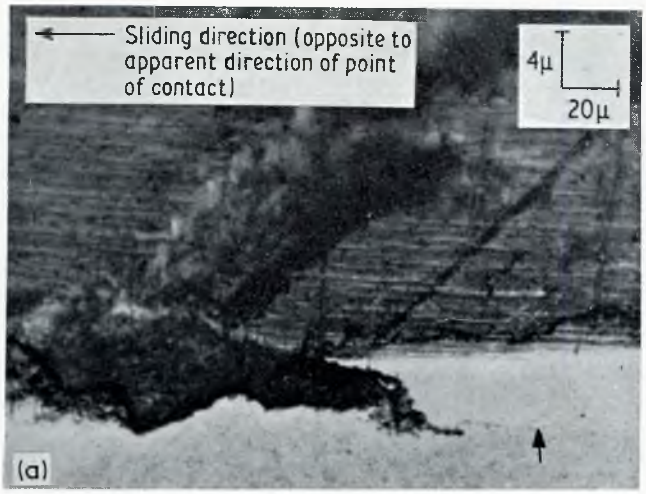


Transverse taper section through shaving marks, showing flowed layer and crack

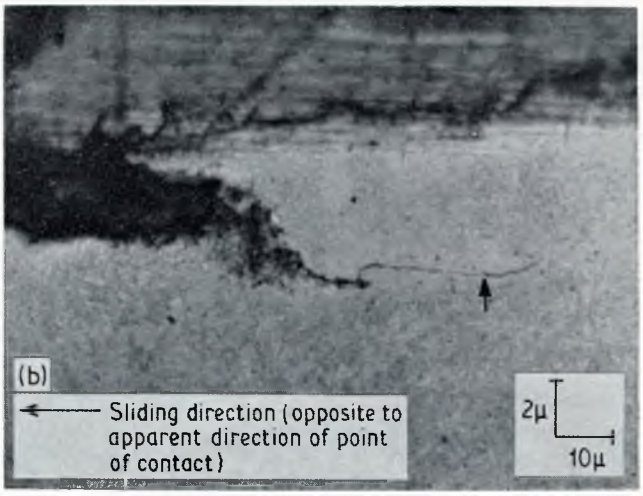


Propagation of crack from edge of shaving mark into matrix

FIG. 14. Initiation and propagation stages of a crack associated with shaving marks on the starboard pinion



Propagation stage of fatigue crack which initiated at region of surface flow across machining mark



Detail of fatigue crack shown in (a)

FIG. 15. Taper sections, showing initial stage in formation of a micro-pit in the dedendum of port pinion adjacent to the pitch line

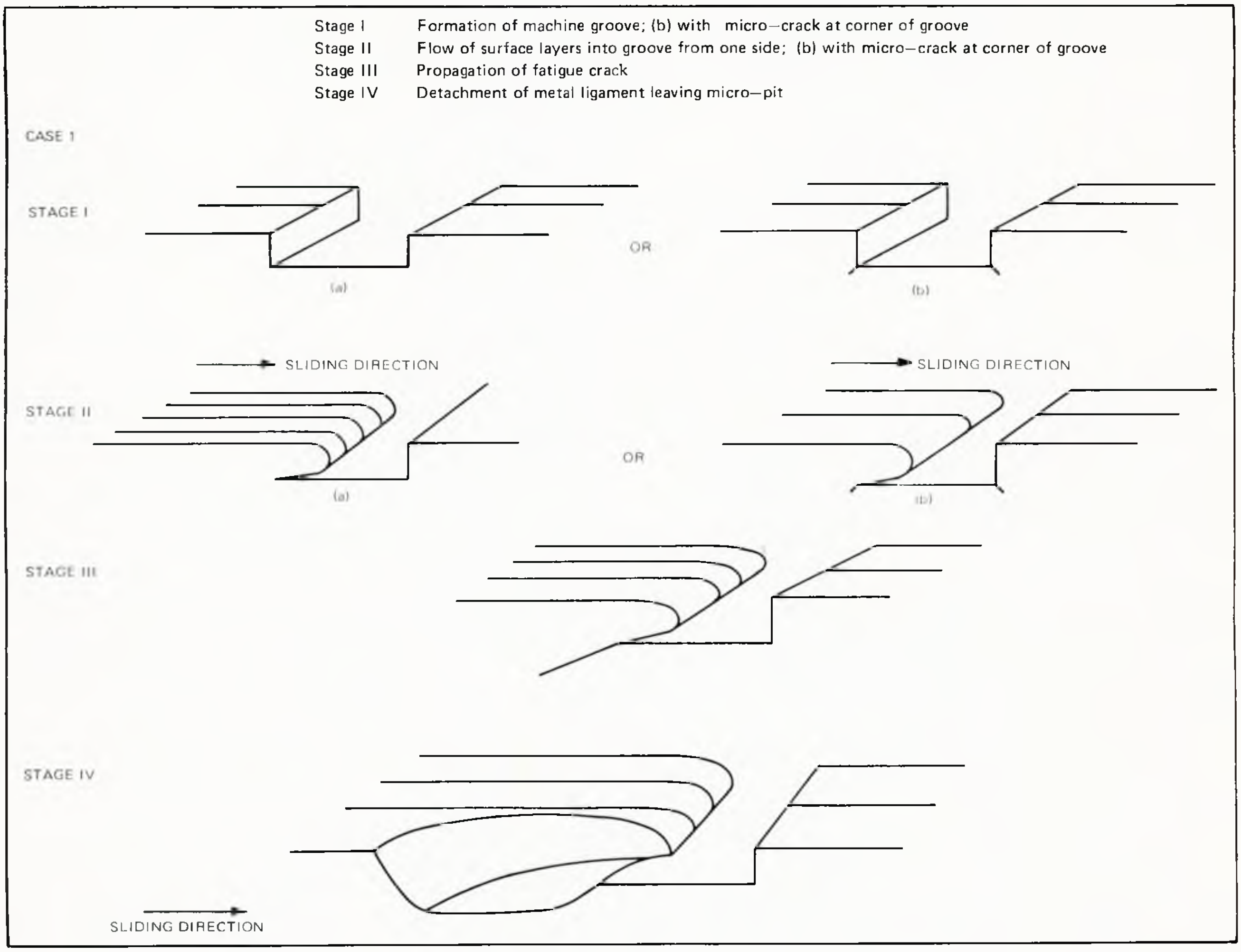


FIG. 16. Schematic representation of sections through various stages of micro-pit formation

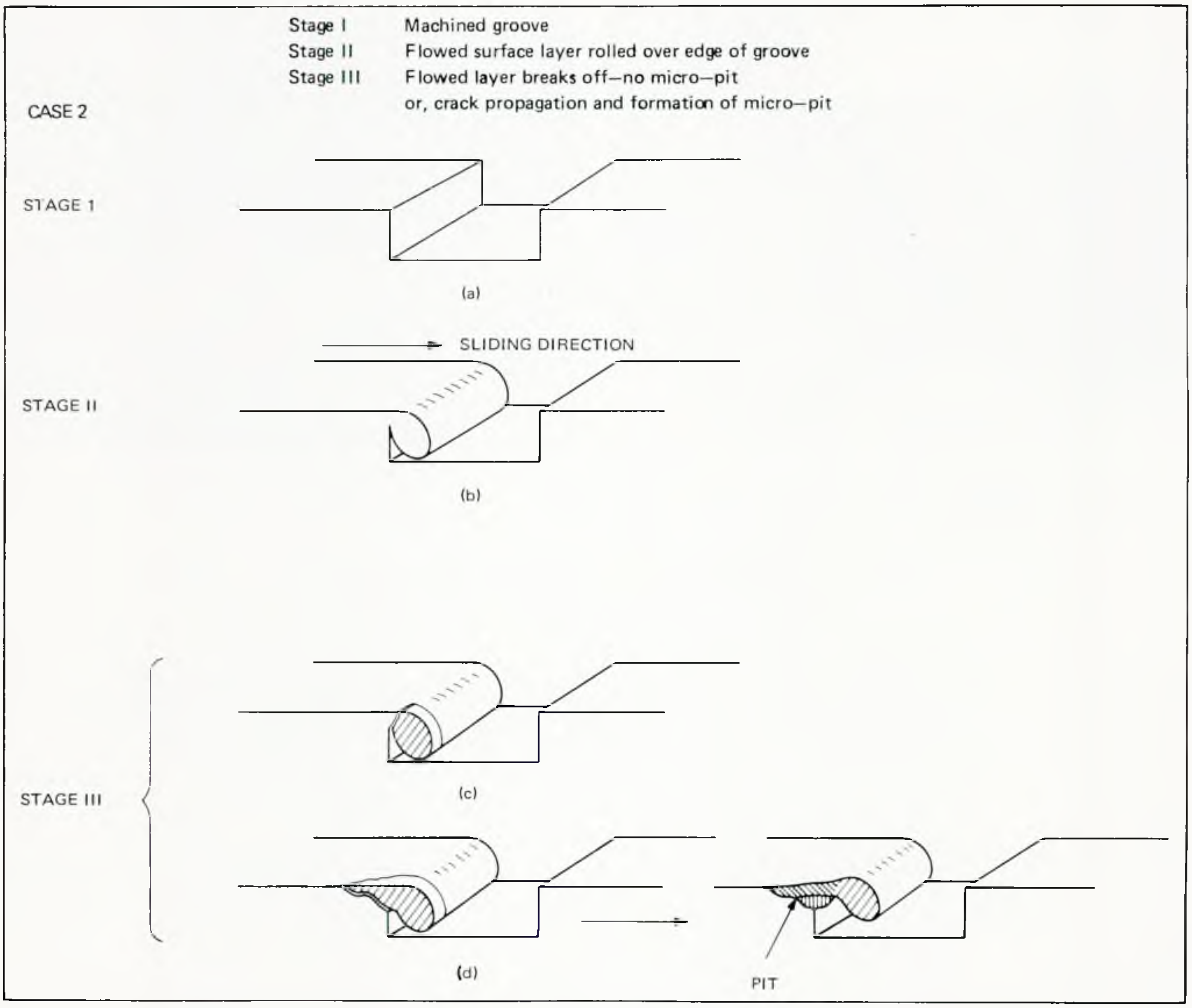


FIG. 17. Schematic representation of sections through various stages of micro-pit formation

The pitting, peening and hammering of both addendum and dedendum would have been brought about by lack of oil protection. At higher than normal operating temperature the oil would be of lower than normal viscosity, thus providing less cushioning. The lubricant restriction caused, not scuffing, but pitting. Another illustration of the same phenomenon was reported in an ASME paper(16).

In the event, the gear was saved and is still operating satisfactorily. The cure was effected by relieving the damaged part of the helix by twisting of the gear case to redistribute the load. The amount of oil supplied was increased by opening up the orifices before the sprayers. Finally, a new charge of full EP oil was used, thereafter topping up with normal double-inhibited turbine oil.

5.4 Design, installation and operation

Scuffing failures in modern gearing are extremely rare but the example in Fig. 21 illustrates graphically what can happen when matters really go awry. The photographs are of the ahead and astern flanks of the final reduction of a gear transmitting 27,000 kW (36,000 shp) at 86 rev/min.

Investigation showed that, for manufacturing reasons, the gear teeth had been cut with a modified addendum/dedendum ratio, resulting in increased sliding speeds. The oil quantities supplied to the gear sprayers, were for various reasons, lower than the design values.

Finally, the gears were subjected to appreciable over-torque at an early stage in commissioning.

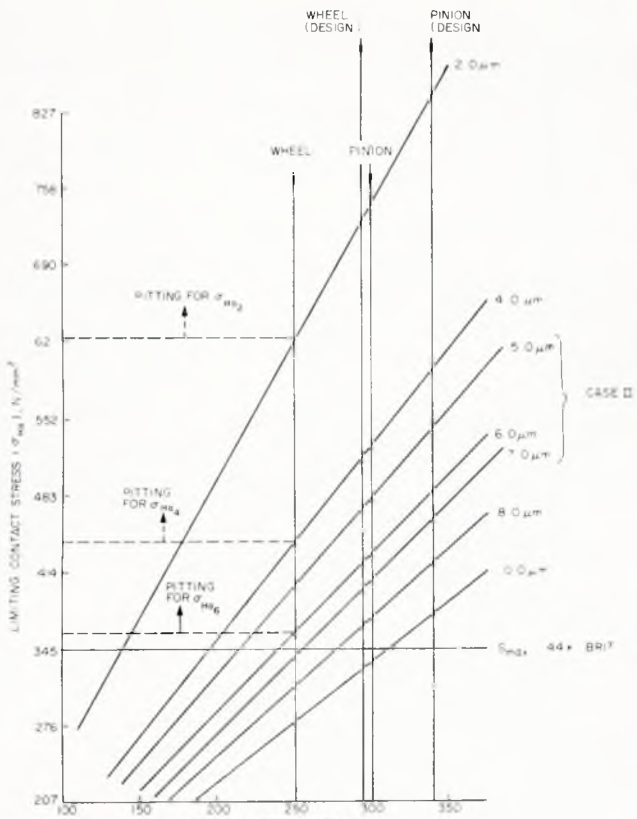
Thus, here was an example where design, installation and operation had all played a part in the failure. This was a compounding of several factors, any one of which by itself would not have been responsible, which caused failure.

6. CONCLUSIONS

Advanced design, manufacturing and installation techniques have resulted in improved performance of gears over the past decade, particularly that of turbine gearing in VLCC. The more learnt about environmental conditions, the closer to its limits a design can be safely operated in the future. However, some caution is needed in two special respects.

For VLCC gearing there has been a general demand for increased power and lower propeller speed. To meet such demand, proven designs have sometimes been extrapolated in size, rather than redesigned. This is of course reasonable but proper regard must be paid to limitations relating to size and there must be sufficient knowledge of the more severe environmental conditions.

Secondly, no matter how excellent is a gear design, it will not prove reliable unless manufactured and installed to high standards of accuracy. This can be achieved only with strict quality control throughout all stages of manufacture to final commissioning. Means of applying quality control still vary among both builders and shipyards. This is considered to be an area where improvement is possible by applying the latest measuring techniques and reducing reliance on shopfloor experience, however valuable.



Limiting Hertzian contact stress, σ_{Ho}, vs BHN for various levels of surface finish (peak-to-valley)

($1\ \mu m$ = 40 micro-inches, 100 N/mm^2 = 6.48 $tonf/in^2$)

FIG. 18. Limiting Hertzian contact stress vs BHN

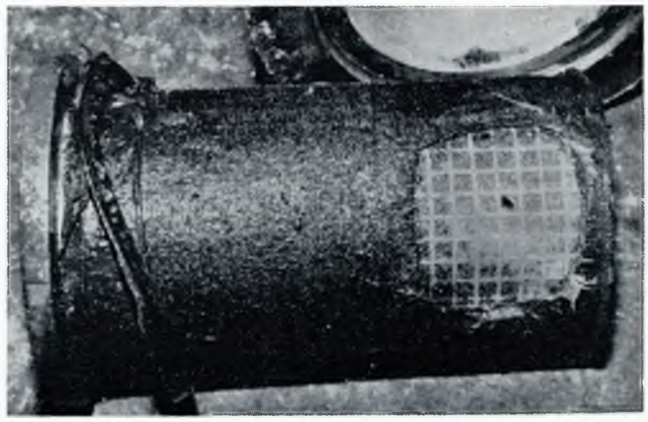
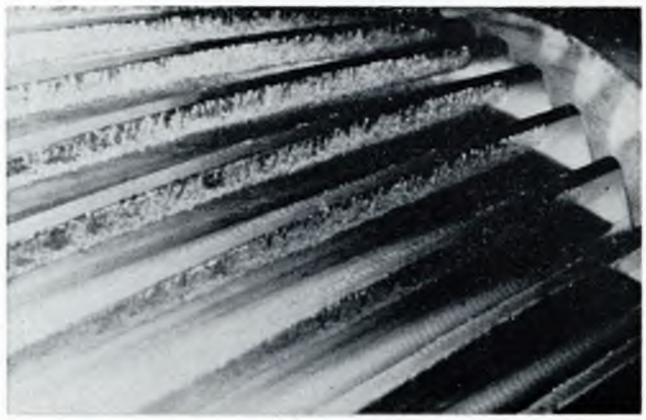


FIG. 20. Contaminated sprayer filter



(a) Ahead flanks

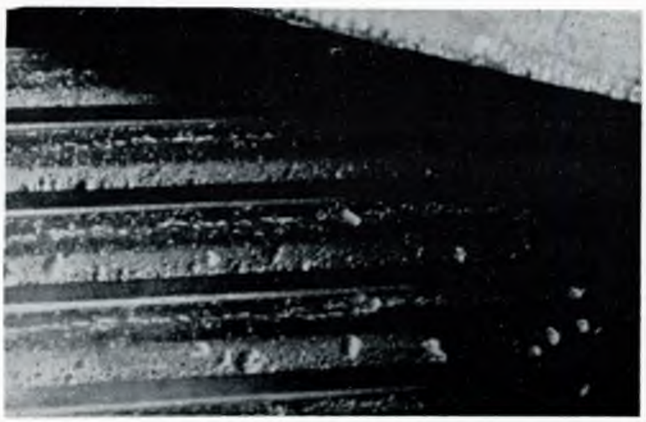


FIG. 19. Pitting of final reduction wheel

(b) Astern flanks

FIG. 21. Scuffing of final reduction wheel

Considerable emphasis has been placed in this paper upon accurate alignment and good surface finish as being basic to reliable operation. Experience has shown this to be necessary in both turbine and diesel gearing. Indeed, in meeting these two requirements a number of uncertainties due to environmental factors are minimized. This is particularly so for shaft alignment variations which are inevitable, irrespective of accuracy. Again, in order to meet these two requisites the authors believe there is a need for a more thorough understanding of surface finish and alignment.

It has been suggested on occasions that very large ship size, stiffer shafting and all-aft machinery can impose problems of shaft alignment. This is perfectly true but, again experience has shown that, if installation is carried out with due regard to thermal and hull effects, there need be no reduction in the reliability of main gearing.

In future the user will have a choice of design philosophies. He is offered by most builders an extension of existing technology, refined to decrease the cost and increase the reliability of conventional plant. In contrast, STAL-LAVAL offer what is described as "the beginning of the road": new technology incorporated in their Very Advanced Propulsion concept. This is now within reach of owners who are prepared to back new technology, matching certain development risks against promises of more economic operation and ultimately higher reliability.

In the context of this discussion VAP is of most interest for the use of epicyclic units in the final reduction-gear, which permits a significantly shorter, narrower main engine layout.

Geared medium-speed diesel installations may well be used more widely for VLCC in the future. This will depend greatly on the operating experience with ships currently in service and on fuel quality over the next decade. In selecting the transmission for m.s. diesel plant in these and other applications, very important factors are the characteristics of engine couplings, the selection of gear materials and their surface treatment.

The authors do not believe that either tanker or container ship operators will in future require higher powers. The demand will be for extreme economy and complete reliability. The latter will be achieved, not merely by good design but by precise execution and application of the product. From that point onwards it is up to the operator to maintain his plant as if it were his private property.

7. ACKNOWLEDGEMENTS

We wish to thank Shell International Marine Ltd. and Shell Research Ltd. for permission to publish this paper, and various colleagues for their help and advice in its preparation; also Rank Taylor and Hobson. The opinions expressed are attributable solely to the authors.

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Discussion

DR SIMON ARCHER, MSc, CEng, FIMarE (Consultant), said that the authors' paper had rendered a signal service to the marine industry. The emphasis on investigational work and the description of the techniques now available were particularly welcome.

Looking back some 25 years to the marine turbine gearing as surveyed in his own paper to the Institute in 1956[1], it would appear that the principal progress in reliability to date had been the virtual elimination of scuffing and tooth fracture as causes of gear replacement, but this still left the "hardy annual" of pitting damage, which fortunately, however, rarely demanded gearing renewals.

On the design side the adoption of higher specific tooth loading had been facilitated by such factors as improved materials, more accurate machining and finishing, better lubrication, greater attention to cleanliness, improved knowledge of the effects of alignment on gear performance, etc.

Of the various factors mentioned as being of fundamental importance in gear rating, he would agree that surface finish and dynamic alignment of meshing elements were probably the most crucial.

On surface finish, as the authors had predicted, the revised BS 1807 would include tolerances for the various quality classes and it was noted that the Shell through-hardened gears, having a specified maximum R_a value of about 0.8 μm (32 μin), would, on the current BS proposals, be representative of a really high quality finish, assuming a minimum wavelength cut-off of 0.8 mm and a sampling length of 2.5 mm taken in any direction. It was to be hoped that other gearing regulatory bodies, such as the Classification Societies, would follow suit before long (BS 1134:1972 was helpful here).

The authors' warning remarks on the possible hazards of the shaving process would awaken echoes in some quarters, and Figs. 12 to 15 in the paper had shown some horrific examples of what could be sparked off by the sort of machining grooves which might be associated with (doubtless) poor quality shaving. As Dr Archer had seen, in some cases, the so-called "improvement" in surface texture achieved by this process, especially on pinions of high tensile strength, he would tend to question the advisability of using this finishing process beyond about 70 t/in^2 (325 Vickers) and, for harder pinion materials would recommend finishing by grinding. This would have the added advantage of enabling a more accurate modification of helical angles to be applied (always a difficulty with shaving) and, usually, also an improvement in tooth profile to be achieved. It was apparent, of course, that with the higher load factors currently being adopted, helix correction was becoming increasingly important. The authors had not touched on the subject of helix correction, and it would be valuable to know of the Shell practice and experience on the ships listed in Table I.

It was noted that nearly 85 per cent of the "L" class ships had gearing designed and/or built by Stal-Laval, whose characteristic philosophy had been to provide sufficient flexibility in the gear box construction to enable compensatory pinion/wheel adjustments to be made in the final reduction, to allow for such factors as hull distortion, temperature effects, etc. It was believed Stal-Laval claimed an enviable record of freedom from pitting in their final reduction gears and it would be interesting to know whether Shell experience had justified that claim.

On the important matter of quality control, rightly emphasized by the authors, operators should be able to safeguard their own interests by insisting that they were supplied with quality control records of all important measurements as a contractual obligation and that they claimed the right of approval, or otherwise, of the results. Alternatively, they could delegate this approval to the Classification surveyor.

Finally, Dr Archer wished to comment on the vexed question of main shaft alignment for rigidly coupled gearing.

Ideally, it would be preferable to isolate the main gears from the thrust and propeller shafting by means of a reliable flexible coupling, as was sometimes practicable with smaller oil engine installations. However, feasibility and cost appeared to have discouraged development in that direction, at least for higher power turbine machinery. Therefore, one had to contend with the consequence of direct coupling.

As the size of main wheel bearing journals had grown, say to 30-in diameter and above, bearing clearances had correspondingly increased (often 0.001 in/in of diameter). This meant that unless the "hot" static load distribution on forward and aft bearings was equal, the resultant dynamic bearing reactions would not only be unequal but, more importantly, they would not be exactly parallel. The consequences of this seemingly small deviation (often no more than 8° to 10°) could be illustrated by a recent case where the following measured and calculated loads applied:

"Hot" static light ship loading	—	11.6 tonnes, forward bearing
		20.5 tonnes, after bearing
Corresponding resultant "hot" dynamic loading	—	47.1 tonnes, forward bearing
		52.3 tonnes, after bearing

Neglecting the influence of oil film action, the calculated effect of that differential loading was to cause a main wheel skew, whereby the forward journal centre was displaced relative to that of the aft journal by 0.0045 in upwards and to starboard at an angle of 55° to the horizontal.

For the fully loaded ship at full power, using proximity gauges, a somewhat smaller deflection of about 0.0035 in was measured at about 65° to the horizontal and the amount of skew increased rapidly with the rev/min.

The important result, however, was the unexpectedly large change of contact (tooth "opening") across the main wheel helices, which, calculated along the lines of action of the four pinion/wheel meshes, amounted to:

LP lower 0.0010 in (HA)	HP lower 0.0010 in (HF)
LP upper 0.0005 in (HA)	HP upper 0.0004 in (HF)

(HA = heavy aft, HF = heavy forward)

These deflections were, of course, comparable in magnitude to any helix corrections for pinion combined bending and torsion which might be applied and must render the task of the gearmaker that much more difficult in his efforts to compensate for that effect.

Therefore, it appeared vital that gearmakers should specify a very much smaller tolerance in differential loading of main wheel bearings than was the current practice. Moreover, in order better to compensate for changes of hull deflection between light ship and fully loaded conditions, any differential loading should be in the sense of heavier forward in the light ship condition, as had been consistently recommended by Volcy[2].

The authors' views on this problem would be much appreciated.

MR I. T. YOUNG, CEng, FIMarE (GEC Marine and Industrial Gears Ltd) stated that it was always refreshing to hear comments on gearing design, manufacture and operation from the standpoint of the operator, and the authors' paper was no exception. They had expressed their views with clarity, and were right to stress the limited choice of geared turbine plant now available. This was regrettable, but also inevitable with the decline in the popularity of steam turbine propulsion. This semi-monopoly position led to certain absurdities in that some designs commonly available were hallowed by continued re-use rather than by the up-to-date logic of their design approach.

The authors were obviously in difficulties when arguing that through-hardened gears were the preferred choice while also admitting that for diesel drive hard-on-soft and hard-on-hard were more appropriate. There was one American gear manufacturer who claimed that the best approach to design was to settle on the largest bull gear that could be fitted into the ship. Most of the gear designs still being produced by the specialist turbine manufacturers seemed to rely on this same approach for the final reduction.

Clearly, where diesel engine centre distance was fixed by considerations of access and engine room space, the shipbuilder would not tolerate inefficient design by the gear manufacturer who was normally independent of the engine builder.

Mr Young was puzzled by the authors' statement in the section on installation: "The best aligned gears are those which require the least adjustment of the gear case in the ship", but in the section on case histories they had paid tribute to the provision on the Stal-Laval AP gear-box for optimizing alignment after installation. He could well understand a distaste for scraping of bearings to effect the improvement and imagined they would have been glad of an adjustment facility on their F class ships. He represented a company that had pioneered adjustable bearings over the last 20 years; and at times had also been glad to have been able to adjust in a civilized manner.

The authors had mentioned surface roughness and waviness as two different aspects of tooth finish, but had proposed a limit for the latter only. Waviness was covered in a relatively coarse manner by profile error limits. His own experience had shown that waviness at a closer spacing than the normal profile errors, but with a wavelength beyond that picked up in a normal surface finish traverse, could have a vital effect on tooth surface endurance. Could the authors comment on that?

Chromium had been mentioned as a valuable alloying element where scuffing was to be avoided. Mr Young would only agree if the pinion were nitrided. Some authorities indeed would look on chromium in through-hardened pinion and wheel as equally undesirable to nickel. Was the authors' view backed up by disc tests? The suggestion that carburised gears must always be deep-frozen to avoid scuffing was new to him. His Company produced many carburised gears, and this extra sub-zero process was seldom used.

The authors' use of EP oils as a healing medium seemed to extend to pitted gears. They were, of course, the oil experts, but he would like an explanation of the action of the oil in such a case. Was it really beneficial, or just a case of overkill?

Finally, a short comment on ISO loading calculations. Classification Societies were interpreting the new loading rules in their own differing ways, to the great discouragement of independent thought. Recently he had had occasion to compare a number of different designs on the latest ISO basis and had plotted the results on completion against Lloyd's K value. The results in terms of safety factor (actual over allowable stress) gave such good correlation that he wondered if the old simple K factor was such a bad idea after all (see Fig. D1).

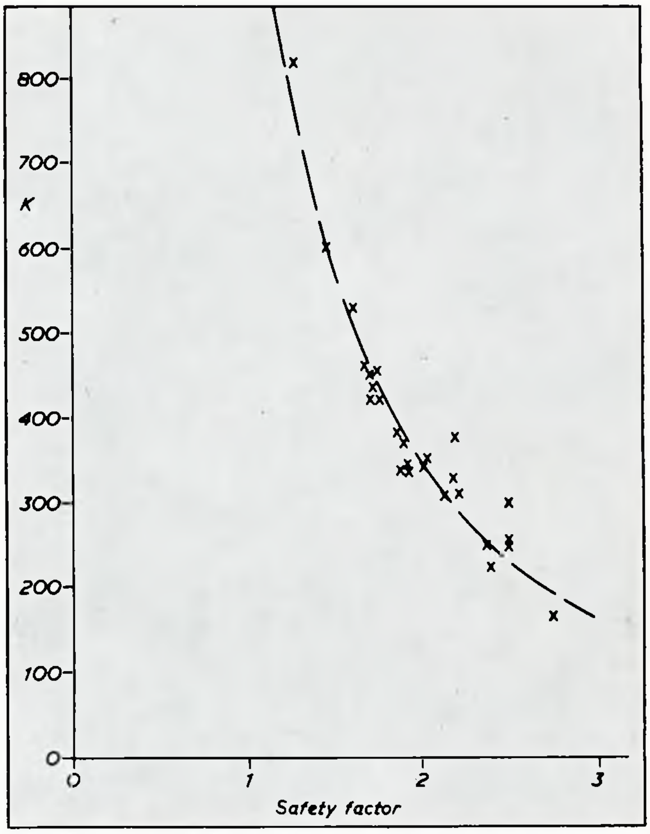


FIG. D1 ISO loading calculations against Lloyd's K value

1) ARCHER, Dr S., 1956, "Some Teething Troubles in Post-War Reduction Gears", *Trans I Mar E, Vol* 68, *pp* 309-352.
2) VOLCY, G. C., 1975, "Reduction Gear Damages Related to External Influences", *Marine Technology, Vol* 12, *No* 4, *October*, pp 363 and 365, especially.

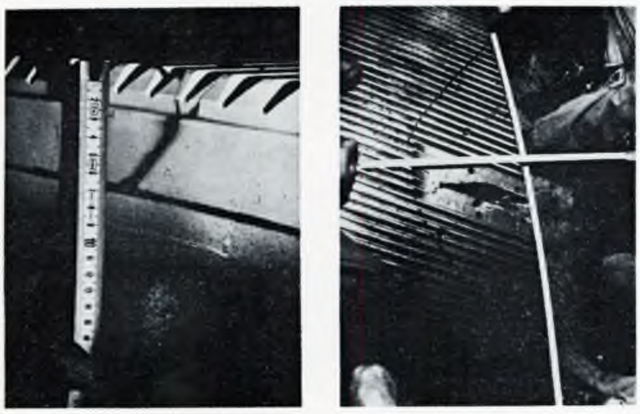


FIG. D2 Cracked teeth and rim of bull gear wheel
shp = 19500 n = 105 rev/min

MR G. C. VOLCY, MSc, CEng (Bureau Veritas) expressed his thanks to the authors for their invitation to contribute, which he did with great pleasure as he had had the privilege of collaborating with Mr Saunders-Davies for many years. Once again, the latter and his colleagues had produced an outstanding paper, backed by the results of interesting research.

They were right in saying that, notwithstanding many papers on the subject, there was little feedback of operator experience and that they were filling up this gap.

He would like to add some examples of such experience, collected during his work in a Bureau Veritas trouble-shooting team, which mainly endorsed the authors' statements; and also to show some "horror pictures".

He agreed with the authors about the important role played by mounting and correct alignment (static and dynamic) of line shafting in the trouble-free operation of marine gearing, which was hobbed and shaved with micron precision and then exposed to external distorting influences of the order of several millimetres (see Ref. 9 of the paper).

Mr Saunders-Davies might recall the case of five jumboized tankers of about 60,000 dwt, all affected by severe pitting even before jumboizing. This reached such an extent that on one of them the bull gear assembly had to be replaced. Thereafter correct behaviour had been ensured by rational re-alignment of line shafting and elimination of the aft bearing on the thrust shaft which had been incorporated into the thrust casing. However, on the other four tankers rational re-alignment, and elimination of contact between the aft bearing on the thrust block and the corresponding journal on the shaft, were sufficient to stop the pitting.

It was worth remembering that in the past (in old-fashioned designs with either integral or independent thrust bearings) journal bearings aiming to support the shaft had been incorporated in the thrust block. In the early sixties, in addition to the increasing flexibility of the double bottom steelwork, it had been one of the most important causes of catastrophic failures, (see Figs. D2 and D3).

In the first case, after replacement of the bull gear, re-alignment and elimination of contact between the journal and bearing, struts had been installed to reduce the excessive flexibility of the thrust bearing and its foundation.

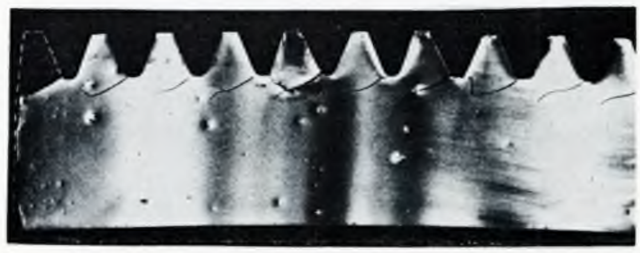
In the second case, with the stiffer foundation of an independent thrust block, the solution had consisted of re-alignment and elimination of the contact between the thrust shaft journal and bearing.

Fig. D4 showed pitting, spalling and heavy wear of the after helix due to analogous causes, but this time with an independent thrust bearing. The remedy was as above. Afterwards, in all cases, the gears were behaving to the full satisfaction of the owner, better than before the changes; in the last case they had not even been shaved[1].

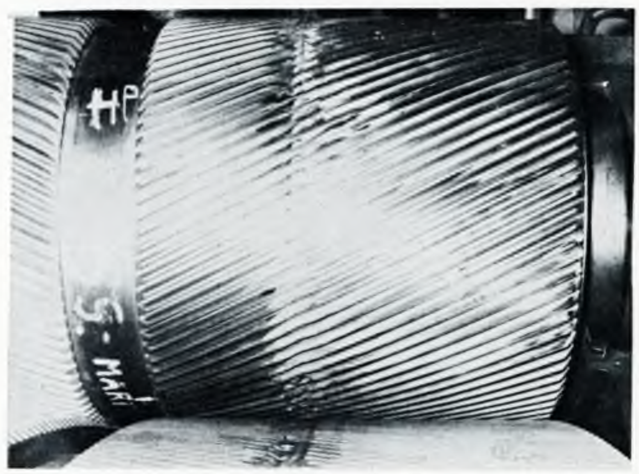
The introduction of thrust bearings with tilting pad supports, without journal bearings and with the casing being attached to

1) VOLCY, G. C., 1968, "Actual behaviour of the marine gear and line shafting conditions of alignment", *Marine Technical News*.

a. Cracked rim of bull-gear wheel.



b. Broken teeth of bull-gear wheel.



c. Damages to teeth of H.P. pinion.

FIG. D3 Damage to main gearing 2nd reduction
shp = 2100 n = 110 rev/min

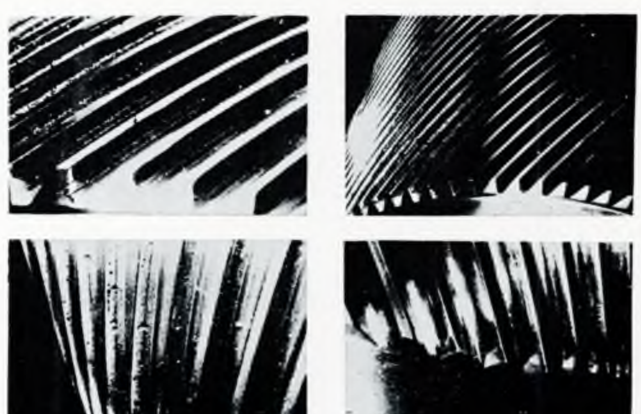


FIG. D4 Pitting, spalling and heavy wear of after helix
shp = 17600 n = 108 rev/min

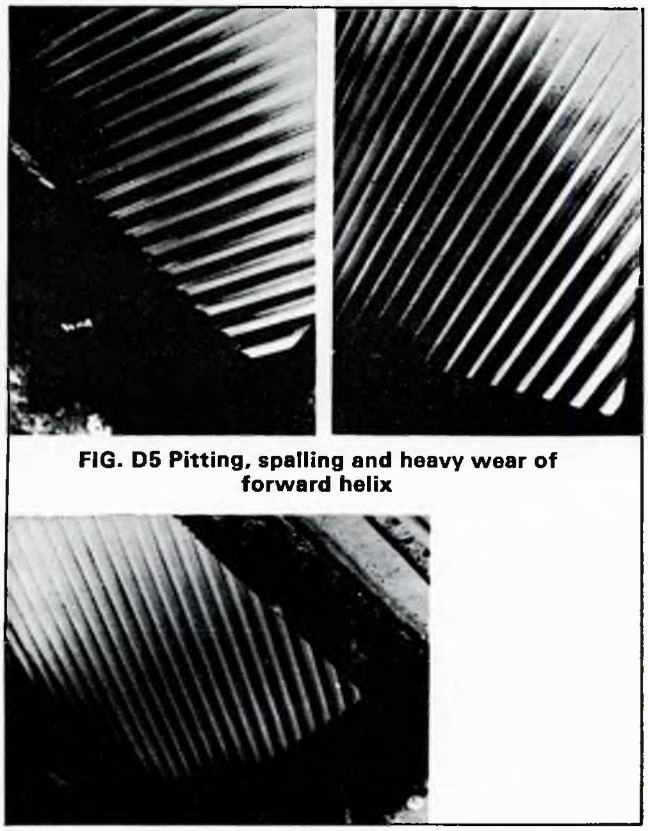


FIG. D5 Pitting, spalling and heavy wear of forward helix

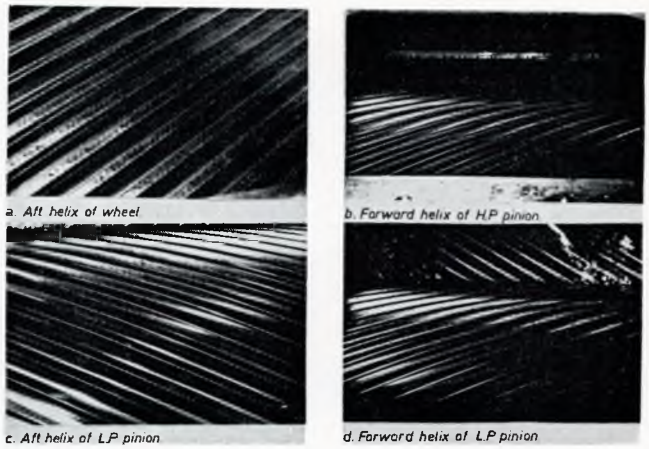


FIG. D6 Pitting, spalling and undercutting of final reduction teeth
shp = 22000 n = 82.5 rev/min

the foundation steelwork on the level of the shaft axis, had solved the inconveniences of old-fashioned thrust bearings.

However, the period of the tanker bonanza had seen new troubles caused by the steelwork of the hull girders and engine room double bottoms.

These phenomena being insufficiently understood at the time, damage by pitting had occurred on the forward parts of forward helices (see Fig. D5).

The remedy had been to decrease the flexibility of the inde-

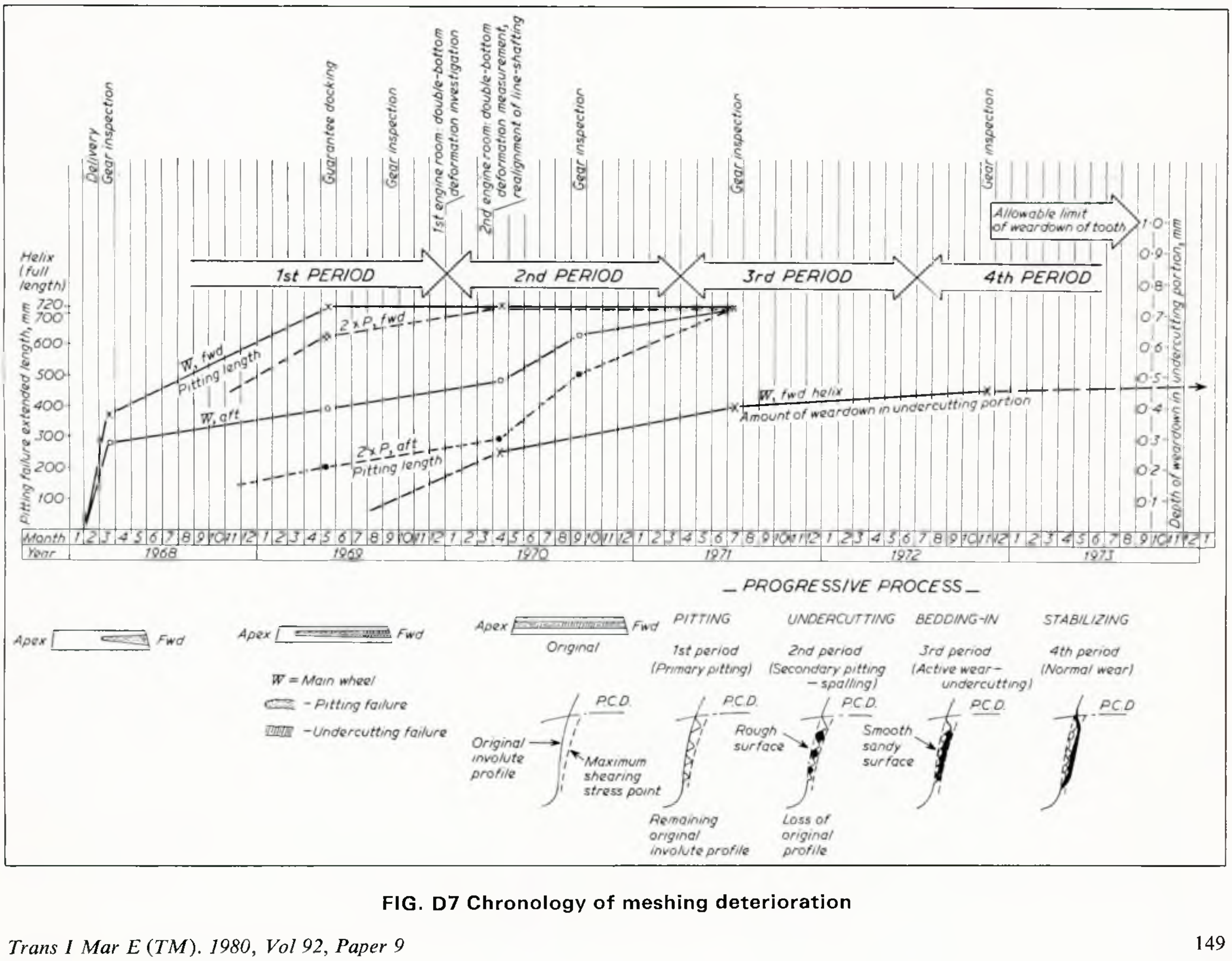


FIG. D7 Chronology of meshing deterioration

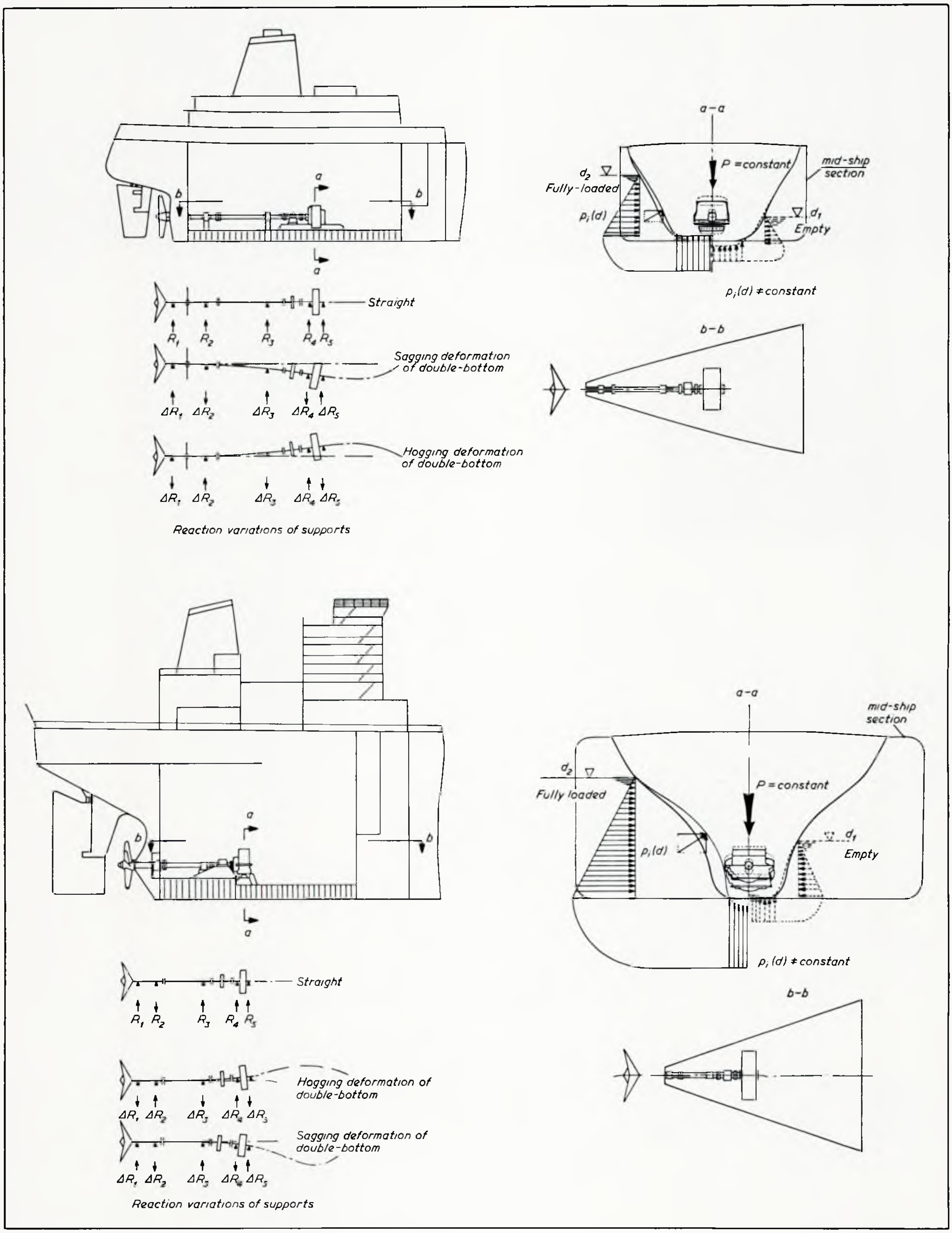


FIG. D8 Hogging and sagging of double bottom deformation

pendent thrust bearing foundation and, especially, rational re-alignment, taking into account the hogging deformation of the aft part of the hull girder and its double bottom(2). After only three to four weeks of operation the progress of pitting had stopped and the gear had behaved correctly.

Another interesting and analogous case concerned two tankers of about 120,000 dwt where pitting had been recorded on the forward part of forward helices of the bull gear (Fig. D6).

Fig. D7 showed the chronology of events from which the rapid deterioration of meshing could be noted.

In the first ship the remedy had consisted in the internal re-alignment of the bull gear and the partial re-alignment of line shafting. In the second ship the line shafting only had been rationally re-aligned.

These countermeasures also required modification of the aft bush of the stern tube by additional slope boring of the whitemetal, as well as slope boring of the stern tube itself. This made it possible to counteract a deformation of about 3.5 mm of the double bottom steelwork.

About 18 years ago, during his researches concerning troubles affecting crankshafts and bearings of diesel engines, Mr Volcy had discovered unexpected hogging of the aft part of the hull girder in fully loaded ships while the rest of that girder was sagging. The converse of this strange phenomenon occurred when going from full load to ballast or empty conditions. He had confirmed these effects by calculations, the results of which had been published in 1967(3) and similar results had been found by others, especially in Japan(4).

During changes in loading conditions, there were also other, local, deformations, superimposed on the general deformations of hull girders. These were due to deformations of double bottom and outside shell. His latest experimental and theoretical work on VLCCs and even ULCCs showed that these two types of deformation were interdependent.

From the point of view of gearing and line shafting, what counted were the double bottom deformations. There was hogging during loading and sagging during discharging but alas, only in conventional types of VLCC, say up to 200,000 dwt.

On the contrary, for bigger ships, especially ULCCs, the importance of outside shell deformations and their effects in the engine room were such that the resultant double bottom deformation was sagging during loading and hogging when discharging. This phenomenon and its explanation were shown on Fig. D8 and more details are available(5).

The above emphasized the complexity of deformation phenomena due to thrust bearing, tilting, hull girder, double bottom, and outside shell, and even the presence of the pillars supporting the superstructures or engine casing!

It would be imprudent to generalize, but he agreed with the authors that a smooth curve of hull girder deformation was usually less important than the local deformation of double bottom steelwork. These different types of local deformation, together with the effects of thermal expansion of steelwork, were the main factors in the correct static, quasi-static and dynamic alignment of gearing and line shafting.

For these reasons he did not favour rigid connection of gear casing with double bottom steelwork, but was in favour of a point support, such as proposed by GEC in the UK, and by Stal-Laval.

Freedom (assured by correct clearances) and flexibility of the line shafting were basic concepts of marine engineering. The exception was the tailshaft which should be stiff in order to move bodily with the flexibility of the steelwork(6).

It was important to understand each problem, then try to create favourable conditions which left the machinery free to compensate for human lack of knowledge by adapting itself to its bearings, which rested on the flexible steelwork.

The new design of Stal-Laval's VAP Installation, well endorsed by the authors, was in line with this consideration and he entirely shared their opinion.

2) VOLCY, G. C., 1969, "Damage to main gearing related to shafting alignment", *IMAS*.
3) VOLCY, G. C., 1967, "Forced vibrations of the hull and rational alignment of line shafting", *Marine Technical News*.
4) MANO, M. and NAKASHIMA, S., 1972, "The deflection of engine room double bottom structures", *Shipping World and Shipbuilder*.
5) VOLCY, G., GARNIER, H. and MASSON, J. C., 1973, "Deformability of hull girder steel-work and deformations of engine room of big tankers", Bureau Veritas Technical Bulletin.
6) N.I. 138 A RD. 3 (Bureau Veritas Guidance Note), "Recommendations designed to limit the effects of vibrations on board ships", June 1979.

MR R. C. BRYANT (General Electric Co. USA), considered that the authors' very timely paper had added a much needed perspective to the subject.

The need for emphasis on quality control during manufacture and on attention to proper installation procedures was certainly confirmed by his own Company's experience. In some areas, such as flushing a new gear unit after installation, it was necessary to go beyond merely defining the procedure, and actually to verify that the objectives had been accomplished, and that no part of the procedure had been overlooked or bypassed. They were pleased to learn that no operating problems had been encountered as a result of the introduction of nitrided gearing to the Shell fleet, and hoped that in future other owners and operators would be encouraged to accept surface-hardened gearing with its attendant advantages of reduced size and/or potential higher margins of safety with respect to tooth surface deterioration.

The authors had correctly pointed out that hull and foundation deflections during operation could introduce gear casing distortions of the same order of magnitude as the internal torsion and bending deflections caused by forces within the gearing. However, this did not mean that corrections should not be made for those deflections which could be predicted. It was possible to calculate quite accurately the internal distortions and to compensate for them, thereby, reducing the effect of the external influences.

MR J. P. KERPESTEIN, CEng, FIMarE (Royal Schelde) said that the authors had quite rightly emphasized the importance of quality for prolonged trouble-free operation of propulsion gearing. That aspect did not only apply to the design and manufacture of gearing and to internal gearing alignment, but also to the setting of the gear unit in the ship and the exercise of the entire propulsion system alignment, the shipyard being strongly involved in the latter two.

Some gearing manufacturers were able to supply the quality which the authors would like to see. In addition, companies who were also involved in designing and manufacturing propulsion gears for naval ships (including his own Company) were able to give their customers control over the quality they would get. Such manufacturers generally complied with NATO requirements concerning an effective quality control programme which applied to organization, procedures, and disciplines, and led to a product which satisfied the requirements of the contract. Once such a programme was implemented it would normally be effective for non-military products also. It might, therefore, be concluded that shipyards and/or owners could get the quality they wished for and that from some gearing manufacturers they could even obtain documented assurance that control existed over the whole process, from early design stage to delivery.

However, shipyards and/or owners seemed to buy on price, not on quality, particularly for medium speed diesel engine gearing. In recent years there had been a number of examples of breakdowns consequent upon this policy; for the shipbuilders and owners concerned the bitterness of that poor quality must have remained long after the sweetness of the low price was forgotten.

However, even when the gearing manufacturer had supplied a quality product he was to some extent at the mercy of the shipbuilder. The latter must coordinate all the setting and alignment data and instructions received from the prime mover and gearing manufacturer and transform these, together with line shafting alignment analysis data, into one integrated system setting and alignment analysis, and associated procedures.

Furthermore, the shipbuilder must provide his part of the gearing lubricating oil system with rather rigid requirements on cleanliness. Shipyard activities had a direct influence on prolonged reliable gearing operation and, therefore, the quality aspect was equally important in the Yards.

When all the above had been done properly one should have a reliable propulsion system.

Health monitoring systems for gearing could be, and had been, devised from the usual simple types which mainly monitored bearing temperatures to much more sophisticated types, e.g. those which were capable of monitoring such a vital condition as load distribution across the gear face width. However, the best they could provide was an early warning of an undesirable situation which already existed.

It would be interesting to learn from the authors which operating parameters they would like to see monitored in turbine gearing and in medium speed diesel engine gearing, so that developments, if needed, could possibly be directed in the area indicated.

COMMANDER M. D. COOPER, CEng, FIMarE, RN (Ship Department, MOD (PE)) wished to supplement the authors' conclusions from the point of view of a warship operator.

First, for knowledge of what constituted severe environmental conditions, warship conditions could be second to none. One could only appreciate the skill of a frigate's transmission designer after surviving the rigours of a fishery protection patrol in the North Atlantic.

Second, he endorsed the authors' point of view on installation and commissioning and would add that, in his view, it was the designer's responsibility to specify the standards which needed to be achieved during this process. These must be both practicable and effectively demonstrated to his satisfaction.

He would suggest that the "new" technology to solve the authors' problems existed and had been well proved by naval gear manufacturers. Far from being "at the beginning of the road" many were a good way down it.

The illustrations of failures presented in the paper were reminiscent of the failures the Navy had had to cope with some thirty years previously. Such problems were solved by the investment of considerable sums of taxpayers' money in research and development. The results of that work were shared by the participating members of what was most recently known as the Navy and Vickers Gear Research Association, commonly referred to by the ugly acronym NAVGRA. Although only one company was named in the title, several major British manufacturers were members. The activities of the Association had been published before this Institute and other learned Institutions.

Over the years, naval gear development had progressed on a broad front and had now reached a position where main reduction gears rated at K factors approaching 1000 could be considered for service. The results of that development had been endorsed by millions of horsepower hours of operational experience. The expert knowledge was available to the participating members of the former NAVGRA organization. They were free to use it in any of their designs, be they for a naval or mercantile operator.

There must be much in common between the requirements for gear-boxes for both warships and merchant ships. Reliability must be at the top of both lists although the subsequent order of attributes might vary. What was likely to differ, however, was the price one was prepared to pay to achieve the standard required. Sadly, in this material world, results had to be bought but he would commend the thought that a NAVGRA naval gear manufacturer could also offer much that had already been paid for by someone else.

MR F. S. LYNAM, CEng, FIMarE (Esso International) said that the authors' comprehensive paper had touched upon practically every subject worthy of comment in relation to the more recent experience of tanker propulsion gearing. Additionally, their findings on the development of pits from manufacturing blemishes in surface finish, which were quite normal on many marine propulsion gears, was an excellent example of the value of feedback of service experience to which they had referred in their introduction.

As they had commented, the various National and Classification Societies' Standards could be improved. However, any potential improvements in gear manufacture must inevitably have the commitment of the gear manufacturers behind them to be successful. The embodiment of improved standards in future propulsion gears would require great care in manufacturing—which had apparently not always been achieved in the past. The cost of a gear made to the highest standards was also high unless its toothed area could be reduced significantly by using, for example, surface-hardened teeth or multiple torque-paths to transmit the loads.

In his opinion, the highest gear reliability was only achieved in a new area of application by the correct blend of good manufacturing standards and design features, including innovations when necessary. The first VLCC applications of geared turbines were just such a case where around 30,000 shp had to be transmitted to a very slow-running propeller, relative to previous practice. To maintain the tooth load/inch, K values, and pinion proportions then current in merchant marine practice, would have required a triple reduction gear-box of the parallel-shaft type. The gear designers' solutions offered and ultimately embodied could be placed in three groups as follows:

1) Double reduction gear, having divided torque paths in the primary reductions and consequently four input pinions to the main wheel, together with some increase upon the K values then current for through-hardened gears.
2) Same layout but having hardened pinions and the even higher K values so permitted.
3) Double and triple-reduction trains with epicyclic gears having appropriately high K values in the hard-on-hard meshes, and with two input pinions to the main wheel operating at the "through-hardened" K values then current.

His own experience was that the second solution was the only one which had produced no out-of-service time, though he believed this was not true of all cases in service. Considering gear component parts separately, the final reduction of the third alternative might also be placed in the same category. This experience related to the years when these gears worked continuously at full load and not to their easy life of the past few years. Environmental disadvantages were basically the same for all of them.

The authors had graphically illustrated the importance of the anti-corrosion additive in geared-turbines' lubricating oil. However, much of his own experience suggested that tooth surface fissures and fatigue cracks were not so well protected as the exposed surface. If this could be proven, then any fatigue damage of the surface became a potential problem for turbine gears, and surface finish should receive proportionately more attention.

The authors' comments on alignment requirements were indeed relevant. However, the changes in alignment that related to the stiff output shaft connections, although usually tolerable, remained an unsatisfactory feature. There was considerable justification for putting a flexible spacer-coupling in the output shaft of future gears of this type, and technically acceptable solutions were available. This would eliminate one source of misalignment.

The other sources of misalignment were internal to the gear-box and could be divided into elastic and thermal deflections, as mentioned by the authors. Elastic deflections could be checked by analysis and test before the design was put into service. Thermal deflections were not so easy to predict but they could be minimized by appropriate care at the design stage in distribution of oil flows and by suitable methods of construction for gear elements. For example, the almost universal trend towards gear wheels of closed construction with internal diaphragms and relatively thin rims was directionally wrong if temperature variations were to be minimized.

Finally, the capability of a turbine to overload its gearing when hull resistance increased should be mentioned. On existing vessels, the value of an accurate torsionmeter should not be underestimated. On future units, this torque increase should be allowed for in the design rating of steam turbine propulsion units if they were to be considered as having effectively constant available power.

MR J. J. DUNFORD, CEng, FIMarE (Ship Department, MOD(PE)) said that Mr Richards had recommended visual inspection of the gear teeth at six-monthly intervals. What inspection frequency did he recommend for journal bearings and for any fine tooth or flexible couplings associated with those gear units, if indeed they were inspected at all? Could he also please indicate whether any instrumentation was used to monitor the condition of the tanker gear-boxes and, if so, what criteria were used to assess the need for inspection or repair action. From his own recent experience he would also like to confirm the value of modern tooth surface impression materials, both to provide a permanent record of gear damage and to assist in the laboratory investigation of its causes.

MR H. MACPHERSON (Y-ARD Ltd) thought the paper was interesting and informative and he hoped that other shipping companies would follow the authors' example by giving a feedback of their sea experience.

The statement that the best aligned gears were those requiring the least adjustment of the gear case in the ship would indicate that the construction of the gear cases referred to must have been very stiff, and therefore strong enough to maintain internal gear alignment against outside influences such as setting up and ship movements. The gear-box and main thrust block seatings also required to be of substantial proportions to resist the torque reaction in the gearing, hull deflections and the thrust from the propeller, because gear-box alignment could be upset by seating deflection due to lack of stiffness.

Another factor affecting gear alignment was the influence of the main shafting. His Company undertook shaft alignment calculations for a number of clients and still found a tendency to have too many bearings in the shaft line, making it a very stiff assembly and thus allowing ship deflections to have a big effect on gear alignment. Did the authors come across this tendency when checking out the shaft alignment predictions of the various shipyards building their ships?

Quality control was of prime importance during construction of a set of gearing. It would be interesting to know how the authors' company ensured that the necessary quality was built into their gear-boxes. Did they use the shipbuilders quality control organization or did they have their own quality control inspectors?

Another area now receiving more attention was condition monitoring of machinery to avoid opening up for inspection. Many ships now had data logging facilities and use was being made of this equipment to monitor propulsion machinery. Had the authors' company ever considered monitoring gear-box health and, if so, what parameters would they consider measuring to ensure an accurate indication of conditions inside the gear-box, and thus increasing the inspection period to 12 or 18 months.

MR B. A. SHOTTER (Westland Helicopters) stated that, for many years, the pitting of through-hardened gears had been the factor limiting their load capacity. The mechanism of pit production was normally a fatigue process associated with the contact stress conditions. The rolling action of the contact area moving over the surface could account for some of the crack development by hydraulic pressure. However, with involute profiles the relative influence of rolling and sliding were difficult to separate as they occurred in virtually the same direction. Observation of pit development of gear teeth having conformal profiles, where rolling occurred along the tooth length and sliding up and down the tooth height, showed that it was the sliding component that initiated the pit[1]. The association of pitting with surface asperities and with the area of maximum EHL traction forces just below the pitch line suggested that not only the surface normal force was important, although this was the only force considered in designing the teeth. If the surface shear forces played a significant part in producing pits, there would appear to be more possibilities of affecting the local conditions responsible for the fatigue crack initiation.

The apparent coefficient of friction with rough surfaces was higher than with smooth[2]. Some lubricants had been developed to give high traction (friction) characteristics for variable speed transmissions. Thus both the lubricant and the surface finish could influence the shear forces present. The character of the surface asperities was also likely to be significant. Abrupt changes of slope were more likely to produce higher stress gradients than a more undulating surface and in consequence would be more likely to fail by a fatigue process. It thus appeared that there was hope that future research and development could make a significant contribution to the improvements in reliability which were being sought.

Regarding the authors comments on scuffing, he would like to sound a word of warning. On some aircraft gearing, occasional scuffing had been a problem and an antiscuffing treatment had been introduced to try to minimize the problem. However, scuffing recurred periodically and this was thought to be associated with slight fluctuations in the oil properties. Following one recurrence a more detailed study of the failure was started and several interesting points came to light; one was the fact that the failures were edge initiated. It was not the general surface conditions which were relevant, only those at the discontinuity where the tooth contact line came to a tooth tip. Modifying the edge geometry led to a significant improvement. A second factor of major importance was that at this edge discontinuity, the "anti-scuffing treatment" rendered failure more likely rather than less likely. Thus, if one found that a laboratory test showed improved scuffing resistance, it was not certain that it would necessarily work for all cases of scuffing. In general, when those aircraft gears scuffed, the whole tooth area was affected. It was only careful observation that revealed where the damage was initiated.

The surface blemishes shown in the authors' Fig. 6 were very interesting. The similarity of the markings would certainly seem more than coincidence. He had seen some tooth surface features which had occurred during the manufacture of the teeth due to metal fragments having been rubbed on the finished tooth surface. In the shaving system there were several processes taking place at the tool interface, of which the cutting action was only one. The other most important action was burnishing. If swarf from the cutting action got trapped under the hands of the shaving cutter it could get rolled into the tooth flank and burnished over. Subsequent operation of the gear could relatively quickly result in the particle becoming detached. Such a hypothesis was formulated after looking at Fig. 6. Reading on further, Mr Shotter had come to Fig. 14a, a section through a shaved surface. Just above and to the left of the scale box one could see an area which could be a fragment of just such a form. The scale here was distorted due to the taper section, but scales became even more distorted when a small object was run over by a steam roller! A further investigation along the lines of this hypothesis might show some interesting results.

1) WELLS, C. F. and SHOTTER, B. A., 1962, "Development of Cirar Gearing", *A.E.I. Engineering, Vol* 2, *No* 2, *March/April*.
2) SHOTTER, B. A., 1959, "Surface Finish and Disc Machining Testing of Gear Materials", *B.G.M.A. paper presented* 5 *May*.

DR J. F. SHANNON wrote that the authors had chosen two distinctly different well-known designs for their tanker fleet, one with epicyclic primary and parallel shaft secondary gears, mounted in a flexible symmetrical final reduction gear-box, flexibly mounted to provide self adjustment to shafting misalignment; the other, a DTA design, with a stiff combined box, fully chocked to the foundation. In the former the thrust block was placed aft and free from the gear-box, while in the latter it was incorporated at the forward end of the main wheel shaft and was accordingly much smaller.

All the gears were through-hardened except the nitrided sun and planet pinions of the epicyclic gears. Would the authors say how these designs had performed and responded to shafting misalignment under different ballast conditions.

He could vouch for a recent KHI gear of approximately 50,000 shp at 80 rev/min of similar DTA design which was also fully chocked. In this superb set of machinery the shaft alignment to give equal loads on the bearings of the main wheel was easily controlled by adjustment to the plummer block aft of the main wheel, to cover the range of ballast and loaded conditions.

There were also British designs of the DTA type with combined boxes which were flexibly mounted and had really flexible couplings on the quill shafts instead of the relatively inflexible gear tooth coupling.

With these advanced developments, it would be advisable to have continuous monitoring of the bending moment at the aft end of the main wheel shaft to indicate the load balance on the bearings of the main wheel shaft, so that optimum conditions could be maintained.

It was surprising that none of the builders of the "L" class ships had taken advantage of the "hard-on-soft" combination of materials especially as it gave an added safeguard. It might be that the benefits were not generally understood. Pitting and wear occurred on the dedenda of the pinion and wheel and this had been explored by disc testing. Referring to Fig. 11 of the paper, the slower disc of the pair operated in the condition similar to that occurring on the dedendum of a gear tooth and was prone to pitting, whereas the faster disc was subject to the conditions occurring on the addendum.

When a surface-hardened disc and a through-hardened disc were used (hard-on-soft) and both had a good surface finish, especially the hard disc, two important points could be observed:

1) When the soft disc ran slower than the hard disc, as in the dedendum condition, the pitting limit of the soft disc was

about 66 per cent of its ultimate Hertzian stress and was not influenced by the hardness of the fast disc.

2) But when the soft disc ran faster than the hard disc, whose pitting limit was very high, the limit of pitting of the soft disc was raised to its ultimate Hertzian pressure which, as shown by Chesters[1, 2], was approximately equal (numerically) to the tensile strength of the material.

When this was applied to a "hard" pinion and "soft" wheel, the dedenda of the soft teeth tended to wear but had to do so in a conjugate manner with the hard pinion, which did not wear, so that the dedenda of the soft teeth polish and work harden, thus raising the load capacity to the full potential of the soft wheel. This had been fully explored on the test bed and in service. It was imperative that the hard pinion should have a good surface finish as above, otherwise a rough surface would shave the soft teeth.

A further benefit was that with the higher K loading permitted, a reduction in helix face width could be made which improved the capacity for misalignment over the alternative wide face width of the soft-on-soft gear, as shown by Young[3].

It would be of value to have measurements of the surface finish and tooth profile of the gears in Table I of the paper, before and after service, to see the likely improvement in surface finish and the wear steps, if any. These wear steps did not detract in any way from the performance of the gears. Such information would be useful to the revisers of BS 1807.

MR P. W. MURRELL (Vickers Shipbuilding Group Limited) made the following comments and observations on the paper:

It was disturbing as a gearing designer and supplier to hear the view expressed by an operator that his specifications were necessarily brief, and that it was not possible greatly to influence basic gear design. Whilst it was true that the user, for his own best interest, should acquaint himself with the design of the plant he would be operating, it was also in his best interest to ensure that the designer fully appreciated the requirements of the user, and indeed the preferences and fears the user might have from his previous experience. One way of ensuring that was through the preparation of a detailed statement of requirements, which any gear designer would accept and appreciate. Joint discussions on the formulation of the gearing design to meet those requirements could provide confidence to the user at an early stage, and would give the designer the opportunity of optimizing the design to the user's requirements. Such a policy had been adopted for many years between MOD(N) and their suppliers, and the naval service experience illustrated the benefits.

Whilst the user might have a limited choice of geared turbine plant, Mr Murrell would suggest that the persistence of through-hardened gears for merchant ships was not due to design preference. He would support Mr Young's opinion that it resulted from the lack of experience, knowledge and capability of designing and manufacturing surface-hardened gears, by some suppliers. For some 20 years and more Mr Murrell's Company had been a protagonist of the use of carburized or induction-hardened gear elements throughout the gearing for turbine and diesel propulsion.

The views expressed by the authors on scuffing were relevant, as would be expected from a manufacturer of marine lubricants. It was interesting to note Mr Galvin's reference to the "D" criterion. The contributor's Company had adopted the reciprocal to this ratio, λ, as a criterion in scuffing assessments since 1976; and, based on the analysis of disc and gear scuffing research, recommended a value of $\lambda = 2$ as a minimum acceptance. Reference to the influence of retained austenite on scuffing was again relevant but he considered this influence to be marginal. In this respect it must be stated that, whilst deep freezing of carburized steels would reduce the level of retained austenite, it would also reduce the bending fatigue strength of the steel. For a slight gain in scuffing resistance, a risk of tooth breakage might result.

It was interesting to note the views of the authors with regard to pitting, and their acceptance, almost, that pitting was inevitable and satisfactory provided that it was not progressive. Pitting of any form in his Company's opinion, constituted a failure of a gear and could be avoided if the design were adequate. The exfoliation of a surface-hardened gear applied to a nitrided or similar shallow case depth treatment and should not occur with a properly carburized or induction-hardened gear.

Finally, in respect of the case histories outlined in the paper, Mr Murrell would support the contribution by Commander Cooper in that the data presented closely resembled failures experienced in the Royal Navy 30 years previously. Through NAVGRA, the technology and experience was now available to avoid such failures and was free to be used by the NAVGRA member firms in merchant marine fields. It must also be stressed, as stated by Mr Kerpestein, that high quality assurance was available, but at a price, and the question was whether the users were willing to pay the price for such increase in quality. With reference to the Stal-Laval new technology incorporated in the VAP machinery being at "the beginning of the road", the NAVGRA member firms were, indeed, well down that road.

Dr-Ing. H. MEIER-PETER (AG Weser, Bremen) wished to comment on one of the case studies mentioned by the authors.

To a gearmaker it was always alarming to learn about distress in one of the supplied gear-boxes. If the same disturbances were reported from a second, or even a third, gear-box of the same design it started to become frightening and, if the first replacement gear started to fail, it was a nightmare.

However, he could add some information to the case of the diesel reduction gear mentioned by the authors by pointing out that, at the time when the first indication of failure was noticed, the second and third gear set had already been put into operation, and the replacement gear set had already been manufactured—being intended as a supplementary spare. Corrective action, duly taken, saved the second and third gear set.

The first gear set had been installed in the new building close to, but not within, the given assembly tolerances. It was agreed to evaluate the alignment according to the dynamic tooth contact pattern found following the commissioning of the vessel. This

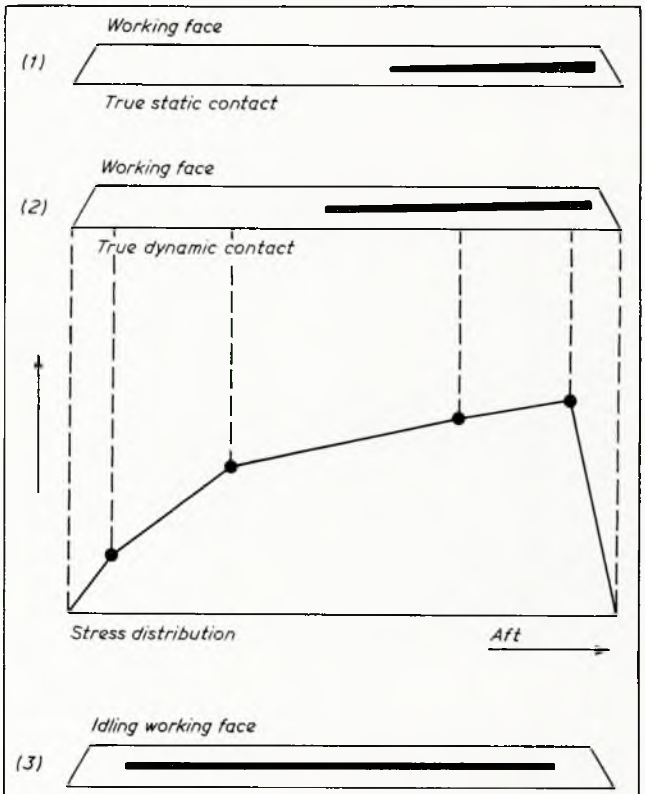


FIG. D9 Tooth contact pattern of idling pinions' working faces

1) CHESTERS, W. T., 1978, "Study of the Surface Fatigue Behaviour of Gear Materials", *International Conference on Gearing, I Mech E, London.*
2) KINTOSHITA, B. and OGVRA, S., 1973, "Experiments on Gear Tooth Surface Deterioration", *J.S.M.E.*
3) YOUNG, I. T. and CHARLES, C. T., 1973, "Evolution of a New Range of Marine Turbine Gear-boxes", *IMAS, Trans I Mar E.*

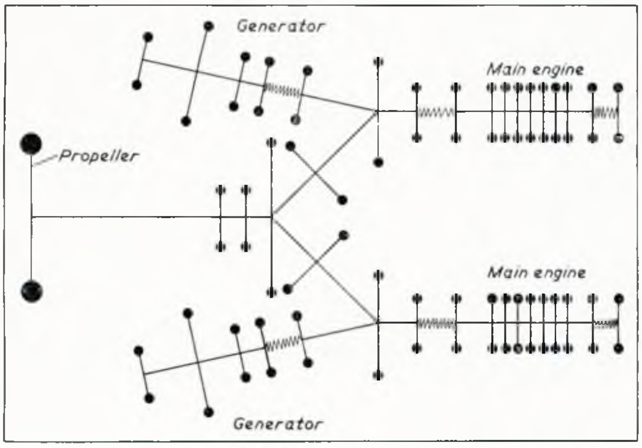


FIG. D10 Mass-elastic system

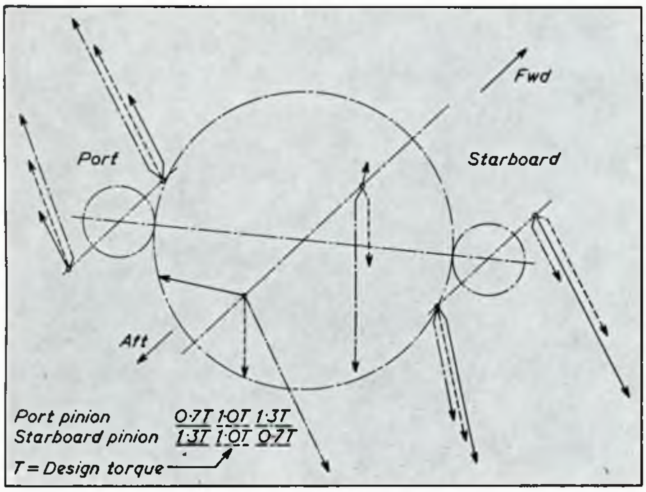


FIG. D11 Effects of torque fluctuation on bearing reaction forces

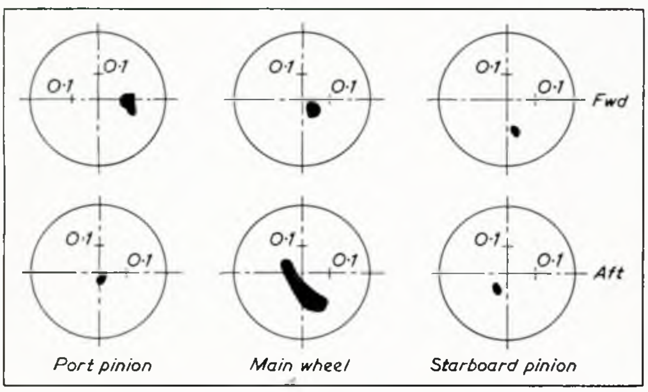


FIG. D12 Engines synchronized to avoid dynamic misalignment

contact pattern was later evaluated independently by several experts and was unanimously judged to be perfect, despite the fact that the gear was heavily misaligned.

The explanation lay in the fact that the ship was the first of a series involving extensive dock and sea trials in "single engine" operation and "part load" operation modes.

Looking at the tooth contact pattern (Fig. D9) of the idling pinions' working faces, after several hours of single engine operation with the other engine, indicated an absolutely perfect contact. The dykem contact pattern of the working faces of the same pinions, after several hours of full load operation with the according engine, indicated severe misalignment. The cumulative contact of these two operational modes looked like an absolutely perfect tooth contact.

He would comment on the vibrational performance of the propulsion system, as that was rather complex with regard to the many possible modes of operation.

Fig. D10 showed the mass-elastic system: note the fly wheel, added at a comparatively late stage of design. After the first formation of pittings had been noticed, measurements revealed that the vibratory torque was just within the given limits of ± 30 per cent. Applying this torque fluctuation alternatively to the port and starboard pinion, taking account of a certain phase angle, resulted in the effects on the bearing reaction forces shown in Fig. D11.

Whilst the effects on the pinion bearings and the forward bull wheel bearing were very limited, the aft bull wheel bearing reaction force was permanently shifting between two extreme directions. This resulted in a wobbling action of the journal and a dynamic misalignment. In order to avoid that the engines had to be synchronized (see Fig. D12).

Finally, it should be highlighted that the replacement gear wheels were manufactured at a very early stage when none of the problems had been recognized. The wheels were manufactured to and complied with a given standard which, at that time (almost a dozen years ago), was considered normal in through-hardened gears.

The figures given and the conclusions drawn in the paper were obtained from the material taken out of the first gear set which, in its service life, had been repeatedly manually honed in order to smooth the distressed area. These honing marks are clearly visible in Figs. 10, 12 and 13.

The contributor fully agreed with the authors' conclusion that high standards of accuracy with regard to manufacturing and installation of modern gear designs were compulsory requirements. Modern measuring techniques and improvements in toothing tool designs had been developed in the past few years and had already resulted in vital improvements in gear accuracy and surface finish.

MR A. D. RUSCOE, CEng, FIMarE, said that Fig. D13 showed the "rabbit wire" pattern of tooth marking in the main reduction gears of a tanker.

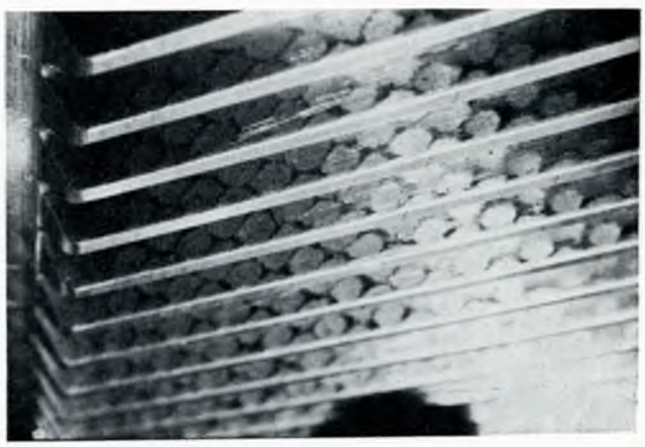


FIG. D13 "Rabbit wire" tooth marking in a main reduction gear

The exceptionally pronounced hobbing marks, besides presenting an extreme example of macro and micro-surface finish profiles, did suggest that something instructive might also be deduced from the state of the tooth flanks, which were not showing signs of distress.

Except where it was parallel with the rotational axis of the gears, the load-bearing surface of the flanks varied from less than 10 per cent of the tooth width to perhaps 25 per cent at the right hand side of the photograph. The whole of the tooth width had the same marking on both helices.

How was it that, with such a small load-bearing surface, the teeth did not scuff? Suppose that with relatively heavy wear, as there was locally with running-in processes, the individual particles of wear debris were very small and did not bridge the oil film at other than the high spots. These tended to collect together in the direction of rubbing until, in the case of a scuff, they did collectively bridge the oil film and score the surface, producing more and coarser debris which accelerated the scuff. It was to avoid this accumulation, amongst other things, that one rotated a plunger, when lapping it in, for instance.

Now suppose that the load-bearing surface was interrupted in the direction of rubbing, as by the hob marks in the tooth flanks of the Fig. D13. The wear debris was deposited in these depressions before it had accumulated and rolled up, and was dispersed in these depressions by the oil present in them and washed away by the oil spray when the teeth passed out of contact. (The same effect obtained when circumferential grooves were provided in a plunger surface.) Thus there was a rapid running-in process without scuffing and, because it was rapid perhaps, pitting did not arise either.

However, if one did not accept the above explanation, but put it down to good alignment (as it was in that case), then taking things literally, it could be said that the large face width of non-case-hardened double helical gears was illusory as far as the face loading was concerned, and that satisfactory performance could be obtained with far smaller face widths. It was hard to believe that the elasticity of such large gears and boxes could to some extent compensate for malalignment because most, if not all, double helical gears seen by the contributor showed some evidence of distress. The only set of case hardened and ground gears the contributor had seen were very narrow single helical gears showing a barely discernible difference in reflected light due to the light polish where the teeth mated, and this was uniform over the tooth width. Two factors helped the full exploitation of the full tooth width: the narrow width and the single helix which permitted *in situ* realignment for helix angle differences. This was not possible without shaving on double helices.

MR D. McKINLAY (Lloyd's Register of Shipping) said that the authors were to be congratulated for producing a paper with such a wealth of practical information on gearing.

In the section dealing with lubrication and maintenance they had indicated that they considered the lubricant should be treated in the same way as the gear steel. Mr McKinlay would like to ask if they were suggesting that the Classification Societies should incorporate requirements for lubricant properties into their Rules.

He was not aware of any Classification Society which did this and Lloyd's Register of Shipping had never considered it desirable or necessary. They were, of course, prepared to give advice when asked, but Rules were another matter. Did the authors consider that the detailed implementation of such Rules on a world-wide basis was possible when dealing with a fluid which, unlike the steel parts, was considered to be a renewable item?

MR P. THORNBLAD (Stal-Laval Ltd) said that in any system, feedback was essential for stable, controllable progress. Technical development was no exception to this general rule. Therefore, for those working on the development of marine gearing, a survey of the kind the authors had presented was valuable.

He also hoped that their aim to generate a lively discussion in such areas where knowledge was incomplete, would be fulfilled. Even if a great deal were known about gearing behaviour today, there were fields in which the designer was left with rather blunt tools to match the relentless requirements for reliability and long life with economy in production. He agreed with the authors' view that the interaction between the tooth flanks, their surface roughness and the oil film was one area in which the practitioners were still waiting for the scientists to provide more useful knowledge.

He thought it true to say that modern gears never failed due to the stresses for which they had been designed. If they did fail it was because the stresses had been increased either from inadequacies in the gears themselves or from external disturbances not allowed for.

Fig. D14, which was essentially the same as Fig. 9 in the paper, showed the allowable K factor (Lloyd's Rules) and the allowable

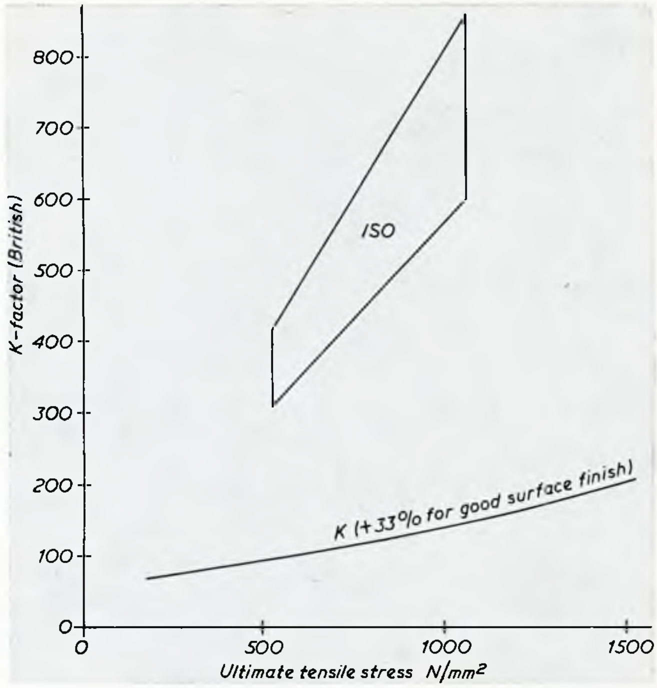


FIG. D14 Allowable K factor and Hertzian stress

Hertzian stress issued by the ISO Committee on gear rating. From the difference between the values it could be concluded that, when a gear designed to Lloyd's Rules started to develop pitting, the actual load must be considerably higher than the design load, and the cause must be either inside the gear itself or external.

Installation and alignment was always raised when something went wrong with marine gears. The Stal-Laval AP machinery suffered a series of epicyclic gear failures to which the authors had briefly referred and which were the subject of Mr T. P. Jones' paper in 1972 (Ref. 14 of the paper).

In a number of those cases running misalignment played a very significant role. A series of actions including extensive measurements to monitor dynamic alignment under various conditions and realignment of a number of gears as a result of those measurements, solved the problems.

The final reduction stages of the AP machinery had had few problems—the two most serious ones were referred to in the Case Histories—and none could be related to misalignment.

Part of that good record was due to careful installation work at the ship-yards and, since many of the ships were built as a series, the alignment procedure was gradually improved through feed-back from sea trials experience. Part of the success was ascribed to the special design features of the gear casing. Each of the wheel/pinion meshes was supported at three points, one under each wheel bearing and one at the middle of the pinion. Thus each pinion rested in a state of balance (see Fig. D15) so if for any reason the tooth load had a tendency to increase towards one end of the pinion, either from internal misalignment or from external disturbances, the balance would alter to relieve the heavier loading and restore the equilibrium. This was made possible by the high flexibility of the gear casing. The spring boxes, one at each corner of the gear case were also shown in principle in Fig. D15. By twisting the gear casing with the aid of these springs the contact marking was adjusted, both when the gear was installed on board and after sea trials.

Case histories 5.3 and 5.4 involved Stal-Laval gears, and neither was related to the ships in Table I of the paper. The gear damaged due to the blocked filter belonged to a tanker commissioned in 1968. The damage was discovered during a survey in 1977 and, as far as it could be established, the cloth must have been put into the filters prior to sea trials. In spite of a gradually decreasing oil flow, nothing abnormal was found in the gears during the subsequent nine years' service.

The repair was all hand work. All pits exceeding 2 mm diameter were smoothed by a high-speed grinding machine. The other

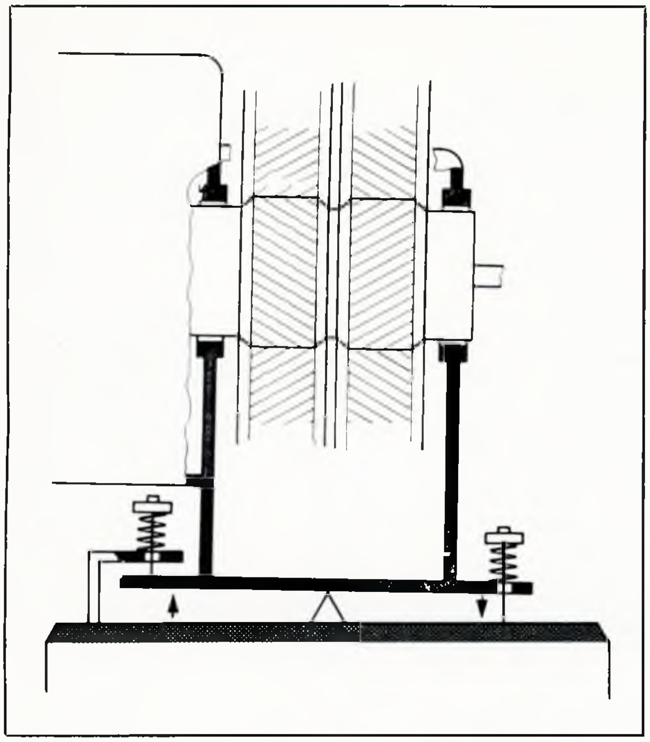


FIG. D15 Gear casing support with adjustment device

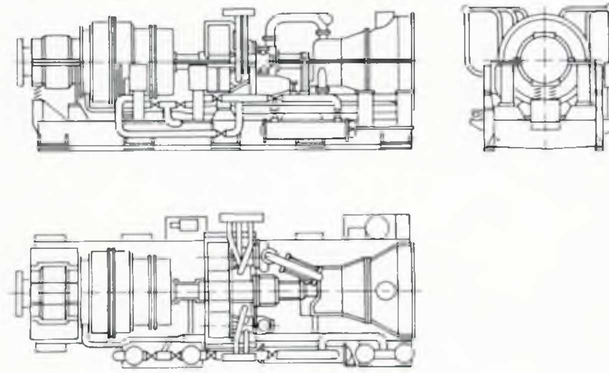


FIG. D16 Layout of VAP plant

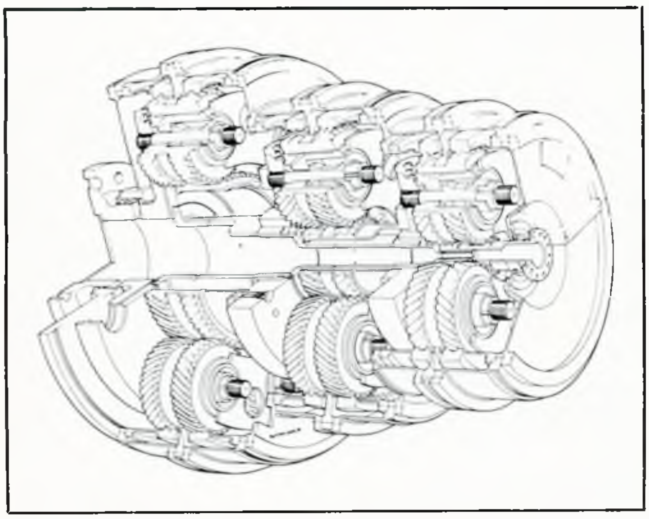


FIG. D17 Three stage version of Compact Planetary Gear

damaged areas were first polished with oil stone and fine emery paper and finished with castings made from a plastic resin filled with carborundum, moulded from the undamaged part of the teeth. It was interesting that the gear was still operating satisfactorily. Considering the limited accuracy that could be achieved with such repair work, it raised doubts as to whether the ultimate quality aimed for in manufacture was worth the extra cost.

The scuffing case occurred during the sea trials in a ship similar to some of those in Table I, but not belonging to the Shell fleet. During the hobbing of the wheel, the cutter fractured and left deep marks in the teeth and the wheel had to be recut with an addendum modification factor of minus 0.66. The pinions were made oversize to maintain the same centre distance. This increased the maximum sliding speed from the normal 1.5 m/s to 2.4 m/s but, since it was a slow speed gear, this value was still well within the company's range of experience. The risk of scoring was checked both by the company and by an independent research institute, and the conclusion was that the gears were satisfactory for the intended service, provided the following requirements were achieved:

i) good load distribution;
ii) specified surface finish;
iii) adequate tip and root reliefs.

As an extra precaution and because the calculations showed the friction losses in the gear meshes to have increased by some 50 per cent, the design of the oil sprayers was changed to give approximately 50 per cent more oil to each mesh. For various reasons this modification was not carried out fully, as all the nozzles were not given increased hole dimensions. The severity of the damage on different parts of the gears could to an extent be related to the size of the nozzles.

The intention was that the increased lube oil flow would remove the increased heat losses and ensure that the gears operated at a normal temperature level. It was possible, although it could never be proved, that had the intended improvement of the cooling been achieved this gear-box would not have failed.

Although it was educational to look back, it was more interesting and exciting to look forward to what was coming.

Stal-Laval's new turbine machinery, the VAP plant, was referred to as a "beginning of the road" technology (Ref. 1) and Mr Thornblad wished to discuss the gearing design.

Fig. D16 showed a layout of the VAP plant. Its most interesting feature from a gearing point of view was the use of a two-stage epicyclic gear as the final reduction. It was mounted overhung on the main shaft and the torque reaction was transmitted to the foundation via a flexible link structure, permitting the whole gear to ride on the shaft and follow its deflections and movements; so avoiding the gear being influenced by external disturbances and making the shafting alignment a less critical process.

The gear was of a new Stal-Laval design, the Compact Planetary Gear (CPG), derived from the Stoeckicht design used in almost 300 AP-installations. The development was based on extensive experience from tests, service, measurements and analysis of these gears. A CPG prototype was built and tested in 1977 and was later installed in a tanker where it had since been in operation for over a year.

Fig. D17 showed a three-stage version of the same basic design. The two first gears of this kind, intended for American wind power plants, had just completed a 200 h back-to-back test programme under various conditions, including measurements of tooth stresses.

All the main components of a VAP prototype plant were being manufactured and would be tested within a year including two final reduction gears which would be run back-to-back at full power and speed.

Finally he would like to ask two questions. The first related to Fig. 8 where the loss of flank area was expressed in mm^2. Mr Thornblad thought the information would be more useful if it was also stated as a percentage of the active flank area. Secondly, in Section 5.2 "dynamic effects" were said to have changed the alignment. Were those "effects" alignment changes during operation as opposed to the static alignment at the installation, or did shafting vibrations have an influence?

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1) NORBERG, L., 1978, "The VAP Turbine Machinery" *Steam Propulsion for Ships in the Changing Economic Environment, Trans I Mar E Conf.*

Authors' Replies

Because of the number of contributions received it was not, unfortunately, possible for the authors to reply to contributors individually, but similar comments relating to specific aspects of marine gear design and operation have been dealt with on a subject by subject basis.

Gear design

The choice of design did not, as was suggested in discussion, lie with the merchant shipowner. In most cases, even if he were to prepare a detailed statement of requirements, he would still be obliged to accept what a particular shipyard offered because the commercial penalty for insisting on a manufacturer not associated with the shipyard was invariably very high.

A change from designs "hallowed by re-use" was indeed overdue in some quarters and there was general need for innovation and more forward-looking design concepts, as mentioned by various contributors. Some older designs seem unduly dependent on maintaining very high standards of manufacture and installation. There was a strong case for using hard-on-soft gear material combinations in turbine as well as diesel application. A number of manufacturers did not, however, have the facilities or the know-how to produce and finish hardened gear elements. The authors' experience of hardened gears was confined to very satisfactory experience with nitrided elements. Case exfoliation, mentioned in the paper, was believed to be rare with a high quality gear. Carburized, case-hardened or induction-hardened gears were also reported to give thoroughly reliable service in merchant applications. With the considerable experience of the Navy also, the authors would readily endorse the use of hard-on-soft gears.

As regards quality control Shell companies had, of course, the assistance of the Classification Societies. But, in addition, superintendents were appointed for all new buildings, and part of their work was to ensure satisfactory product quality.

Naval applications were exceptional, since there was not nearly so much concern with commercial considerations and the specifications could, therefore, be much more demanding. Again, the operating profile of a Navy vessel called for sustained full power to far less an extent than that of a merchant ship, particularly the tanker which, until recently, was operated at full power continuously whilst on open sea passage. In Great Britain, The Netherlands and elsewhere there had been considerable spin-off from Naval gear research which had certainly benefited merchant marine gear design. Unhappily, this special expertise was not shared by all manufacturers.

A final design point concerned the application of helix correction. Its necessity was dependent on the pinion design and loading and, in most cases, the gearing in Shell ships had been corrected for combined torsion and bending. Helix correction was considered desirable in principle, but had to be applied intelligently and accurately to be effective in practice. As Dr Archer had mentioned, it could be applied more accurately by grinding than by any other method. Correction for thermal effects would be beneficial, but more difficult to predict accurately.

Installation

The importance of good quality carried right through design and manufacture to installation. It was most important that the responsibilities of the shipyard and the manufacturer were thoroughly understood. There was still too little regard to covering the interfaces where responsibilities should be shared.

In respect of alignment there was no conflict between the statements in Sections 3 and 5 of the paper. The remarks were intended to emphasize that the smaller the amount of adjustment needed to achieve good alignment in the ship, the more likely the gear was to perform well in service. An additional benefit came from designs which incorporated some convenient device for optimizing an already good alignment, e.g. spring boxes or adjustable bearings.

Much of the uncertainty in alignment between gearing and propeller shaft could be removed by using a flexible coupling. Once again, it was largely a matter of cost. As long as there was no urgent need, indicated by a high incidence of gear/shaft problems, such couplings were unlikely to be developed.

Another relevant point concerned the differential loading of main wheel bearings. Dr Archer had offered a graphic illustration of the influence it could have on gear alignment. The extent of misalignment due to main wheel skew was not always appreciated and the case quoted was a timely reminder of how helix correction, carefully calculated and applied, could be completely nullified by such effect. In those cases within the authors' experience, bearing reactions as measured by both the jacking and strain gauge methods, had been remarkably close to those predicted by computer for a particular shaft/bearing arrangement. Main wheel bearing reactions had been well within manufacturers' tolerances for both loaded and ballast conditions. In the General Purpose ships, one or two cases arose in which load sharing between the main wheel bearings was found to be outside limits after the ships had been in service for some time, but in no case was the differential greater than 20 per cent. The authors agreed that the forward bearing should be the heavier loaded in light ship condition, thus compensating for the hogging of the structure in loaded condition giving generally an increase in loading of the aft bearing.

The shuttling or wobbling of the pinions mentioned by Dr Meier-Peter was a phenomenon discovered by full-scale tests carried out, using strain gauges in the main wheel teeth and position transducers, to locate the relative positions of the pinion journals in their bearings. It would be expected to be present in any single helical installation of similar configuration and would be influenced both by engine phasing and by load distribution between the main wheel bearings.

The integral thrust design was not particularly favoured and could, in the case history concerning alignment, have been at least partly responsible for the basic cause of trouble. Mr Volcy's classic contribution underlined, again, the importance of understanding hull/machinery interactions. In very large ships particularly, there was no single alignment, however accurate, which could perfectly satisfy all ship conditions. Intelligent compromise was necessary, and was reached by what he had described as rational alignment.

A final point regarding propeller shafting concerned that of single or two-bearing designs. Most Shell VLCCs had two intermediate shaft bearings, but some had a single bearing design, as had all the General Purpose ships. The only cause for concern had been two single bearing designs which showed a tendency to whirling. That was both observed and measured and found to be, in fact, counter whirl in both cases. The amplitudes were not considered dangerous and the phenomenon was regarded as undesirable but containable. In general, both single-bearing and two-bearing designs have proved satisfactory and the authors would hesitate to judge which was the better. Certainly the conflict between the relative stiffnesses of hull and shafting would appear to favour the choice of a single-bearing design. The real answer lay in making sure that, whichever design was adopted, the installation was accurate and a thorough check of dynamic shaft behaviour was made before the ship was accepted for service.

Pitting, scuffing and wear

Pitting and manufacturing blemishes were most certainly not accepted by the authors as normal or inevitable. However marginal, any case of pitting which occurred in Shell ships was thoroughly investigated. Pitting was recognized as a warning which might, or might not, herald more sinister developments but in any case it should not occur. Even light initial pitting was considered undesirable in modern gears. Absence of pitting was attributable to design and high quality manufacture of the gears, using the correct material combination for pinion and wheel, and installing the set to give optimum alignment. To answer a specific question, no pitting at all had been experienced with the final reduction gears of STAL AP engines.

In a similar context a number of contributors made highly pertinent comments concerning surface wear and one or two referred to surface blemishes. A particular reference to the possible association of swarf with surface blemishes appeared valid, but the effect had not been proven in practice on Shell ships. Mr Lynam had referred to the action of anti-rust additives where fissures or machining-related damage existed. The variation in type and nature of the rust inhibitors precluded a definite answer. There was evidence that a rust inhibitor prevented water from initiating fatigue cracks, at least in rotating bending tests. In the fissures referred to, the additive was likely to be less effective. In gear applications where gross damage had occurred, pre-formed cracks might extend owing to a hydraulic crack propagation mechanism.

Several comments had been made about surface finish. In measuring gear teeth, several cut-off lengths were used as a measure of the interaction of waviness and surface roughness but the authors had no direct evidence as to their relative importance.

Unfortunately, in the cases of damage quoted in the paper, surface replicas were not made prior to the gears entering service and that was now recognized as an important omission. The practice in future should be to obtain replicas, whenever possible, of new gears.

As regards the use of EP oils with respect to pitting, in the particular case of the diesel application the oil was not used as a healing medium; the change from one type of EP oil to another was, as stated in Section 5.2, merely a change to a higher viscosity oil in an attempt to increase oil film thickness. Some EP oils might cause, by chemical wear, a smoothing of grossly roughened surfaces. Thus, the load-carrying area of the surfaces was increased and a general smoothing out of the pit profile resulted in a reduction in specific loading. However, it was known that with other EP oils the corrosive action of the lubricant had caused pitting. In the case of turbine gears quoted in Section 5.3, the damage was so gross that it was believed only an improvement could result. It should be remembered that the pitting in that ship was due to oil starvation, not to overload from misalignment or other causes.

Two case histories, Sections 5.3 and 5.4, were very usefully amplified by Mr Thornblad and the addition of such details was much appreciated. In answer to his specific questions, the loss of flank area depicted in Fig. 8 could be related to the original flank area as a percentage. The flank area was approximately 116 cm^2.

The reference in Section 5.2 had referred not to shaft vibration but to some form of alignment disturbance to the pinions and wheel in their bearings, under the influence of loading and speed. Certainly it was established that the good static alignment did deteriorate under dynamic conditions.

Dr Shannon's remarks were very valuable. However, in considering the innocence or otherwise of wear steps, it was necessary to distinguish between the performance of through-hardened and hard-on-soft gears. It was believed that in the work referred to by Dr Shannon the wear steps resulted from removal of very fine wear particles, wear gradually ceasing. However, cases with through-hardened material combinations had occurred where coarse particles were eventually removed from the surface, which commenced to wear by fine spalling but finished with gross pitting.

Metallurgy and lubrication

On the question of austenite and scuffing, the position taken by the authors was that, if austenite encouraged scuffing then both retained austenite (produced on the surface during heat treatment and manufacture) and austenite produced during or in the events leading to scuffing were deleterious to the performance of the gears.

In the authors' laboratories, tests had been carried out on both disc and gear rings to study the nature and morphology of surface layers developed during the scuffing and scoring of case-hardened gears(1). Examination of the surface layers that formed during the scuffing tests, by transmission electron microscopy and Mossbauer spectroscopy, identified a heavily deformed mixture of austenite and martensite. That was consistent with the idea of a layer without a clearly defined structure type, formed from a complex combination of high temperature, resulting from frictional heating, high contact pressure and severe deformation. Consequently, if austenite was considered to take part in the scuffing process, then the presence of austenite-forming elements could be considered to be detrimental to the steel's scuff resistance, while ferrite-forming elements could be considered to be beneficial.

Metveevsky *et al.*(2) working on Fe- 1 % C alloys showed that, while additions of Cr and W were beneficial towards scuff resistance, Ni was not. In the discussion of that paper it was pointed out that steels such as EN 56 (0.3 % C, 1 % Ni, 13 % Cr) were difficult to lubricate and that was thought to be due to the absence of chromium carbides, the presence of nickel and the effect of high chromium contents decreasing thermal conductivity and thus stimulating higher contact temperatures in friction. However, the authors were not aware that the low chromium contents present in through-hardened gear steels were deleterious to scuffing resistance; disc tests had not been carried out.

In a recent review, Parrish(3) had considered the role of retained austenite and suggested that, for bending fatigue and impact resistance, the retained austenite should be limited to 25 per cent; in cases of severe sliding the limit should be 20 per cent while for contact fatigue pitting the lower limit should be 10 to 15 per cent. Parrish also pointed out that practical experience had necessitated the imposition of a limit of 15 per cent retained austenite in the steel for case-hardened worm shafts owing to the tendency to scuff when the hardness fell below 730 Hv. Because of those ranges of desirable austenite content, which depended on the component operating conditions, it might be necessary to carry out secondary processes such as sub-zero quenching, particularly in the case of highly alloyed case-hardening steels. It was not the authors' intention to suggest that all case-hardened steels needed to be sub-zero quenched and they were grateful to Mr Young for pointing out the anomaly in the text of the paper. The "rabbit wire" pattern of tooth marking shown in the slide provided by Mr Ruscoe did not come within the authors' experience but it was surprising that no damage occurred to the gears.

The length of service and design loading of the gears had not been mentioned but such a surface would, it was believed, eventually suffer pitting.

There seemed to be some confusion in the terminology used in Mr Ruscoe's contribution. "Scuffing" was the term normally used to describe gross damage characterized by the formation of local welds between the sliding surfaces. It was associated with the thermal breakdown of the oil film which did not require a third body abrasive, although in a near critical scuff situation the presence of wear particles might disrupt the oil film and accelerate scuffing. "Scoring" was defined as grooving of the surface in the direction of sliding, often caused by abrasive particles of foreign matter or wear debris. Neither phenomenon was evident in the photograph.

As far as the Classification Rules were concerned it was understood that it would be virtually impossible to include specific requirements for lubricants. The plea inferred in the paper was for designers to consider the lubricant at an early stage and not merely to present the oil companies with a *fait accompli* and a request to "lubricate this please". Early consideration of loads, speeds, temperature, oil film thickness, and "D" ratio would give some idea of the lubricant type required. Then consideration could be given to the possible effects of such oil on metals. In addition, it should be borne in mind that one lubricant might not only have to lubricate quite different sets of machinery, e.g. turbines and gears, but also have to carry out its other functions of cooling, protecting against rust and corrosion shedding water, etc.

Condition monitoring

Several questions relating to condition monitoring had been asked. Before any system was adopted it had to be considered what advantages it would give over carefully kept engine room logs, regular and conscientious reporting by ships' engineers, and regular and thorough visual inspections with meticulous recording of the results. Some conditions, e.g. on gear teeth, might be detected much more readily by a trained observer than by instrumentation.

Having decided that a monitoring system was worth considering the following were some of the questions to be answered:

i) What points were to be monitored and why?
ii) What types of transducers were to be fitted and how were they to be calibrated and maintained?
iii) How were the data to be recorded, analysed and used?

1) CAMPANY, R. G., 1977, "Scoring and Scuffing Failure in Lubricated Sliding Components", *Limits of Lubrication Conference, Imperial College, London*.
2) METVEEVSKY, A. M., SINAISKY, V. M. and BUYANOVSKY, I. A., 1975, *Journal of Lubrication Technology*, 97 (3), *pp* 515.
3) PARRISH, G., 1976, *Heat Treatment of Metals*, *Vol* 3, *No* 4, *pp* 101-109.

iv) What were the maintenance and capital costs and what was the life of the system?

In the Shell ships a fairly simple approach had been adopted. Alarm systems were comprehensive and for the most part designed to satisfy u.m.s. requirements. Gear-boxes and gears were visually inspected and blue tapes used to ensure that no changes in general condition were occurring. If damage was seen on the teeth, replicas were made. Bearing oil temperatures and pressures were measured and sometimes logged. In general, gearing had proved to be reliable. Consideration might be given to fitting position transducers to measure axial movements and to measuring oil temperatures near the mesh but those additions were not really essential.

Journal bearings have also proved reliable. Inspection of selected bearings would probably take place no more frequently than at refit; wear-down readings would be taken.

Flexible couplings were inspected at refits but, depending on design, accuracy of manufacture and heat treatment, they could be subject to wear which might necessitate earlier inspection.

On-engine instrumentation included the measurement and logging of all important pressures and temperatures using multipoint mini-recorders. Trend recording for selected engine data had proved most effective in monitoring engine performance and for safety and maintenance purposes. Its use could well be extended.

Vibration measurements could be very useful in monitoring trends but not all faults manifested themselves in changes in vibration.

Vibration signatures at key points on new engines provided a useful yardstick for assessment of future performance. They could be taken readily and were particularly helpful if taken just before a ship was due for a refit.

Published for THE INSTITUTE OF MARINE ENGINEERS by Marine Management (Holdings) Ltd., (England Reg. No. 1100685) both of 76 Mark Lane, London EC3R 7JN. Printed by Eastern Counties Printers Limited at The Jefferson Press, Ely, Cambs., England.