Paper read at Lloyd's Register of Shipping, on Tuesday, 22 April, 1975

SOME FACTORS IN MARINE GEARING FOR CLASSIFICATION PURPOSES

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The paper gives a brief history of the evolution of the Rules of Lloyd's Register of Shipping up to the adoption of the basic I.S.O. formulae for design, and analyses the main defects in both steam turbine and oil engine gearing over the past seventeen years. With this background, the values applicable to the factors for surface loading and tooth bending strength are discussed. Finally the manufacturing tolerances and installation practices, particularly with regard to external effects, necessary to obtain the conditions at the mesh appropriate to the permissible loadings, have been outlined. I.S.O. notation has been used in the paper and, where the coefficients in Lloyd's Register's published Rules differ, the correlation has been indicated.

INTRODUCTION

Classification Societies' Rules are framed within the general concept of providing the reliability and fitness of the component for the service required. For this reason they have tended to be rigid enough to guide the manufacturer away from unsatisfactory features but to allow sufficient flexibility for progressive ideas in design. This has entailed constant monitoring of difficulties encountered in service as well as of improvements in manufacturing and installation practices.

For approximately the first thirty years of marine gearing the loading was based on the original Parsons basis, being directly proportional to the product of the length and diameter of the pinion. In 1946, Lloyd's Register of Shipping published its *K* value for steam turbine gearing, based on the Hertzian stress related to the load per unit length of face width. In those days the pinions were 3·5 per cent nickel alloy steel, having a tensile strength of not less than 620MN/m^2, whilst the material for the wheel rims was carbon steel, having a minimum tensile strength of 480MN/m^2. The increasing use of alternative alloys led to a revision in 1956 including an allowance for tensile strength.

Over the next ten years the accelerating use of surface hardening for turbine gearing and the upsurge in geared diesel installations, considered in conjunction with the work of Archer(1) and Davis(2), resulted in the revision of 1966. These modifications also took account of the incidence of tooth failures and consequently bending strength was included in the assessment.

Increasing use of computers, allowing the possibility of iterative methods and the incorporation of additional factors, coupled with the agreement in I.S.O. on a basic formula in 1967, has given greater scope to the Rule concepts, and in 1973 an extensive revision was made, incorporating the basic I.S.O. formulae for design. I.S.O. has still not agreed all the factors, but experience has been used by Lloyd's Register to determine Rule values, where necessary. Having settled the formulae for the Rule loading in a way which allows scope for the future and reflects the immense progress over the second half of marine gearing history, the emphasis now has to be concentrated on the manufacturing and installation aspects.

BACKGROUND

Table I gives the statistics for main propulsion geared installations classed with Lloyd's Register of Shipping over a period of 12 years.

TABLE I—DISTRIBUTION OF MAIN PROPULSION GEARED INSTALLATIONS LLOYD'S REGISTER CLASS SHIPS

PRIME MOVER	YEAR	ENGINES AFT	ENGINES AMIDSHIPS	TOTAL
STEAM TURBINES	1960	516	409	925
	1965	602	329	931
	1972	691 (366)	292 (78)	983 (444)
OIL ENGINES (OVER 500 BHP)	1965	324	623	947
	1972	887 (758)	1007 (871)	1894 (1629)

NOTE: Figures in parenthesis apply to vessels built since Dec. 31st, 1959.

Whilst the overall number of steam turbine installations has varied little over the period, a greater proportion are now placed as far aft as possible and the individual powers have increased with the size and type of vessel. Whereas, in 1960, 56 per cent were placed aft, for vessels built since that time the proportion has risen to 83 per cent. This trend is likely to continue, giving relatively stiffer installations and thus making the gearing more susceptible to external influences.

There has been a large increase in the number of geared diesel installations, actually doubling their number in seven years. There has been the same tendency to increased individual powers as in steam turbine installations. More diesel engine installations than steam turbines are situated amidships but, even here, the percentage has dropped from 66 per cent to 54 per cent of the total.

The results obtained from the constant monitoring of defects in geared installations are given in Tables II and III for steam turbine and oil engine installations of over 375 kW (500 bhp) respectively. The periods between the major changes in the Rules have been broken down as follows:

Period A) From the end of 1955 to the end of 1960 (5 years) for ships built after 1946 and in class at the end of 1960.

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TABLE II—DEFECTS IN MAIN PROPULSION TURBINE GEARING LLOYD'S REGISTER CLASS SHIPS

		ENGINES AFT						ENGINES AMIDSHIPS						TOTAL (AFT AND AMIDSHIPS)					
Type of defect	Period	1st Reduction	2nd Reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum	1st Reduction	2nd Reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum	1st Reduction	2nd Reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum
Scuffing	A	–	14	14	0·54	2	0·08	–	1	1	0·05	–	–	–	15	15	0·32	2	0·04
	B	–	3	3	0·10	–	–	–	–	–	–	–	–	–	3	3	0·06	–	–
	C	–	1	1	0·04	–	–	–	–	–	–	–	–	–	1	1	0·03	–	–
Pitting	A	9	28	29	1·12	4	0·16	5	8	11	0·54	2	0·10	14	36	40	0·86	6	0·13
	B	1	17	17	0·56	3	0·10	1	6	7	0·43	–	–	2	23	24	0·52	3	0·06
	C	1	7	8	0·31	2	0·08	–	–	–	–	–	–	1	7	8	0·26	2	0·06
Tooth Fracture	A	5	9	13	0·50	9	0·35	3	2	5	0·24	3	0·15	8	11	18	0·39	12	0·26
	B	11	7	18	0·60	11	0·37	2	1	3	0·18	1	0·06	13	8	21	0·45	12	0·26
	C	4	3	7	0·27	7	0·27	–	–	–	–	–	–	4	3	7	0·23	7	0·23
Rim slipped	A	–	2	2	0·08	2	0·08	2	–	2	0·10	–	–	2	2	4	0·09	2	0·04
	B	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
	C	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Rim fracture	A	1	1	2	0·08	2	0·08	–	1	1	0·05	1	0·05	1	2	3	0·06	3	0·06
	B	2	–	2	0·07	1	0·03	–	–	–	–	–	–	2	–	2	0·04	1	0·02
	C	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Total gear sets defective	A	13	44	50	1·94	18	0·70	10	12	18	0·88	5	0·24	18	56	68	1·47	23	0·50
	B	14	24	37	1·23	15	0·50	3	7	10	0·61	1	0·06	17	31	47	1·01	16	0·34
	C	5	11	16	0·63	9	0·35	–	–	–	–	–	–	5	11	16	0·52	9	0·29

TABLE III—DEFECTS IN MAIN PROPULSION OIL ENGINE GEARING LLOYD'S REGISTER CLASS SHIPS

Type of defect	Period	ENGINES AFT						ENGINES AMIDSHIPS						TOTAL (AFT AND AMIDSHIPS)					
		Single reduction	Double reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum	Single reduction	Double reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum	Single reduction	Double reduction	Total gear sets defective	Percentage sets defective per annum	Total gear sets renewed	Percentage sets renewed per annum
Scuffing	B	1	–	1	0·06	–	–	1	–	1	0·03	–	–	2	–	2	0·04	–	–
	C	2	–	2	0·04	1	0·02	–	–	–	–	–	–	2	–	2	0·02	1	0·01
Pitting	B	3	–	3	0·18	–	–	7	1	8	0·26	–	–	10	1	11	0·23	–	–
	C	7	4	11	0·21	1	0·02	6	3	9	0·15	1	0·02	13	7	20	0·18	2	0·02
Tooth Fracture	B	4	1	5	0·31	4	0·25	1	–	1	0·03	1	0·03	5	1	6	0·13	5	0·10
	C	13	–	12	0·23	12	0·23	3	–	3	0·05	3	0·05	16	–	15	0·13	15	0·13
Total sets Defective	B	7	1	8	0·49	4	0·25	9	1	10	0·31	1	0·03	16	2	18	0·38	5	0·10
	C	21	4	25	0·47	13	0·25	9	3	12	0·27	4	0·07	30	7	37	0·33	17	0·15

Period B) From the end of 1960 to the end of 1965 (5 years) for ships built after 1946 and in class at the end of 1965.

Period C) From the end of 1965 to the end of 1972 (7 years) for ships built after 1959 and in class at the end of 1972.

The last period does not include the failures in epicyclic gears which have been described by Jones[3]. These were special types and no troubles have been experienced since they were brought into line with the general consideration of parallel-shaft gears.

Scuffing has shown a dramatic decrease in turbine gearing during the later periods, reflecting the change from the all-addendum gears formerly used in the primaries and the 14·5° deep tooth in the secondary gears. Its virtual elimination has also been aided by the greater machining accuracy including finer surface finish and improvements in lubrication. The phenomenon of scuffing has not been a serious factor in oil engine gearing during the periods covered. As a result of these findings, no special requirements to counteract scuffing have been incorporated in the Rules other than warnings on tooth form.

The incidences of pitting in steam turbine installations have decreased, with the total number of sets defective falling from 1 in 116 years total service in period A) to 1 in 386 years in period C). In addition to the changes in tooth form etc. mentioned in connexion with scuffing the problem of pitting may have been reduced by the greater use of surface hardening coupled with post-hobbing processes. In oil engine installations pitting has not been quite the same problem. Nevertheless, there has been a welcome drop in pitting in amidships installations but aft end installations appear likely to occupy attention for some time to come. For both turbine and oil engine installations the question of load distribution across the teeth and therefore alignment has to be tackled if the full potential is to be achieved.

The greatest disaster in gearing is tooth fracture because, even if there is no consequential damage due to a piece passing through the mesh, it may mean renewal of one or more components and time loss in service. For this defect also, oil engine installations have a somewhat better record than turbine installations. However, for both types of prime mover there appears to be a persistency in the number of total failures over the complete period surveyed, with those situated aft suffering the majority of the casualties. This indicates the adverse effect of uneven loading across the teeth and puts emphasis on good initial alignment since the shafting of such installations will not be able to supply the degree of flexibility available in amidships installations. The incidences in the primary mesh of steam turbine gears and in single reduction oil engine gears would also indicate the need for care with the connexion between the gearbox and the prime mover.

The experience shown by these statistics has formed a basis for consideration of the various coefficients in the Rules.

PART 2

SYMBOLS AND TERMS

Suffixes 1 and 2 refer to pinion and wheel respectively,

B = total axial length over the gear face, including gap, where applicable,
d = reference diameter of gear,
d_o = diameter at root of teeth,
d_s = diameter at shrinkage surface of gear rim,
E = modulus of elasticity,
h_F = bending moment arm for tooth root stress,
$K_{F\alpha}$ = transverse load distribution factor (due to pitch errors) for tooth root stress,
$K_{F\beta}$ = longitudinal load distribution factor (due to misalignment, deformation etc.) for tooth root stress,
$K_{H\alpha}$ = transverse load distribution factor (due to pitch errors) for Hertzian stress,
$K_{H\beta}$ = longitudinal load distribution factor (due to misalignment, deformation etc.) for Hertzian stress,
K_I = application factor,
K_L = lubrication influence factor,
K_V = dynamic load factor,
K_{XF} = size factor for tooth root stress,
K_{XH} = size factor for Hertzian stress,
$m_n = \frac{p}{\pi}$ = normal module,
n = rev/min,
p_n = normal pitch,
P = radial pressure at shrinkage surface of gear rim,
r_f = root fillet radius,
S_F = factor of safety for tooth root stress,
S_H = factor of safety for Hertzian stress,
S_{nF} = tooth thickness in the critical section at the tooth root in the normal section,
u = gear ratio $= \dfrac{\text{number of teeth in wheel}}{\text{number of teeth in pinion}}$
V_w = pitch circle velocity,
W = load per unit length of face width on the reference cylinder in the transverse section,

$$\left.\begin{array}{ll} Y_\beta & = \text{helix angle factor} \\ Y_\varepsilon & = \text{load sharing factor} \\ Y_F & = \text{tooth strength factor} \\ Y_S & = \text{stress concentration factor} \end{array}\right\} \text{for tooth root stress,}$$

Z_H = zone factor $= \sqrt{\dfrac{\cos\beta_b \cos\alpha_{tw}}{\cos^2\alpha_t \sin\alpha_{tw}}}$

$$Z_M = \sqrt{\frac{1}{\pi(1-\nu^2)}\;\frac{2E_1 E_2}{E_1+E_2}}$$

Z_{RF} = roughness factor for tooth bending stress,

$$\left.\begin{array}{ll} Z_{RH} & = \text{roughness factor} \\ Z_V & = \text{speed factor} \\ Z_\varepsilon & = \text{contact ratio factor} \end{array}\right\} \text{for Hertzian stress,}$$

z_n = virtual number of teeth,
α_n = pressure angle in the normal section on the reference cylinder,
α_{nF} = pressure angle in the normal section for tooth stress,
α_t = pressure angle in the transverse section on the reference cylinder,
α_{tw} = working pressure angle in the transverse section,
β = helix angle, in degrees, on the reference cylinder,
β_b = helix angle on the base cylinder,
ε_α = transverse contact ratio,
ε_β = overlap ratio,
σ_F = bending stress in the critical section at the tooth root,
σ_{Flim} = fatigue strength for tooth root stress,
σ_{FP} = permissible bending stress at the tooth root,
σ_H = Hertzian stress,
σ_{Hlim} = fatigue strength for Hertzian stress,
σ_{HP} = permissible Hertzian stress,
σ_U = ultimate tensile strength of gear material,
ν = Poisson's ratio.

BASIC FORMULAE

The I.S.O. formulae[4] were chosen for development of the Rules for gearing since they were not only considered the most suitable means of expressing new experience but it was also thought that they offered the most promising basis for unification throughout the Classification Societies. These formulae consider a thin section of a spur tooth and they apply factors for finite dimensions, manufacture and service. The advantage lies in the fact that such factors may be reduced, simplified or expanded as desired. Helical gears are considered as special types of spur gears.

The factors will be discussed following the statement of the basic formulae for suiface loading and tooth bending stress.

a) *Surface Loading*

The Hertzian stress at the operating pitch circle is given by

$$\sigma_H = Z_H Z_M Z_\varepsilon \sqrt{K_I K_V K_{H\alpha} K_{H\beta} \frac{W}{d_1} \frac{(u+1)}{u}} \qquad (1)$$

where, for steel gears, $Z_M^2 = 72\,060$ MN/m².

This stress must be less than the allowable Hertzian stress which I.S.O. defines as

$$\sigma_{HP} = K_L Z_{RH} Z_V K_{XH} \frac{\sigma_{Hlim}}{S_H} \quad (2)$$

I.S.O. advises that K_L, K_{XH} and $K_{H\alpha}$ may be unity provided the factor of safety be chosen sufficiently high.

Since the Lloyd's K value is defined as

$$K = \frac{W}{d_1} \frac{(u+1)}{u}$$

the criterion gives

$$K \leqslant \frac{1}{Z_M{}^2} \cdot \frac{Z_v{}^2}{K_v} \cdot \frac{1}{Z_H{}^2} \cdot \frac{1}{K_I} \cdot \frac{1}{K_{H\beta}} \cdot \frac{1}{Z_\varepsilon{}^2} \cdot \frac{(\sigma_{Hlim})^2}{S_H} \cdot Z_{RH}{}^2 \quad (3)$$

In Lloyd's Register of Shipping published Rules any constants in the coefficients have been merged into an overall constant and with the coefficients in the same order as in equation (3) becomes

$$K = \text{constant} \cdot \frac{Z_v{}^2}{K_v} \cdot K_1 \cdot K_2 \cdot K_3 \cdot B \cdot S^2 \cdot Z_{RH}{}^2 \quad (4)$$

b) *Tooth Strength*

The calculation for tooth strength assumes the load acting at the tip of the tooth and the critical root section to be as shown in Fig. 1 for external gears. For internal gears the tooth form is taken as that in the normal section

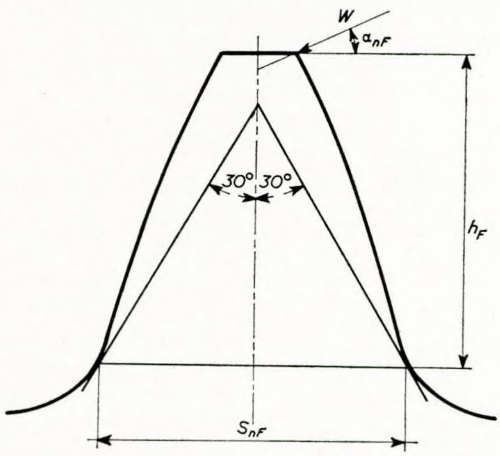

FIG. 1—*Tooth form (normal section) for strength factor*

of the basic rack but with the same tooth depth as the actual gear. In this case $\alpha_{nF} = \alpha_n$.

The stress at the root of the tooth is given by

$$\sigma_F = Y_F Y_\beta Y_\varepsilon K_I K_V K_{F\alpha} K_{F\beta} \frac{W}{m_n} \quad (5)$$

In the I.S.O. formula, the allowable stress is defined as

$$\sigma_{FP} = Y_S K_{XF} \frac{\sigma_{Flim}}{S_F} \quad (6)$$

$Y_\beta = 1 - (\beta/120)$ but not less than 0·75 i.e. β is not to be taken as greater than 30° for calculation purposes.

$$Y_\varepsilon = \frac{1}{\varepsilon_\alpha}$$

$K_{F\alpha}$ and K_{XF} are considered to be unity and the criterion gives

$$W \leqslant \frac{1}{K_I} \cdot \frac{1}{K_{F\beta}} \cdot \frac{Y_S}{Y_F} \cdot \frac{120\varepsilon_\alpha}{120-\beta} \cdot \frac{1}{K_V} \cdot \left(\frac{\sigma_{Flim}}{S_F}\right) m_n \quad (7)$$

In Lloyd's Register's published Rules this is written in the form

$$W = \text{constant} \cdot Y_1 \cdot Y_2 \cdot \frac{Y_3}{Y_4} \cdot \frac{\varepsilon_\alpha}{Y_5} \cdot \frac{1}{K_v} \cdot U \cdot m_n \cdot Z_{RF} \quad (8)$$

The basic gear is assumed to be that in the "as hobbed" condition and the effects of post-hobbing processes and surface finish of "better than average" are considered separately whether for through-hardened materials or surface-hardened gears. The various factors are discussed in this context.

Dynamic Factors (Z_v, K_v)

Since these factors have been generally combined into a single factor based on pitch line speed they are erroneously termed velocity factors. This has led investigators to measure the overall effect, leading to confusing conclusions where an increase in velocity has sometimes been claimed to be beneficial to load carrying capacity and at other times detrimental.

The consideration made by I.S.O. has enabled the two main phenomena to be treated separately. The dynamic load due to such effects as the roughness of the surfaces of the teeth (K_v) will increase the specific loading. This will increase with increasing velocity and will affect both the Hertzian stress and the bending stress at the root of the tooth. The influence of the hydrodynamic oil pressure (Z_v) will augment the capacity of the gear to withstand the Hertzian pressure and may therefore be considered as a factor in the permissible Hertzian stress. It will not be effective in the bending stress. Crook[5] found that, for a given viscosity, the thickness of the oil film is proportional to the oil feeding rate. This will increase with an increase in pitch line velocity.

Experience indicates that the variation of K_v and Z_v is of the form shown in Fig. 2(a). The combination of the two factors for surface pressure in the form ($Z_v{}^2/K_v$) is shown in Fig. 2(b).

Since

$$V_w = \frac{\pi}{60} d_1 \left(\frac{n}{1000}\right) \text{ m/s}$$

where d_1 is in mm,

in the Lloyd's Register notation ($N = (n/1000)$), for the range normally used in marine gearing, the factors can be linearized to

$$\frac{Z_v{}^2}{K_v} = 1 + \frac{dN}{5850} \text{ for surface loading}$$

$$\frac{1}{K_v} = \frac{13000 - dN}{17500} \text{ for bending stress.}$$

Application Factor (K_I)

The type of prime mover and any couplings in the system will affect the mean pressure on the teeth and the distribution across the teeth. This will apply equally to surface loading and bending stress. Table IV gives the values chosen by Lloyd's

TABLE IV—APPLICATION FACTORS

TURBINE PROPULSION GEARS	PRIMARY	SECONDARY
Tandem	1·00	0·88
Dual tandem	0·93	0·81
Tandem articulated	1·10	0·97
Dual tandem articulated	1·02	0·97
SINGLE AND DOUBLE REDUCTION OIL ENGINE PROPULSION GEARS	SINGLE ENGINE DRIVE	MULTI-ENGINE DRIVE
Hydraulic coupling or equivalent on input	1·10	0·97
High-elastic coupling on input	1·00	0·88
Other couplings	0·85	0·74
ALL AUXILIARY GEARS	1·30	

NOTE: A "high-elastic coupling" is one providing sufficient torsional, axial and angular flexibility to the particular installation to minimize the effect of load variations and malalignment on the load sharing of the gear teeth.

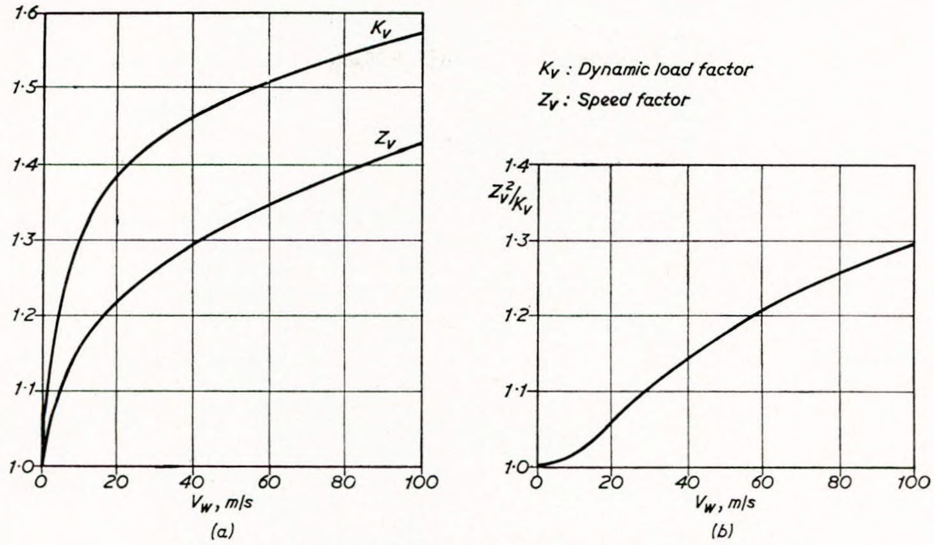

FIG. 2—*Dynamic load factor* (K_v) *and speed factor* (Z_v)

Register of Shipping for the variation in permissible loading, for both the type of gearing and the prime mover, compared with that appropriate to the primary mesh of tandem turbine gearing.

The values for main propulsion steam turbine gearing have been used with little change since 1956 and experience has shown that they are reasonable.

Oil-engine gearboxes have been series-built and have been used with a variety of engine types. Because of this, reliance has been placed upon the guidance notes for torsional vibration characteristics which require that oil-engine manufacturers limit the torque variations on the gears so that the maximum does not exceed 133 per cent of the full mean transmission torque. Comparison of the earlier numbers of defects in oil-engine and steam turbine installations appeared to make an additional installation factor for torque variations unnecessary.

As pointed out in Part 1 the latest statistics indicate the need to consider the question of load distribution across the teeth in oil-engine installations more closely and application factors have been added as a result. The note under Table IV is intended to indicate that although a short description has been used the coupling is not the sole criterion. The mere fitting of a certain type of coupling between engine and gearbox will not ensure the use of a particular coefficient. The characteristics, axially and radially as well as torsional of the complete installation have to be investigated. Pinnekamp[6] has detailed the problem of attempting to counteract external effects.

From the purely torsional point of view it would be expected that a "high elastic" coupling would be able to accommodate 6° torsional movement or have high damping characteristics.

Load Distribution Factor ($K_{H\beta}$ and $K_{F\beta}$)

Measurements carried out on gears have indicated the load increase due to the non-uniform load distribution across the face width cannot be neglected. Davis[2] gave a peak to mean value of 1·6 for a typical uncorrected gear whilst Harrison and Mudd[7] claim higher values are attainable in designs which may conventionally be thought to be very good.

Formerly it was the practice to calculate the bending and torsional deflection of the pinion for a uniform load across the face width for a rigid body. Provided the maximum tooth opening estimated was not greater than 15 or 25 microns for primaries or secondaries respectively it was considered there was no need for correction and this led to a limitation on face width to diameter ratio without the necessity for an extra supporting bearing. However, the loading could be similar to Fig. 3.

Davis, Harrison and Mudd, Wellauer[8] and Niemann[9] have considered the flexibility of the teeth. With the advent of the computer the shaft twisting flexibility and the Hertzian flexibility of the tooth surfaces can also be included or alternatively the whole concept be investigated by a finite element approach.

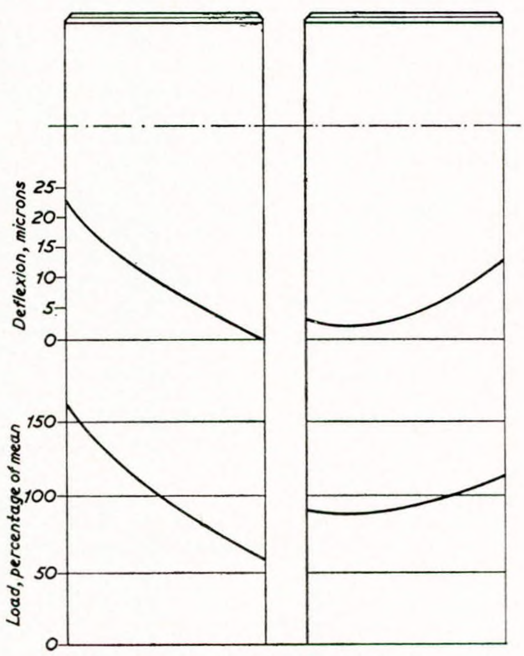

FIG. 3—*Typical load distribution*

As a result of these investigations good average values for the variation of the load distribution factors with face-width to diameter ratio, for both uncorrected and corrected gears, are as shown in Fig. 4. Consideration was given to the case of a pinion meshing with two wheels (dual tandem primary) where, in the old classical method, only torsional deflexion was considered; the estimated tooth opening was smaller than when the load was only on one side of the pinion. However, having regard to the effect of the flexibilities now being considered and the fact that it is difficult to ensure the accurate sharing of the load along the tooth, as instanced by the different patterns of wear on the two wheels found in some cases, it was decided to make no distinction in the factors.

In the Lloyd's Register notation

$$K_3 \text{ or } Y_2 = \frac{1}{K_{H\beta}} \text{ or } \frac{1}{K_{F\beta}}$$

$$= 1 - \frac{1}{16}\left(\frac{B}{d_1}\right)^2 \text{ for uncorrected gears,}$$

$$= 1 - \frac{1}{90}\left(\frac{B}{d_1}\right)^3 \text{ for corrected gears.}$$

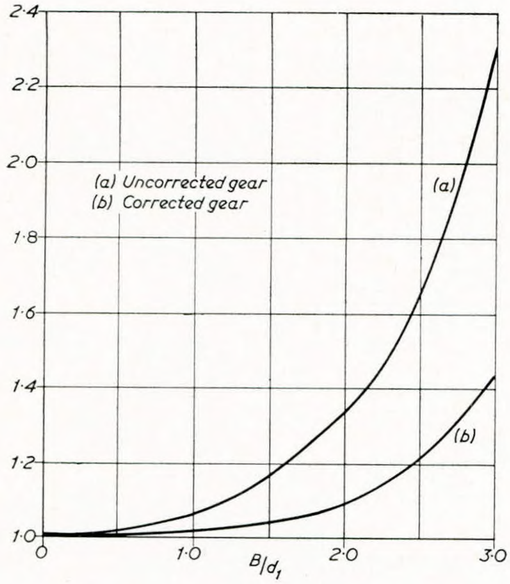

FIG. 4—*Load distribution factor*

A corrected gear is defined as that for which appropriate measures to counteract pinion deflexion have been taken. The normal method used is helix correction. This means that the combined deflexion is calculated and corrective shaving provided along the face width so that, theoretically, under load all the face width will take its fair share. In view of the difficulty in following the exact deflexion curve it is assumed in some instances to be parabolic. However, it is still very difficult to measure the corrective amounts exactly. For this reason, some manufacturers have machined with a linear variation to a proportion of the maximum deflexion on each helix in the manner shown in Fig. 5 for example. Then, if necessary they

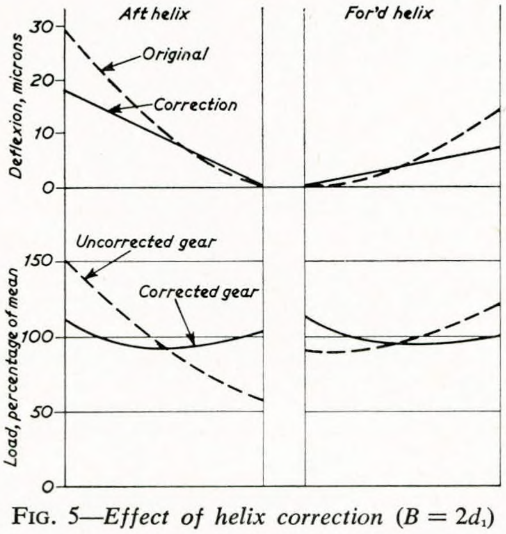

FIG. 5—*Effect of helix correction* ($B = 2d_1$)

have been able to deliberately misalign the pinion by adjustments at the bearings to obtain the required load distribution. This has the advantage that the calculation need not be exact and therefore can be simplified and there is less danger of removing too much metal during machining.

The essential purpose is to obtain a good result for tooth contact in operation and whether correction is required can only be a matter of experience. Fig. 5 illustrates the aim for loaded conditions. To wait until the results of trials may be too late.

Contact Ratio Factor (Z_ε)

For both helical and spur gears, the I.S.O. formula for surface loading considers the load acting at the reference cylinder. The contact ratio factor has been published in DIN3990 for some years and can be written

$$Z_\varepsilon^2 = \frac{\varepsilon_\alpha(4-\varepsilon_\alpha)(1-\varepsilon_\beta)+3\varepsilon_\beta}{3\varepsilon_\alpha} \text{ for } \varepsilon_\beta < 1 \qquad (9)$$

$$Z_\varepsilon^2 = \frac{1}{\varepsilon_\alpha} \text{ for } \varepsilon_\beta \geqslant 1 \qquad (10)$$

For the same Hertzian pressure the load is proportional to $(1/Z_\varepsilon^2)$, all other components of equation (1) being equal. Thus the ratio of the contact ratio effect on the load for spur gears ($\varepsilon_\beta = 0$) compared with helical gears is

$$\frac{3}{\varepsilon_\alpha(4-\varepsilon_\alpha)}$$

This is curved, equal to unity at $\varepsilon_\alpha = 1$ or 3, with a minimum value at $\varepsilon_\alpha = 2$. I.S.O. allows the formula to be used for values up to $\varepsilon_\alpha = 2{\cdot}5$ which covers the normal range of gears viz: $\varepsilon_\alpha = 1{\cdot}25$ to $2{\cdot}3$. The latter value has been used by manufacturers for spur gears, i.e. a large transverse contact ratio and high tooth form, for optimum meshing conditions. However, Fig. 6 shows that the contact ratio factor allows it some 77 per cent of the equivalent helical gear whereas the stub tooth with a low transverse contact ratio may be allowed up to 87 per cent.

Archer(10) examined the relationship for the load on the spur gear at the commencement of single point contact, giving the highest surface stress for a given spur tooth passing through the meshing zone, and the helical gear at the reference circle. The helical equivalent of the spur gear was taken as that having an equal number of teeth on the same reference circle and with the same transverse pressure angle. This favoured

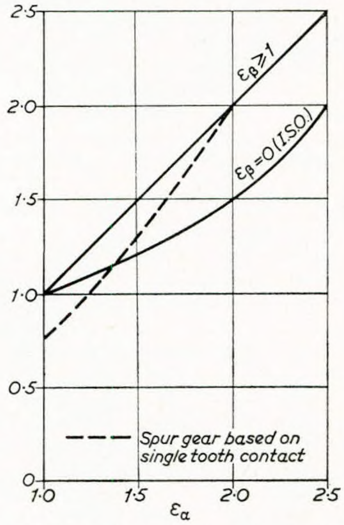

FIG. 6—*Effect of contact radio factor* $1/Z_\varepsilon^2$

the spur gear since, in the practical case, the normal pitch and pressure angle of the two gears would be equal. It was assumed that the non-uniformity of line contact pressure in the helical gear was cancelled out by the dynamic increment in the spur gear.

For equal Hertzian stress, the permissible loading on the spur gear was shown to be approximately 60 per cent of that on the helical gear for a transverse pressure angle of 22° 48′ for both gears and a helical angle of 30°. For the number of teeth in the pinion below 30, which would apply to the small oil-engine gears, the ratio dropped off rapidly. The zone factor in the I.S.O. formula will account for approximately half of the derating of the spur i.e. to about 80 per cent of that of the equivalent helical gear. In order to account for the rest of the derating by means of the contact ratio factor a more appropriate form of equation (9) would be

$$Z_\varepsilon^2 = \frac{(5-\varepsilon_\alpha)(1-\varepsilon_\beta)+3\varepsilon_\beta}{3\varepsilon_\alpha} \ngtr \frac{1}{\varepsilon_\alpha}$$

This is shown, for $\varepsilon_\beta = 0$, by the dotted line in Fig. 6. This would make little difference to the normal spur gear even where single tooth contact will definitely occur whilst for transverse contact ratios in excess of 2 the former consideration that they could be related directly to helical gears would still apply.

In the interest of unity and to maintain alignment with the present I.S.O. formulae, equation (9) and (10) have been used in the Rules. Representations have been made in an attempt to obtain the required change in I.S.O.

Tooth Strength Factor (Y_F)

The tooth strength factor in the I.S.O. concept is determined in the normal section, with the critical section as shown in Fig. 1. For gears having a standard basic rack the strength factor is represented in a diagram dependent on the virtual number of teeth and the addendum modification.

$$Y_F = \frac{6h_F \cos \alpha n_F}{S_{nF}^2 \cos \alpha_n} m_n \qquad (11)$$

The standard I.S.O. tooth form has a total depth of $2{\cdot}25\ m_n$; and a fillet radius of $0{\cdot}25\ m_n$;. For this radius, the stress concentration factor Y_s is considered to be unity in the formula.

Over the normal range of fillet radii used in gears it is sufficient to consider a linear variation and in the Lloyd's Register Rules this has been incorporated in Y_4 from the formula

$$Y_s = \frac{p_n - 2r_f}{0{\cdot}84 p_n} = \frac{m_n - 0{\cdot}637 r_f}{0{\cdot}84 m_n}$$

The tooth strength factor given by equation (11) may be obtained by drawing out the section. However, for use with a computer, it is preferable to have a mathematical formula, if possible. Fig. 7 gives the variation of Y_F for the standard

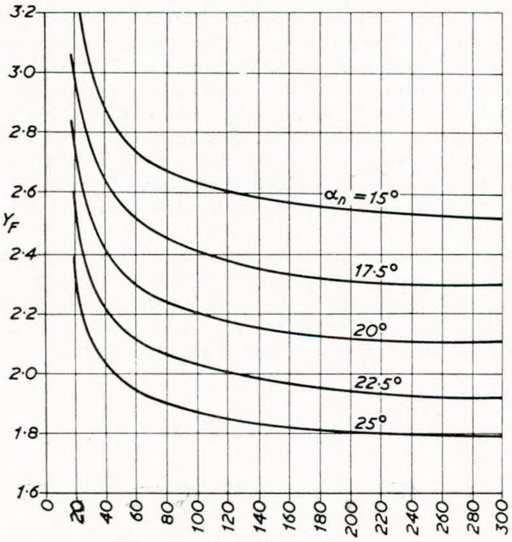

FIG. 7—*Strength factor* (Y_F) *for standard tooth forms*

tooth form with respect to the virtual number of teeth (z_n) in the gear considered. A good approximation for marine gears is given by

$$Y_F = \frac{1}{15}(51 - \alpha_n)\left(1 + \frac{22}{3z_n}\right)$$

In order to accommodate the stub tooth and the high tooth form, Y_F has been assumed proportional to the total depth of the tooth.

The strength of the tooth can be modified by cutting the teeth with the datum line of the basic rack profile not coincident with the reference circle of the gear. This is termed addendum modification and the distance between the datum line and reference circle is given by xm, where x is the addendum modification coefficient. When the datum line is moved away from the centre of the external gear (x positive) this will give a greater tooth root thickness. It should be remembered that where addendum modification is contemplated it must be considered in relation to the gear pair in order to obtain proper meshing conditions. The modification may be different for each gear of the pair but it must be related to the average required. Maag[11] gives advice on determining the appropriate modification. Positive addendum modification in gears having less than 15 teeth will tend towards an excessively pointed form and for a greater number of teeth, too great a negative value may lead to undercutting. Fig. 8 shows the effect on Y_F for teeth with a normal pressure angle of 20°. Similar effects are obtained for other pressure angles.

Taking all the factors together

$$Y_F = \frac{1}{10{\cdot}7}(51 - \alpha_n)\left(1 + \frac{22}{3z_n}\right)\frac{h}{p_n}\sqrt{\frac{J}{J+x}}$$

where
$$J = \frac{z_n + 13}{31{\cdot}3}$$

If the tooth strength factor is obtained by drawing, it can be used in the Rules by the substitution

$$\frac{Y_3}{Y_4} = \frac{h}{10{\cdot}7\, Y_F(p_n - 2r_f)}$$

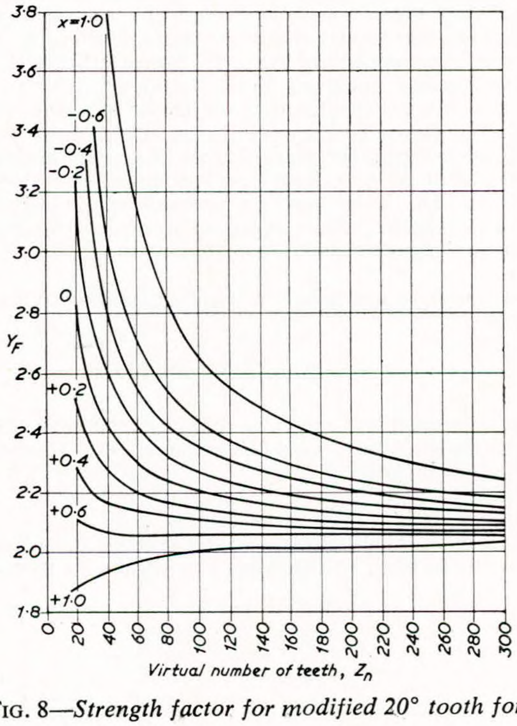

FIG. 8—*Strength factor for modified 20° tooth form*

Permissible Stress (σ_{Hlim} and σ_{FP})

For the basic formula it remains to determine the allowable stress for both flank pressure and bending stress with suitable factors of safety. For bending stress, both pinion and wheel need consideration and the permissible stress can be related to the tensile strength of each.

Previously, the surface stress has been related to the pinion material or to an "augmented" tensile strength of the wheel material whichever is the lower. The augmentation was based upon the gear ratio, following an analysis of BSS 436 by Merritt[12] showing that the same effect occurs in the pinion and wheel when

$$\frac{\sigma_{u1}}{\sigma_{u2}} = u^{0{\cdot}2}$$

In the present Rules, in accordance with the I.S.O. formula, the lower tensile strength has been used for the fatigue strength for Hertzian stress without the factor for gear ratio. For through-hardened materials in both pinion and wheel, the value chosen is

$$\sigma_{Hlim} = \frac{1}{6}(\sigma_u + 1670) \text{ MN/m}^2$$

where σ_u = lower value of the specified minimum tensil strength of pinion or wheel material, in MN/m^2.

Tests have indicated that for surface hardened pinions meshing with through-hardened wheels the pitting resistance of the wheel is increased, particularly with the lower tensile strengths. This has been reflected in the Rules where the chosen value is.

$$\sigma_{Hlim} = \frac{1}{6}(0{\cdot}9\sigma_u + 2380) \text{ MN/m}^2$$

This gives an increase over the case where both gears are through-hardened of 25 per cent for σ_u = 700 MN/m^2 and 18 per cent for σ_u = 1300 MN/m^2.

For carburized and ground gears the permissible stress is 1·67 times the Rockwell C hardness; suitable values based on experience have been chosen for nitrided and induction hardened gears. Induction hardened gears have been given the same rating as gas-nitrided gears in view of the difficulty of achieving the best results unless great control is exercised during the procedure for the former.

Since the I.S.O. concept for surface stress is based upon the Hertz criterion it is logical to follow that criterion for the position of the maximum shear stress and hence the depth of case required for surface hardened gears. Previously, when considering the variation of the stress beneath the surface for carburized gears, designers have considered it logical to specify the case depth such that the factor of safety for the case material should be the same as that for the core material. This will be too great for nitrided gears but, nevertheless, it is considered that the maximum stress should not occur within the core material. The Rules have defined the minimum case depth for such gears so that the maximum stress will occur at a position not greater than 87 per cent of the depth to core hardness.

For bending stress where the gear is of through-hardened carbon steel

$$\sigma_{FP} = \frac{1}{22}(\sigma_u + 1275) \text{ MN/m}^2$$

and for carburized and ground gears the permissible stress is 0·27 times the Rockwell C hardness of the case with appropriate values for other materials.

Shrunk Gear Rims

The foregoing formulae are all concerned with the loading imposed on the teeth by the drive. For shrunk-on rims there will be an additional stress at the root of the teeth due to the hoop stress imposed by the shrinkage.

For this, the basic shrunk gear chosen was a carbon steel rim with a cast iron centre and a shrinkage allowance of 0·075 per cent of the shrinkage surface diameter, since most experience had been gained with such a gear. For the normal thickness of such a rim it was assumed that $d_o = 1{\cdot}05\ d_s$ and a single helical gear was taken since for this the load will extend over the surface whereas for a double helical gear it will be less by the extent of the gap. With an assumed coefficient of friction of 0·15 and, for a rim without the aid of dowels, the factor of safety of the shrinkage against slipping due to the torque was 10. This factor was the largest offered when such gears were used.

Comparison of the hoop stress at the root of the teeth due to the shrinkage and the bending stress, including the stress concentration factor, due to the I.S.O. formula, indicated that, for the same factor of safety, the loading for the shrunk rim should be limited to 80 per cent of that appropriate for a solid gear of the same carbon steel material with a minimum ultimate tensile strength of 480 MN/mm^2. For higher grade materials it is reasonable to relate the effect of the hoop stress to the permissible stress at the root of the teeth in the I.S.O. formula and this has been allowed in the Rules.

The shrinkage of the gear on to the shaft is not considered since the "effective rim thickness" will be so much greater than the aforementioned proportion and the hoop stress at the root of the teeth negligible. Carburized gears with shrunk-on rims will approach 100 per cent of the permissible load for solid gears. Those designers who do not desire to use the formula which includes a small amount of iteration in the initial stages since it is related to the proposed tooth loading may use the factor of 80 per cent to determine the Rule loading.

Roughness Factor (Z_R)

In the Rules prior to 1973, the permissible loadings for surface-hardened gears were based upon the subjection of such gears to an approved post-hobbing process. Where through-hardened teeth in the pinion and wheel were subjected to such a process, or where it could be demonstrated in advance that the finish and profile of the teeth was equivalent, an allowance was given. In order to emphasize the importance of surface finish in all cases the allowance has been separated from the metallurgical process. This enables one to determine the allowance for both surface loading and bending stress without total reliance upon processes which may show variations in results.

An exact correlation between surface roughness and the level of stress experienced by two bodies in contact has not been obtained. Experience, which for surface loading will also include the effect of differing oil film thickness, indicates that empirical values as shown in Fig. 9 are appropriate. It should

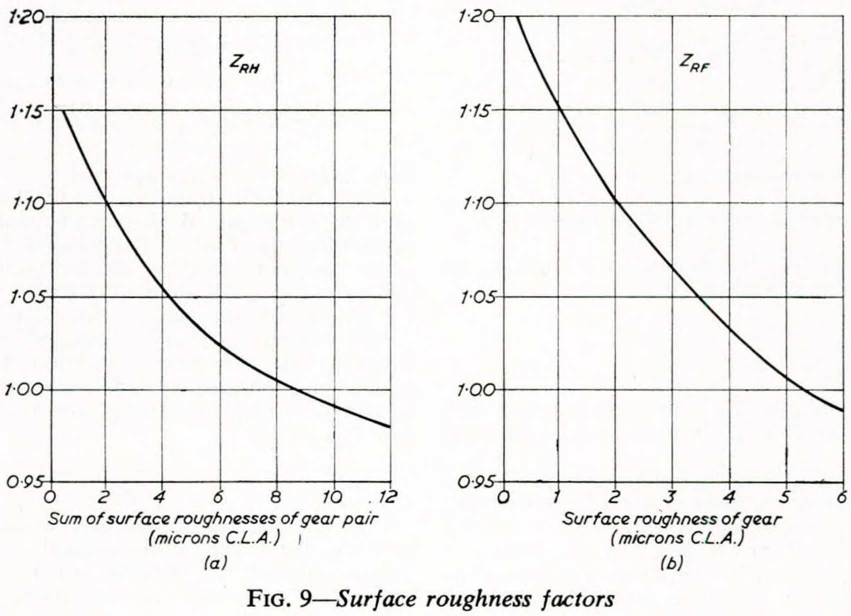

FIG. 9—*Surface roughness factors*

be noted that the surface loading is proportional to Z_{RH}^2 and is related to the sum of surface roughness of the two gears in mesh. Dawson[13] has shown that the ratio of film thickness to this sum affects the failure rate. Z_{RF} is related to the surface roughness of each gear.

Even with the same type of cutting machine significant variations in surface roughness may occur which may be reduced by determining the most suitable conditions through experience. Until such time as the overall factor can be separated into its component parts with confidence, Fig. 9 may be used as guidance for the allowances given for surface finish in the Rules.

Ice Navigation

As well as normal operation in ocean-going conditions, the strength of gearing has to be increased for vessels desiring to trade in ice. At present two authorities have published regulations for vessels trading in their waters in winter. Unfortunately these are based upon different considerations. The Finnish/Swedish regulations for service in the Northern Baltic are concerned with the efficient use of their ice breakers. On the other hand the Canadian government are concerned with the prevention of pollution in Arctic waters which may result from the stranding of the ship. In an attempt to ensure the ship is able to maintain continuous progress the Canadian requirements for gearing are based upon an increase in torque for the different classes in a similar manner to the old Classification Rules. These make it simple for the manufacturer to design the gearing to the service requirements.

The Northern Baltic requirements are based on impactive forces and sudden stopping of the machinery due to the propeller hitting the ice. This requires a knowledge of the inertias of the components of the complete installation so that the effective increase in torque may be calculated. This makes it impossible for manufacturers to recommend the boxes on the classical basis of power ratings leaving the remainder of the installation to the main contractor. The choice of box cannot be finalized until a very late stage in the negotiations.

Since these are statutory requirements the Classification Society has to apply them for vessels intended for service in these waters and requiring the appropriate Ice Classes.

PART 3

The design of the gear may be carried out by the computer, even to the extent of plotting the required tooth form and a finite element analysis for the stress pattern if desired. However, this must be related to a basic gear achievable during manufacture and the possible service conditions as a basis for installation. Here, the human element assumes greater importance and the opinions become more subjective. Strictly, for the purposes of Classification, apart from the design appraisal for loading and the absence of excessive noise etc. during running, it is simply a matter of obtaining good contact marking on sea trials. One would need to be supremely confident to wait until that stage without an efficient quality assurance procedure from the commencement of manufacture. Compliance with Classification Rules often forms part of a contract but it is prudent for a client to include the standards he requires in excess of such minima.

Causes of Pitting and Fracture

Pitting may be caused by one outstanding error or may be a combination of many reasons. Amongst the more usual are:

a) excessive tip relief with pitting occurring in the dedendum;
b) unsuitable correction along the face width, the pitting being in the vicinity of such correction;
c) undulations and unsatisfactory surface finish;
d) mismatching of helical angles, pinion to wheel;
e) profile errors and pitch errors;
f) hand bearing usually at the ends of each helix;
g) malalignment;
h) vibration due to pitch errors, excessive backlash or unbalance.

The majority of these apply also to tooth fracture with the additional necessity for obtaining a smooth transition between the profile and the fillet, a good fillet radius and the absence of stress raisers such as machining marks.

Quality Assurance Procedures

For any gear pair it is prudent to lay down the procedures of inspection in the logical order for manufacture and in a manner efficient enough to achieve the required standards of accuracy. This implies inspection of the blanks, the hobbing and post-hobbing machining, the teeth and the assembly conditions. In order to ensure good judgement of the finished product a Classification surveyor needs to make an overall assessment of the inspection procedures and standard of workmanship without the necessity for interfering with the even flow in the workshop. The Classification Society is available to give advice on procedures or standards if requested.

It has been found necessary to insert in the Rules certain requirements during manufacture in a general form. Future experience may demand that these be expanded or standards be laid down in a more formal way. Where the contact marking obtained in the workshop is deemed to be less than satisfactory, records of measurements of pitch errors, undulations, axial pitch errors, tooth thickness and backlash should be available to the surveyor if he desires them to assist his judgement of remedial measures. Where the rating of the gear pair is dependent upon a certain degree of surface finish, records of the finish should also be taken.

For the more highly loaded gears the teeth should be cut under conditions of temperature control with a total temperature variation not exceeding 2°C at least for the finishing cut. The blank should be allowed sufficient time to stabilize to the room temperature before cutting. This may require a waiting time of up to eight to ten hours dependent upon the ambient temperature.

The standards of accuracy normally required for Classification purposes are in general agreement with those set out for the appropriate grades in I.S.O. Recommendation No. 1328, BS 436 Part 2 (1970) or BS 1807, Part 1 (1952) now being revised.

Although profile errors have been included in the causes of pitting, measurements are normally taken only when agreed between the client and the manufacturer. Profiles are difficult to measure accurately and, as well as deviations from the involute profile, records may indicate the related errors of base circle and pressure angle. See Fig. 10 for typical examples.

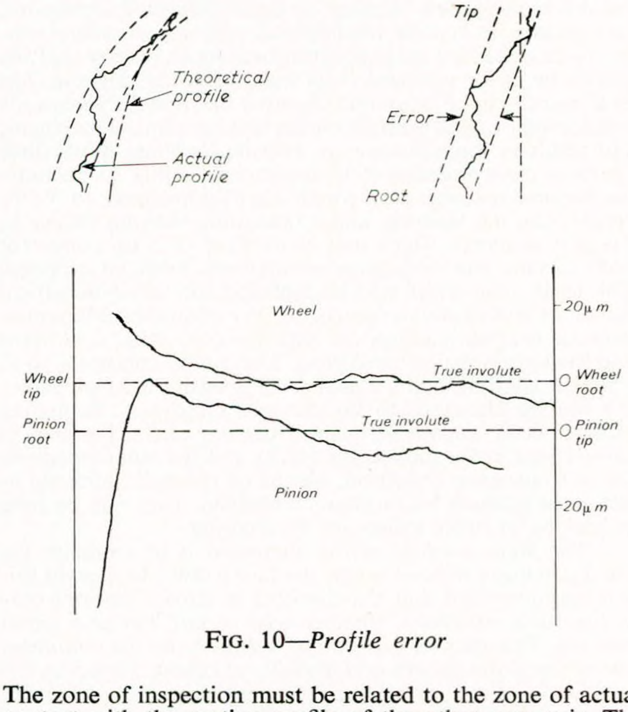

FIG. 10—*Profile error*

The zone of inspection must be related to the zone of actual contact with the mating profile of the other gear pair. The errors in the measuring equipment, particularly for the larger gears, are commensurate with the tolerances of accuracy to be proved. However, despite the problems, it is recommended that measurements be taken on at least a representative number of gears in a box.

The test in the workshop to determine the contact marking on each helix should aim to indicate what is likely to occur when the gears locate in their load positions. Records of such contact marking should be preserved on transparent adhesive tape. This will enable the pattern of contact marking, to be achieved on installation in the ship, to be known.

It is recommended that the accuracy of the cutting machinery be inspected as often as possible. Measurements should be taken at appropriate stages in the manufacturing process as experience dictates in order to obtain consistency in the finished product. Good quality control will reduce the subjectivity of opinions on the reliability and fitness of the product for its purpose. For this reason it must be a continuously improving system based on experience both in the shop and with operation of the finished gears in service. In this connexion some companies having monitored instances of pitting have laid down quality assurance methods resulting in a great improvement in surface finish. Nevertheless, they have also applied copper plating to the teeth to take care of the running-in period. The plating has not completely worn away within a year of service and, it is claimed, has been successful in eliminating pitting.

Environmental Considerations

However sophisticated the mathematics applied to the design of gear teeth and however careful the machining and inspection, consideration must be given to the thermal and external effects.

In former times, with smaller vessels and powers, the machinery installation was relatively flexible compared with the hull structure. Then it was possible to consider the box as sitting on a reasonably solid foundation; with a rigid gearcase, the designer had done what was required. Nowadays, the higher powers and the tendency to place the machinery as far aft as possible has led to a reversal of the relative stiffness. The rigidity of the gearcase and the design of the mountings must be related to the surrounding structure. The mounting, in particular, should allow symmetrical movements of the gearing whatever the movements of the hull support.

These movements of the hull in a seaway can be the subject of a finite element analysis of the aft end and the results used in consideration of alignment procedures. The present trend is to use as few bearings on the line shafting as possible, consistent with specific loading and whirling considerations. In any case the forward supporting bearing on the line shafting should be as far removed from the gearbox as possible. The final gear has to be supported on either side and the necessarily small span for these bearings means that even small movements can result in large changes in bearing loadings. With three bearings close together it is almost impossible to maintain the required loadings at all conditions of ship operation. When considering the loadings under operation, thermal effects on the gear supports, which may move from 0·25 to 0·6mm for both turbine and oil-engine installations, must be included. The tooth load must also be included for non-symmetrical drives. Manufacturers of gearboxes give allowable differentials between bearing loadings for both the cold static conditions and the hot operating conditions. The overall concept is so to align the shafting that the gear shaft does not oscillate across the bearing clearance to tilt the gear relative to its mating pinion. Such alignment, whilst carried out according to calculations in the static condition to give the requirements in the hot operating condition, has to be checked indirectly as far as the gearbox bearings are concerned. This may be done by jacking or strain gauges on the shafting.

The main purpose of the alignment is to minimize the maldistribution of load across the face width. In view of this it is recommended that the checking be carried out not only in the static condition, whether cold or hot, but also whilst running. This may be carried out indirectly by the continued use of the strain gauges on the shaft, of course. However, the techniques of measuring the effects at the mesh by telemetry are now being progressively used. This enables strain gauges of very small dimensions to be placed in the root of the teeth along the face width to measure the strain and hence the stress and by this means the disribution of load. The receiver and transmitter have been miniaturized to approximately the size of a matchbox and thus can be attached to the gear without unduly affecting the balance. The encapsulated batteries have a life of approximately 36 h and with suitable switchgear the equipment may be used for monitoring if required.

With suitable coating of the teeth the meshing contact on sea trials may be checked. Standards normally give the area to be expected for the appropriate grade of gear. However, it is the quality of the marking that is the real criterion and opinion on this is very subjective. Some Firms have used carbon and examined the tapes with a light meter in order to achieve objectivity. These methods suffer from the fact that the contact recorded will be a combination of that at all speeds from low to full load. The use of telemetry gives a direct instantaneous reading and may be determined at all loads.

Care must be taken to ensure that any connexion to the prime mover whether via the shafting or piping is sufficiently flexible to minimize the effect on the mesh of any movements on the prime mover side. With the higher speed and thus lower torque on the input there are many flexible couplings which can take care of the shafting. A word of warning is required here to ensure that the weight of such a coupling does not have an excessive effect upon the gear shaft or, alternatively, that the manufacturer takes it into account in lining up his gearing.

In the future it is likely that couplings of sufficient capacity, or diaphragm plates, may be produced to be fitted on the aft side of the gear box to minimize the effect of movements from the line shafting side.

CONCLUSIONS

The formulae for tooth loading have been framed in such a way that, as experience is gained for various phenomena, more factors may be taken into account. Whilst this may complicate the calculations and one cannot use simple calculations for the design as in the old Classification Rules, this should prove no great problem in these days of the computer.

However exact the calculations and idealized the gearing on paper, greater emphasis must be placed on quality assurance procedures during manufacture. Sufficient measurements should be taken to ensure the greatest accuracy attainable. These will be of great assistance in making an objective assessment of the finished product and serve as guides for replacements etc. if required. Wherever possible, direct measurements on the teeth and of the meshing conditions should be made in order to lessen the reliance that must be put upon opinions which can only be given after great experience. There is scope for new methods which may be used as diagnostic tools and also for improvements in the accuracy of present equipment particularly for the larger gears.

At the present time there is a need for designers to consider the strength of the gears with regard to the complete environment of the installation. However, in the future, there is ample opportunity for attempts to isolate the gear from the effects of components both at the driving and driven sides of the gearbox.

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Discussion

MR. I. T. YOUNG, F.I.Mar.E., said that in their approach to gearing rules Lloyd's Register of Shipping, in the past, had had their feet very much on the ground, and had shown a marked reluctance to follow the example of some of their brother Classification Societies where theory rather than practice had seemed to be the major influence. In the era of the slide-rule rather than the computer or electronic calculator, the designer had often had reason to be thankful when Lloyd's were specified rather than, say, "Brand X".

It had therefore been with something like dismay that the succesive complication of Lloyd's Rules had been witnessed. The 1966 version had been troublesome enough, but in general the loadings permitted had been satisfactorily above normal design standards. When the 1973 revision had taken place, however, it was realized that the age of instinct and simplicity had passed and that henceforth the computer would hold sway.

Mr. Toms, as the major architect of this revolution, had done a remarkable job in adapting the ISO universal formulae to this limited field. Admittedly he had not attempted the daunting task of adapting simultaneously to gears in all fields, but had demonstrated that Lloyd's still had their feet on the ground by trimming the various coefficients to suit the Society's very wide experience. All the same, he wondered if in fact Mr. Toms had not tried too hard in some places to fit commonsense into a doctrinaire strait-jacket. More than once Mr. Toms had admitted that Lloyd's had conformed with the ISO dictates against their own better judgment.

It had been said that the Pharisees in biblical times had been so careful in observing the laws of diet that they would strain out the gnat from their drinking water but could swallow a camel without noticing. He hesitated to compare Mr. Toms with a Pharisee, but was he not in danger of straining out the gnat and swallowing the camel?

For example, the "gnat": the dual tandem articulated turbine primary gear gained by 2 per cent in Table IV over the single engine diesel drive with high elastic coupling at input. However, the "camel" came up on the following page, where it was admitted that the torque on the diesel drive could increase by as much as 33 per cent due to torsional vibration in the running speed range without any penalty from the Rules. Mr. Young would hazard a guess that such defects as there had been in the oil engine gears shown in Table III had in many cases been associated with such vibrations, albeit within the Rules.

Another "gnat". The Rules, in common with ISO, adopted an unnecessarily elaborate approach to transverse contact ratio ε_α, no doubt with the object of attaining precision within a few per cent. Then there was the "camel" where a 20 per cent bonus was allowed for helix correction without specifying precisely what this meant. Very often the theoretical amounts were of the same order as the errors of contact allowable under the highest grade of BS 1807, and implied a precision of measurement not easily attainable. The removal of lacquer from the teeth when running was a poor indication of load distribution, and that shown in Fig. 3 of the paper would probably indicate almost perfect contact at all powers.

The rule that the maximum Hertzian shear stress in nitrided gears should occur at not greater than 87 per cent of the case depth was a considerable barrier to design, especially when a hard-on-soft combination of steels was used for diesel application. The underlying theory, of course, hardly stood up to close examination, as Fig. 11 showed:

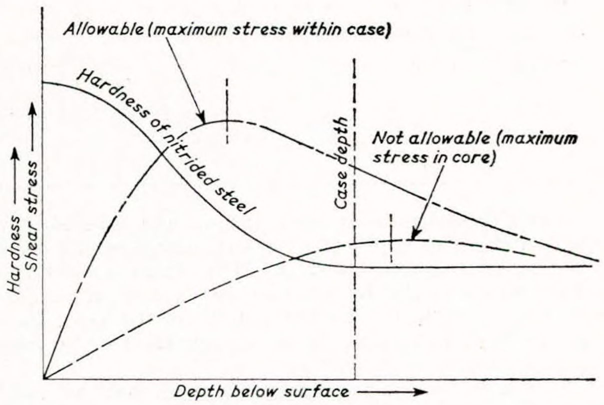

FIG. 11

Experiments with discs had shown considerable improvement in load carrying capacity over through-hardened steels, even when the maximum shear stress was well within the core. Unfortunately, the Rule covered depth of maximum shear irrespective of its value so that, even when the stress itself was relatively low, the gear was reckoned to be inadmissible simply because the peak was below the 85 per cent depth. He hoped that Lloyd's would consider waiving this entirely for the hard-on-soft case.

Mr. Toms admitted the difficulty of profile measurement on large gears and the taking and interpretation did indeed present a considerable problem. Why, then, did Mr. Toms advocate that measurements be taken on a representative number of gears in each box? There was nothing more dangerous in uninformed hands than the profile measurement of one gear of a pair, and he strongly recommended that these records be used for investigation only where meshing contact was unsatisfactory.

He questioned an item not touched on in Mr. Toms' paper but which caused considerable trouble in practice, this was the new Rule regarding checking of meshing contact in the case. Oil clearance must be taken up and the load must be sufficient to overcome pinion weight. In addition, the gears must be in their load positions within the bearing.

The most practicable method of meshing was to have the gear in the bottom of its bearing, and the properties of an involute showed that parallel movements in the two bearings would not affect the mesh. Bearing clearances could be measured and corrected for. Why, then, this unnecessary complication which, in his opinion, only led to further error?

Nevertheless he thought everyone would agree that the new Lloyd's Rules represented a major step forward and approached much closer than any others to reasonable design standards for gears in service.

MR. P. E. LARSSON said the application factors referred to in the Rules at present considered the various gearing configurations, coupling arrangements, etc. The intention of the ISO load factor was to describe the loading resulting from driven and driving components. Mr. Toms had distinguished in his paper between engines mounted fore and aft, and referred to vibration torques for oil engine gears up to 133 per cent. This led to a question. Had the author observed any particular features that could result in an application factor for future Rules which also reflected the type of ship or service which might be characteristic for the installation.

TABLE V—APPLICATION FACTOR K_I

	Load increase per cent	
	Steam turbine	Diesel
Propeller pulses (axial, torsional)	6 to 15	3 to 8
Hull fouling	2 to 5	—
Shallow water/rough weather	5 to 10	—
Deviation from nominal load	5 to 8	—
Engine pulsations	—	15 to 30
	18 to 38	18 to 38
	K_I = 1·18 to 1·38	

For a comparison—a steam turbine and a diesel engine—he had made an appraisal of external loads which needed to be considered (see Table V). This listed a number of factors which might be relevant for a tanker. Different magnitude of influence on the gear from the same source could be explained by the plant configuration or by engine characteristics.

For a turbine drive the LP turbine had an inertia which was considerably larger than the inertia of the propeller. The inertia of a medium speed engine was of the same order of magnitude as the propeller, Fig. 12. Therefore, the distribution and size of the inertias resulted in about twice as high load increases on the turbine gear as on the oil engine gear, from impulses at the propeller as shown in Table V.

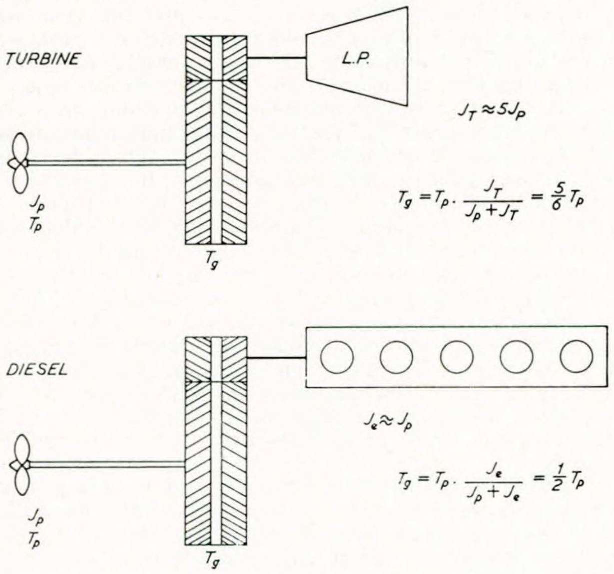

FIG. 12

The different torque characteristic of the turbine and diesel gave different contributions to the load factor from hull growth, shallow water operation and rough weather, since the turbine had constant power and the diesel had constant torque. One had to expect deviations from nominal power in a steam plant if, for example, a bleed was closed or pass-in steam was excessive. Engine torque might vary and had to be accounted for.

Another difference—not accounted for in Table I—was that the turbines normally ran full power, but the diesel ran at normal cruising power which was lower than the m.c.r.

He had raised this question since he believed the Classification Society to be in an unique position to have access to the information needed. One could never expect ISO to present the application factors for the marine field of engineering—these must come from their own experience.

As a natural consequence of strain gauge measurements it was also necessary to understand and evaluate the actual stresses which occurred. At present nominal figures were referred to and did not define how the space between the nominal figure and the material limit was utilized.

DR. SIMON ARCHER, F.I.Mar.E., said it was now some 13 years since Lloyd's Register had published a technical paper on marine gearing, and thus Mr. Toms' contribution was both timely and welcome.

He said he had been brought up to believe, rightly or wrongly, that the Rules of the Classification Societies should be confined to those matters for which service experience had demonstrated the need for some measure of regulation; furthermore, that in formulating Rules, simplicity should be aimed at whereby the maximum safe "elbow room" should be allowed the designer. This meant that Rule formulae should include only those factors or parameters which were not only relevant but could also be reasonably confidently quantifiable. It was for these reasons that Lloyd's gearing rules had developed step by step as they had.

Starting post World War II, the K factor had been aimed at minimizing pitting, at that time probably the most prevalent gearing trouble. A study carried out in 1962 (Mr. Tom's reference [10]) had shown that surface loadings for helical gears, when limited by the simplified K factor formulae, would give an accuracy in calculated Hertz stress of within 3 per cent for turbine gears and within 5 per cent for oil engine gears with their lower reduction ratios, based on teeth with a standard ratio of working depth, h, to normal pitch of $2/\pi$. Although the study confirmed that both pressure angle and helix angle had negligible effect on Hertz stress for a given loading, it did suggest a possible case for incorporating a tooth form factor in the formula, thus:

$$W = K\left(\frac{\pi}{2}\cdot\frac{h}{pu}\right)\left(\frac{u+1}{u}\right)\underline{d}$$

However, at that time it had been judged, and he thought rightly, that having regard to the undoubtedly very much greater uncertainties on the effects of such factors as pitch line speed, dynamic loadings, pitch errors, surface finish, lubrication, pitting resistance of materials, etc, there was no clear justification for further complicating the Rule.

In 1956 accommodation had been made for a wider range of gearing steels with allowances for higher UTS to assist in coping with the heavier loadings then increasingly being proposed. In some cases these higher contact loadings contributed to tooth fractures; in other designs surface-hardened gears had been adopted. The 1966 Rule revisions had taken note of this experience and, for the first time, legislated for tooth bending stress and also gave load factors for surface-hardened gears.

The latest Rule revision of 1973 incorporated the 1967 agreed basic ISO formulae, which represented a brave attempt to legislate for the design of all shapes and sizes and service applications of involute gears and, for completeness, embodied a long series of factors, whether currently quantifiable or not. Lloyd's had wisely tempered some of the factors in the light of well-established marine experience (Table IV, for example). Others they had omitted, in effect, by equating to unity, e.g. lubrication, pitch, errors, size factors; and still others had been quantified, purely empirically, pending more reliable data, e.g.

surface roughness, speed and dynamic factors. This seemed a balanced and practical approach to the very real problem of digesting the ISO formulae and, of course, emphasized the truth that no computer, however sophisticated, could conjure up numerical values for design factors or parameters; only experiment and experience could do that. It was to be hoped that the potentially greater discrimination in assessing gear designs inherent in the ISO type formulae would indeed be reflected in further improved reliability, in the future, and/or reduced weight and cost of gears.

It would be particularly interesting if the author could indicate in a general way the kind of impact which the new ISO-based formulae had had in their application to current designs. For example, were allowable contact loadings significantly different from the pre-1973 values, and had minimum allowable tooth pitches changed appreciably on average, either up or down?

Turning to matters of detail in the paper, Dr. Archer said his first point concerned the statistics of gearing defects in Tables II and III. These were very interesting and by and large indicated improved reliability over the periods concerned. However, as would more readily be seen from a diagram (Fig. 13), periods A, B and C were not strictly comparable statistically, in that not only were the longest periods at risk different, i.e. 14, 19 and 13 years respectively, but so also were the longest periods at risk before defect data began to be collected, i.e. 9, 14 and 6 years respectively.

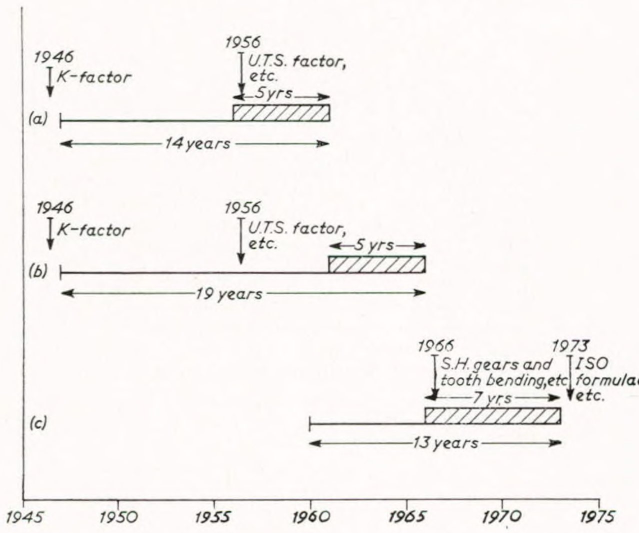

FIG. 13

Nevertheless, it would be fair to draw the following general conclusions:

a) For turbine gears, second reduction defects in all three periods represented 65 to 75 per cent of the total, and, as the author pointed out, the bulk of these were in aft end installations. This would tend to confirm his conclusion that this location exposed the secondary gears to greater external loadings than when installed amidships, and pointed strongly to the desirability of developing suitable means for isolating the gears more effectively from the influence of the propeller shafting, for example, flexible couplings, etc.

b) For oil engine gears, which had only been taken for periods B and C, the overall improvement was less marked, with again the bulk of the defects occurring in aft end installations. Since some oil engine gear designs were of the step-down/step-up type, often with co-axial input and output shafts, might it not be preferable in Tables III and IV to substitute "Tandem" for "Double reduction" in the column headings?

With regard to the author's Fig. 2, showing empirical curves for dynamic load factor (K_v) and speed factor (Z_v), also the curve for Z_v^2/K_v, the author's linearized equations (2) to (8) did indeed show acceptable deviations of not more than 2 per cent over the pitch line speed range (V_w) from 20 to 100 m/s, which covered most marine applications.

The "Application factors" (K_1) had surely been tabulated as ($1/K_1$) in Table IV. Should this not be made clear?

When discussing elastic couplings for oil engine installations, the author had stated that the oil engine manufacturers should limit the torque variations on the gears so that the maximum did not exceed 133 per cent of the full mean transmission torque. Would it perhaps be clearer to use for example the term "peak total torque" rather than "maximum"? Also, was the 6° torsional movement total range or $\pm$?

In Fig. 4 it would be clearer if either the ordinates or the title were indicated as ($K_{H\beta}$) or ($K_{F\beta}$).

In Fig. 5 the torque input to the pinion was assumed to be on the after helix, which was unusual, but this was important information which was not given.

On the question of contact ratio factor, it would seem that the DIN formula, equation (9) did indeed leave something to be desired, particularly with regard to its application to spur gears. Here there seemed to be an illogicality in that, as the author had pointed out, the formula discriminated unfavourably against the low pressure angle, high contact ratio designs compared with low contact ratio stub tooth gears. The author's proposed modified equation (9) would appear to be an improvement in this respect and also gave closer agreement with the results of the work in Ref. [10] on spur gears with single tooth contact, when taken in conjunction with the zone factor, Z_H. The reason that in Ref. [10] the spur gear equivalent of the helical gear was, for purposes of theoretical comparison, taken as that having an equal number of teeth on the same reference circles, and with the same transverse pressure angle, was justified in that these assumptions were the only possible ones, given a common input torque, face width, gear ratio and centre distance.

It was noted that the standard ISO basic rack tooth form had a fillet radius of 0·25 of the normal module. This represented a reduction of one third from the L.R. pre-1973 minimum value for the actual gears. Could the author indicate how this compared, say, on an average marine turbine second reduction pinion and main wheel respectively? Further, could the stress concentration factor (Y_S) really be justified as unity for such a small fillet radius. Fig. 14 showed the disastrous effects which could ensue from too small a fillet radii. It was a cross-section of a manganese silicon pinion which had fractured from practically every tooth from the fillet radii. It was really a glorious example of how stress concentrations could contribute to disaster.

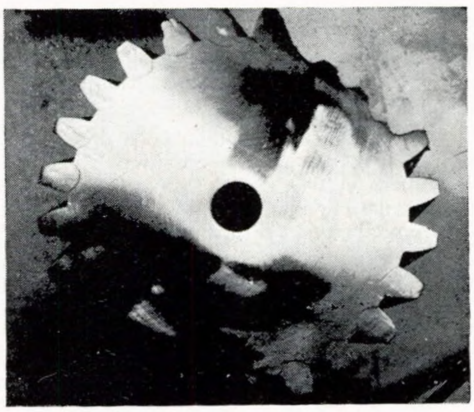

FIG. 14

For carburized and ground gears it was concluded that the units for permissible stress were the given multiple or fraction of the case UTS corresponding to the Rockwell C hardness.

Finally, he congratulated the author on a most useful and timely paper.

MR. A. E. WOLSTENCROFT said this paper was of great interest to all engineers concerned with gear transmissions. Mr. Toms had access to records which were not available to most people. The number of failures were of interest. The wide range of individual designs and the low actual number of failures meant that the figures were probably not amenable to statistical analysis except on a simple "Trouble"/"No Trouble" basis. It might be valuable if Mr. Toms were to comment on any correlations there were between rim and tooth fracture, rim slipping, manner of construction and material. Perhaps he could say how many different designs were involved rather than the simple number of failures.

Gear failures usually occurred because there was a prime cause external of the normal considerations of nominal tooth loading, particularly excess of end loading perhaps due to faulty articulation or inadequate analysis of gear case structural stiffness.

With regard to the Rules themselves, Mr. Wolstencraft's company found themselves very concerned with what they considered to be an unnecessary restriction. (Mr. Young had already mentioned this.) The question specifically referred to was the position of the maximum Hertzian shear stress in relation to case depth for surface hardened gears. In the paper Mr. Toms said: "Since the ISO concept for surface stress is based upon the Hertz criterion it is logical to follow that criterion for the position of the maximum shear stress and hence the depth of case required for surface hardened gears." This was one giant stride, and he was not convinced that the stride was necessary. There was a further giant stride when Mr. Toms said: "This will be too great for nitrided gears but, nevertheless, it is considered that the maximum stress should not occur within the core material." The large nitrided gear was thus effectively rejected, and this was a serious matter as high torque gearing was increasingly of the surface hardened type and manufacturing problems were minimized by nitriding. Smaller nitrided gears at reasonable loadings would generally satisfy the new Lloyd's criteria.

As an example, Mr. Wolstencroft's company had made a study of an hypothetical large single engine diesel gear of 1250 mm centre distance and 16 module pitch with a speed ratio of 470 to 184 rev/min, such as might be used on one of the new 1000 or 1500 horsepower (745 or 1100 kW) per cylinder diesels now coming on to the market. They had assumed a highly flexible coupling at the input. It had all been put through the computer using the methods recommended by Mr. Mudd, see Ref. [7] of the paper. These procedures had been developed over a number of years. They were very mathematical. The programmes were very well established and time and again they had been subjected to the most rigorous re-examination, and so far as could be judged, with all the information at their disposal, his company were at the stage where they could actually correlate analysis with the relatively few known failures which existed.

In the first place, the permissible K by Lloyd's Rules (basic) was 420 (British units were used because some of the programmes were not yet converted), the accuracy allowance permissible K would be 560; the total case depth requirement by the new Lloyd's Rule would be ·0345 inch (for 420K) which the company had in mind and with which the total depth case was unlikely to exceed 0·030 inches, could not be met with a high alloy non-aluminium steel.

He considered this to be a very good gear, although the company would not use loads as high as the maximum Lloyd's K value of 560.

Mr. Wolstencroft gave a very brief description of the methods used.

Fig. 15 showed a gear tooth under load.

The stress variation at a particular point was illustrated. Initially at any chosen point there was Hertzian stress followed by bending stress as the point of contact travelled further up the tooth. Residual stress moved the zero position.

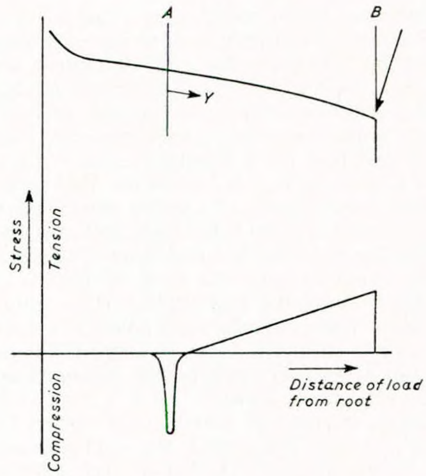

FIG. 15—Cycle of stress in direction Y at point A

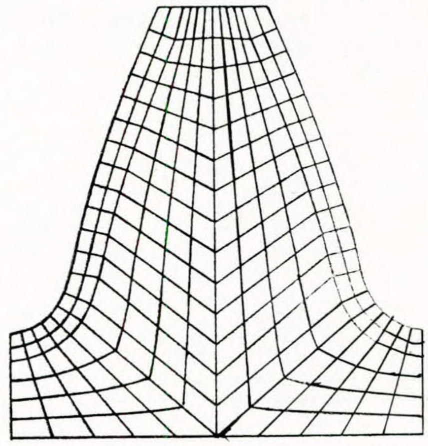

FIG. 16—Finite element idealization of gear tooth

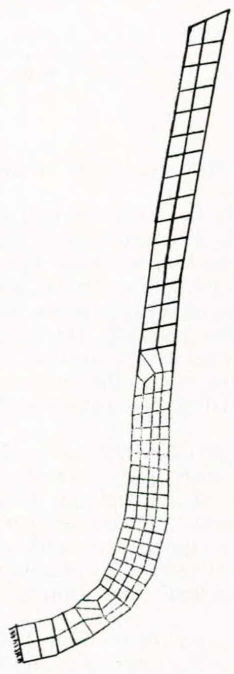

FIG. 17—Idealization of loaded flank

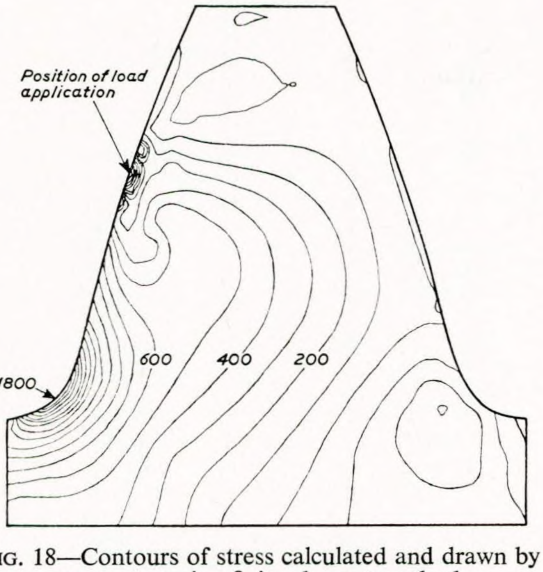

FIG. 18—Contours of stress calculated and drawn by computer using finite element method

Fig. 16 showed a finite element idealization of the tooth. Essentially the computer did the vast amount of arithmetic necessary to solve all the stress equations.

Fig. 17 showed a finite element idealization in the region of the tooth surface.

Fig. 18 showed a typical computer plot of the resulting stress pattern.

Fig. 19 showed a typical print out of the calculated results tabulated according to distance from the tip and distance below the surface.

The basic procedure was to find the theoretical failure point for prolonged operation and the loads and stresses at which the failure occurred. All other points under consideration were then ratioed to this and the results printed out as ratios of permissible stress to actual stress.

Any chosen tooth form or pattern of hardness and residual stress could be incorporated.

After years of work this procedure had been reduced to a gear rating formula and the company would rate their hypothetical gear at a K value of the same order of magnitude as the Lloyd's Rule for an accurate gear on a basis one failure in one hundred taking everything into consideration except application factor. To be rid of this one failure the load would be reduced and finally divided again by a marine application factor: to settle the value of this was difficult but the 133 per cent guidance figure for torsional vibration mentioned by Mr. Toms gave an order of magnitude.

All in all the company would not be very brave about K factor as they were very mindful of their responsibilities. On the other hand, at the K factors which would be used, the Hertzian shear stress would certainly not cause failure.

They were not alone in this view. At the Ninth Round Table Conference on Marine Gearing Mr. I. T. Young had presented his paper "Nitrided Teeth—the Shear Stress Myth" and had drawn the conclusion that the following points were adequately established:

1) Load-carrying capacity was reduced as the depth of maximum Hertzian shear increased relative to case depth but these were not necessarily cause and effect;
2) This reduction was gradual, and the points of total or effective case depth had no special significance;
3) Even when the maximum shear stress occurred well within the core, the hardened surface imparted substantial protection relative to an unhardened steel.

Mr. Young had then drawn the further conclusions, stating that for these reasons it might be appropriate if Classification Societies and Standards Authorities, rather

GEARS HTZ EXAMPLE

33.000 CRS 100 TEETH (30 IN MATE) 2.000 DPN 20.0 DEG NPA 9.983 DEG HELIX ANGLE

51.488TIP DIA. 192991. ALLOWABLE HERTZIAN STRESS AT P.L. MATERIAL EN40C 60T .025CASE
14621. ALLOWABLE TOOTH LOAD

DIST. FROM TIP

DEPTH	0.002	0.077	0.152	0.227	0.302	0.377	0.452	0.527	0.602	0.677	0.752	0.827	0.902	0.977	1.052	1.127
0.0000	2.16	1.99	1.81	1.68	1.64	1.68	1.71	1.75	1.79	1.82	1.83	1.85	3.44	3.09	2.76	3.51
0.0028	2.35	2.10	1.90	1.71	1.68	1.69	1.71	1.74	1.76	1.78	1.79	1.79	3.88	3.50	3.15	4.00
0.0057	2.54	2.21	1.95	1.73	1.68	1.68	1.68	1.70	1.71	1.72	1.72	1.72	3.47	3.14	2.85	3.62
0.0085	2.91	2.43	2.09	1.82	1.75	1.75	1.75	1.77	1.78	1.78	1.77	1.77	3.12	2.85	2.60	3.29
0.0114	3.00	2.43	2.08	1.82	1.75	1.74	1.73	1.74	1.74	1.74	1.74	1.73	2.84	2.61	2.40	3.02
0.0142	2.99	2.43	2.03	1.77	1.69	1.66	1.65	1.65	1.65	1.64	1.63	1.63	2.47	2.28	2.11	2.65
0.0171	2.85	2.25	1.88	1.61	1.54	1.52	1.50	1.50	1.50	1.50	1.49	1.48	2.01	1.87	1.75	2.19
0.0199	2.52	2.01	1.68	1.45	1.37	1.35	1.32	1.30	1.29	1.28	1.27	1.25	1.64	1.50	1.42	1.60
0.0228	2.46	1.94	1.61	1.33	1.22	1.18	1.15	1.14	1.12	1.11	1.10	1.09	1.40	1.32	1.26	1.56
0.0257	2.44	1.91	1.57	1.24	1.13	1.10	1.07	1.05	1.03	1.02	1.01	1.00	1.27	1.20	1.15	1.43
0.0285	2.62	2.03	1.65	1.29	1.17	1.13	1.10	1.08	1.06	1.05	1.04	1.02	1.28	1.22	1.18	1.46
0.0314	2.82	2.16	1.74	1.36	1.23	1.18	1.14	1.12	1.10	1.08	1.07	1.06	1.30	1.25	1.22	1.49
0.0342	3.01	2.26	1.82	1.43	1.29	1.23	1.19	1.17	1.14	1.13	1.11	1.10	1.33	1.28	1.25	1.52
0.0371	3.24	2.42	1.93	1.50	1.35	1.26	1.21	1.18	1.16	1.14	1.13	1.13	1.35	1.30	1.28	1.54
0.0399	3.48	2.58	2.06	1.57	1.38	1.30	1.25	1.22	1.19	1.17	1.16	1.14	1.37	1.32	1.31	1.55

AT P.L. AT SURFACE (FOR FAILURE AT P.L.)

BENDING STRESS = -92505.
HERTZIAN STRESS = 248240.
RESIDUAL STRESS = 58000.
TOOTH LOAD = 24190.

RR AT P.L. = 2.058 ENDURANCE/UTS= 0.450

FIG. 19—Print out

than specifying minimum case depth, would instead reduce the permissible loading progressively for those gears where the depth of maximum shear exceeded the effective case depth.

Mr. Wolstencroft echoed Mr. Young's views almost entirely and would ask Mr. Toms if he could please give a further look at the case depth requirements, with particular reference to nitrided gears.

MR. D. E. M. YATES, M.I.Mar.E., said that as all manufacturers of engineering plant knew, the feed-back of performance information was a very essential part of the improvement of the product and its reliability. The paper had shown one of the ways in which the performance information was transmitted back to the manufacturer and, indeed, reflected the improvement in main propulsion gearing and its understanding since the war. It was much to the credit of the author that Lloyd's Register's Rules were under constant review to take into account all aspects of improvements in technology.

Whilst not disagreeing with the author's statement regarding epicyclic gear failures reported by Jones, Ref. [3], it should perhaps be added, that although some gears were modified to incorporate coarser pitches, other modifications to the installations, not covered by the Rules, were carried out at the same time. Since the paper referred to had been published, the gears delivered had given excellent service.

Of course, the author was right to argue for an efficient quality assurance system to be applied throughout the design and manufacture. The keeping of records at all stages of manufacture was an important part of the system. However, the most important check of all was the test of the gear under the design load and speed when all relevant operational measurements could be taken, including efficiency, noise and vibration levels, and the quality of the tooth contact marking. Full load testing using the back-to-back principle could easily be applied to epicyclic gears. The knowledge that the gears had undergone a full load test to the satisfaction of the surveyor gave a datum from which to begin the diagnosis of any shipboard troubles if they occurred.

The thermal and hull distortions which were so troublesome to aft mounted parallel shaft gearing with the two adjacent main wheel bearings were relatively unimportant to an epicyclic gear in designs where the planet carrier was overhung from the slow speed shaft. Misalignment between the carrier and the high speed drive could be accommodated by already developed elastically flexible couplings.

Elastically flexible couplings were used with success in cases where a primary epicyclic gear was mounted on the turbine subframe. Misalignment between the subframe and main gear case was accommodated by the coupling.

As mentioned by the author, the modern techniques of telemetry and strain gauging teeth were now well developed and were being increasingly used in epicyclic gear development by Mr. Yates's company, both on board ship and in a special (20 000 hp) 14 900 kW development test rig (Fig. 20).

MR. D. C. A. LEGGATT, speaking as someone who tried to apply the Rules on a day to day basis, found it most illuminating to have an insight into how they had evolved. It was quite a comfort to know that in spite of the intensely theoretical basis from which the ISO formulae had been derived, there came a point in the application of these for rule formulation when an arbitrary, although experience based, decision on the value of some constant or multiplying factor had had to be made. This made the Rules more open to negotiation and the formulators less God-like.

It should be appreciated that although the loadings for the gears were derived after a very intricate and complicated set of calculations, there calculations were not a guarantee of success or even a complete answer to the questions that were there. The calculations could not take into account all circumstances. The multiplying factors, coefficients, etc, existed only because somebody had considered this particular aspect, and that same person or somebody else was prepared to hazard a value to put on it. There might be many other things which in fact did not occur at all to the people who were formulating. In looking at the Rules and at the paper it occurred to him that some of the things which seemed to be gaps, omissions or points of negotiation, were the mention of overlap ratio and transverse contact ratio, and these were calculated in great detail from accepted formulae. The Rules also specified the need to apply end relief to pinions, and tip relief to the teeth in certain cases. He did not think there was any indication of how the overlap ratios or the contact ratios should be modified to take account of these factors, which surely must be because total face width was one of the datum points in the Rules.

Referring to experience on industrial gears: where there was a helix correction on the assumption of a certain load this could lead to a great deal of discussion at a later date if the helix correction did not quite work out. Contact was still not fully across the tooth largely because the load calculated for was not being attained.

MR. E. J. MYERS said his company had done a lot of work on calculation loads using similar methods to those described in the paper, from which the load distribution factor graphs were very similar in shape to the ones in the paper, but the values were always very much higher. These values were given in the Mudd and Harrison paper, Ref. [7].

Figs. 21 to 24 showed three of the reasons why they were getting load distribution factors always in excess of 1·3 or 1·4, and why the values even for a helix corrected gear, should never approach unity.

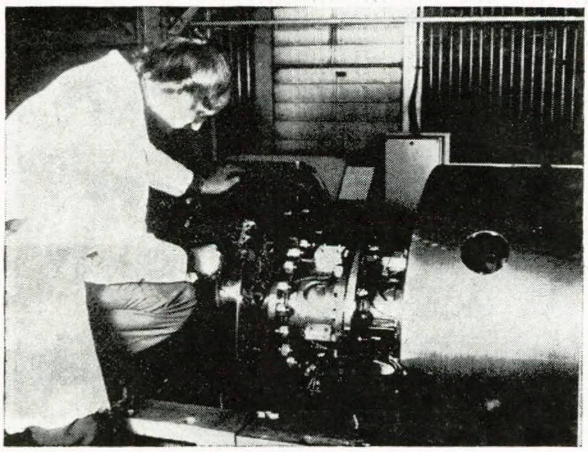

FIG. 20

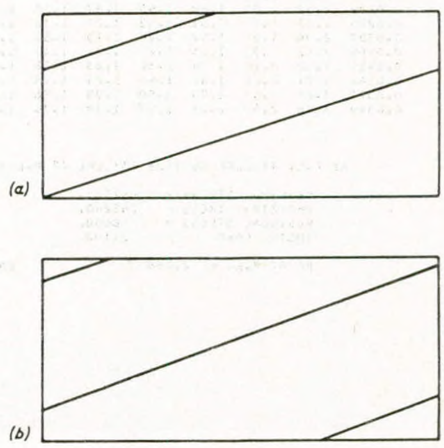

FIG. 21

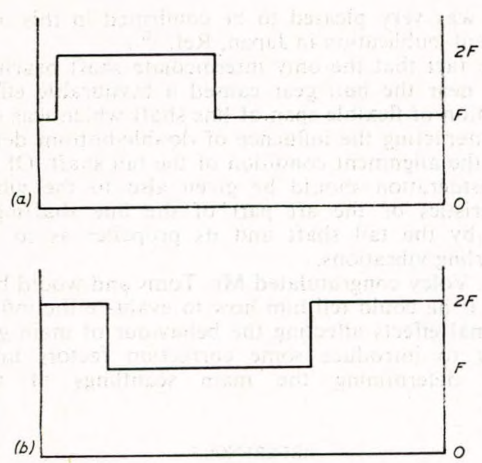

FIG. 22

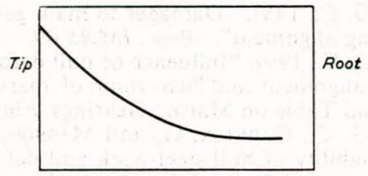

FIG. 23

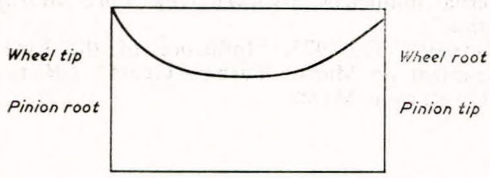

FIG. 24

MR. R. M. HOBSON said that as Mr. Toms was aware, he himself had participated in the approval of gearing for Classification purposes, after the advent of the mathematician but, fortunately, before the advent of the electronic expert. However, he wished to speak about practical aspects of gearing design and, in particular, the statistical analysis which Mr. Toms had given.

Mr. Toms had talked of turbine gear sets. 131 sets were defective, and of these sets a total of 48 had been renewed. Of course, some of these renewals had been on account of rim slip or tooth fracture. Such fractures could be very clearly reported by surveyors in the field.

But, referring to pitting and scuffing, he said that pitting, reported by one surveyor might well have been considered to be of little consequence had it been examined by a colleague at some other port of call.

Statistics should therefore, be looked at very guardedly. However, if one broke down the totals rather simply—perhaps over-simplifying them—one had a total of 91 turbine sets defective on account of pitting and scuffing and of these defective sets only 13 had been renewed.

Turning to oil engine gearing, the figures were 35 sets defective on account of scuffing or pitting, and only three sets renewed.

When one came down into the lower horsepower category covering trawlers and work boats, the proportionate number of defects in relation to the number of sets at risk was very much higher.

He said he would really like to know how many of the tooth fractures were initiated by pitting?

Gearing problems presented one of the biggest headaches to those responsible for operating a ship. There was adequate guidance for acceptable diminution of shell or deck plating. One could gauge a furnace crown and assess the stresses involved. It did not require any great degree of heart-searching to renew a bank of defective boiler tubes, but gearing problems were different. One was inundated with the advice of others that the pitting or scuffing was no worse than last time (or even better than last time). The makers, if called in, had usually seen much worse. A decision had to be made, and he thought that perhaps the paper could have been more helpful here.

He asked whether the power had been reduced on any of the defective cases under notice? Were the ships scrapped prematurely?

On the balance of the statistical evidence as given, some owners were living more or less successfully with scuffing and pitting.

He concluded by asking those responsible for the sophisticated analysis of gear design to spare a thought for their colleagues in the industry wrestling with the less exact science of maintenance and renewal.

Correspondence

MR. P. H. DAWSON wrote that he was glad to note that effects of gear tooth roughness and speed were to be built into the permitted Hertzian stress or surface loading and wished to make three comments on this aspect.

The 13th item in the author's Bibliography and subsequent papers, indicated that there was a relationship between pitting and the ratio of the surface roughnesses to the thickness of the oil film between them for a particular pair of through-hardened steels. This effect was not however quantified in terms of permitted Hertzian stress. The thickness of the oil film could be taken roughly as the square root of the product of the rolling velocity and the oil viscosity. An examination of figures 2(a) Z_V and (9a) Z_{RH} showed a different relationship. For example a doubling of Z_{RH} (say from 2 to 4 or from 4 to 8) was not counteracted by an increase of four times in Z_V (say from 20 to 80). The contributor would welcome a brief indication of how Figs. 2 and 9 were deducted from the statistics.

In the context of these comments it was disappointing that there was no factor reflecting the possible effects of oil viscosity. Such a factor would have to take account of the increase in temperature due to higher losses associated with higher viscosity oils but would make the criterion a more true indication of the actual behaviour of gears.

Finally, although there were great benefits to be obtained from extremely good finishes it was doubtful whether such finishes could be maintained in service. An inadequate flush after a hurried inspection or overhaul could lead to some particles passing through the mesh. It could therefore be dangerous to permit ever increasing loadings for very fine finishes and it was suggested that a cut off be provided, say at 1 micron CLA in Fig. 9(a).

MR. G. C. VOLCY, M.Sc., F.I.Mar.E., wrote that having studied the text of Mr. Toms' very valuable paper he was very pleased to note the interest of Lloyd's Register in environmental conditions related to the correct behaviour of marine gearing. For about 15 years the question of external influences upon such behaviour of propulsive plants had been one of the main items of his activities.

Long ago he had arrived at the conclusion that even the best conceived and manufactured gearing could be seriously damaged due to external influences such as nonrational shafting alignment and deformations of double-bottom steelwork as well as the tilting of thrust block foundations.

The results of his researches related to main gearing were presented in 1968, see Ref. (1). More details concerning this problem were given in a paper, Ref. (2). This problem was further developed at the occasion of the 7th Round Table on Marine Gearing, Ref. (3). In the paper written by Mr. Jones (Ref. [3] of the Bibliography), some references had been made to Mr. Voley's previous works.

From his researches, especially for VLCC the over 200 000 tdw, he could state the importance of the deformation of the outside shell steelwork of the engine room on double-bottom deformations. The essential results of the researchers related to steelwork deformability had been presented in another paper published in France, Germany, Holland and Japan, Ref. [4].

Having been prompted by many people to collect, in one paper, the main topics of his different experiences in the field of damages to gearings due to external influences, he had presented this in December 1974, Ref. [5].

All the above References were mentioned in order to put at the disposal of those who might be interested in such problems the results of work which had been judged useful by many who had requested the assistance of Bureau Veritas.

He said that he agreed with Mr. Toms' statement that the stiffness of today's line shaftings of aft situated propulsive plants was too high for today's flexible huge tonnage vessels. For this reason such line shafting should have as few bearings as possible. He hoped that Mr. Toms would accept an argument that the writer would like to present to him in respect of this statement "in any case the forward supporting bearing of the line shafting should be as far removed from the gearbox as possible". The results of his own previously mentioned experience had led him to another conclusion which was that the only intermediate shaft bearing should not be removed too far the main gearing. This was due to the fact that as this intermediate bearing was also influenced by the deformation of the double-bottom it was counteracting the harmful influence of hogging double-bottom deformation on the equilibrium of bull gear shaft supports and especially on unloading of forward bull gear wheel shaft bearing.

He was very pleased to be confirmed in this opinion by a recent publication in Japan, Ref. [6].

The fact that the only intermediate shaft bearing was situated near the bull gear caused a favourable effect on the creation of flexible span of line shaft which was needed for counteracting the influence of double-bottom deformation on the alignment condition of the tail shaft. Of course due consideration should be given also to the vibratory characteristics of the aft part of the line shafting constituted by the tail shaft and its propeller as to lateral and whirling vibrations.

Mr. Volcy congratulated Mr. Toms and would be very grateful if he could tell him how to evaluate the influences of external effects affecting the behaviour of main gearing in order to introduce some correction factors into the formula determining the main scantlings of marine gearings.

REFERENCES

1) Volcy, G. C., 1967, "Actual behaviour of the main gearing and line shafting alignment conditions"—*Nouveautés Techniques Maritimes.*
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4) Volcy, G. C., Garnier, G. and Masson, J. C., 1973, "Deformability of hull steel-work and deformations of engine rooms of large tankers"—*Technical Bulletin of Bureau Veritas.*
5) Volcy, G. C., 1974, "Reduction gear damages due to external influences, *SNAME New York Metropolitan Section*".
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Author's Reply

Mr. Toms wished to thank all those who had participated in the discussion since by offering their experience they had enhanced its value.

In view of the whole discussion before dealing with detailed sections, a general note would appear to be warranted. Many manufacturers had requested that the Classification Societies should attempt to unify their requirements. Lloyd's Register considered that this could be best achieved by adopting the I.S.O. formulae as a basis since these were the results of deliberations by manufacturers themselves. The various factors which would be based upon an individual Society's experience could then be compared directly.

I.S.O. had 12 grades for the complete range of industrial gearing and in the paper it was decided that the loading formulae in Part 2 should cover the 8 grades in production precision gearing viz. Grades 3 to 10. Part 3 dealt with the requirements for relating these to marine gearing. In particular, the Standards quoted stated that for contact marking and pitch errors the appropriate grades should not exceed those given below. For completeness, the corresponding roughness figures for a single gear (double these would apply to the pair) which limited the use of Figure 9 were given:

i) Turbine primary gears (Grade 5) = 1·0 micro-metre.
ii) Turbine secondary gears (Grade 6) = 1·5 micro-metre.
iii) Oil Engine high speed gears (Grade 5) = 1·0 micro-metre.
iv) Oil Engine low speed gears (Grade 8) = 2·5 micro-metre.

As someone concerned with the manufacture of both steam turbine and oil engine gears, Mr. Young's comments were always worthy of deep consideration. He had criticized the application factors for turbine gears and oil engine gears in conjunction with the previously unpenalized limitation of 33 per cent torsional vibration in the latter. However, this should have been related to the multi-engine drive with "other couplings" and not the single engine drive with "high-elastic couplings". As stated in the paper other effects in addition to torsional vibrations had been taken into account and the application factors reflected the sum total of all. Incidentally, the factors had been printed exactly as in the Rules and, as Dr. Archer pointed out, were equivalent to

$$\frac{1}{K_I}$$

The tandem articulated primary gears having no problems of loadsharing between two wheels, negligible vibrations and being protected from aft end misalignment by the articulation might be considered to take 100 per cent mean load. Then dividing its application factor by those for the other cases one could determine the overload assumed for each. For the multi-engine drive with "other couplings" this gave 148 per cent, which included the torsional vibration component of 33 per cent. It was on this basis, and the note below Table IV, that the single engine drive with "high elastic couplings" was considered little worse than the dual tandem articulated primary gears.

One manufacturer might design within narrow limits for contact ratio but when one considered that, universally, it might vary from 1·25 to 1·8 for normal gears and up to 2·3 for some high tooth spur gears it would be noted that its effect was significant.

The Rules specifically avoided the use of the words "helix correction" although this was the normal method for counteracting pinion deflexion. The paper had given details of helix correction but stated that the need for correction of any type could only be determined by experience. As Part 3 had indicated the removal of lacquer did not give a true indication of load distribution unless the areas of heavy marking could be easily determined. For this reason telemetry had been considered a better method.

Mr. Wolstencroft had joined Mr. Young in the criticism of the shear stress criterion and had quoted the latter at great length. This giant stride was not new since the equating of the factor of safety based on shear stress for the case material to that of the core material in order to determine the case depth had been used for many years. It was this concept of safety factor that would reject the nitrided gear and so the concept quoted in the paper was devised in an attempt to avoid the danger of shearing at the case-core junction, exfoliation etc. The basic problem concerned that which constituted a true nitrided gear and it was difficult to see how this could be resolved without specifying a case depth. The Rules were framed with a basic gear in mind and providing a gear was considered inherently good, nothing in these Rules prevented derating if the basis was not achieved in any respect. The Rules covered "soft nitriding" or liquid salt bath nitriding with a thinner case than "gas nitriding" and an allowance of 80 per cent of the permissible loading for the latter. Where gas nitrided gears did not meet the case depth requirements in the Rules they would be derated down to this percentage according to how far the depth of maximum shear stress exceeded the case depth.

The author had not advocated profile measurements on a single gear but had stated that it must be related to the other gear of the pair. The lower diagram in Figure 10 illustrated this since if the relationship of each was taken separately in relation to the true involute line the errors would be outside the permissible limits. If the true involute lines were tilted to eliminate the small pressure angle error for each, the profile errors would be within the permissible limits and the profiles were conformal. Aware of all the difficulties the Rules had refrained from mentioning profile errors. However, this could not be divorced from the method of determining meshing contact. If the latter were to be relied on to determine good profile then it was essential that the gears be in their load positions so that the zone of inspection was related to the zone of actual contact with the mating profile of the other gear in the pair.

The author agreed with Mr. Larsson in his comparisons of turbine drives with oil engine drives in general. However, no analysis of the features relative to ship type or service had been undertaken to date.

Dr. Archer was in a unique position to give the history of Lloyd's Register gearing rules and he had performed it in his usual masterly manner. In general, the minimum fillet radius in the Lloyd's Register Rules prior to 1973 was still applicable but one had to cover the I.S.O. requirements and adjust the factor accordingly. The stress concentration factor (Y_S) could be justified as unity provided the factor of safety be adjusted in an appropriate manner.

It was difficult to indicate how the loadings and tooth pitches had been affected by the latest revision since there were so many new factors e.g. transverse contact ratio, face width to diameter ratio, surface roughness and, for oil engine installations, application factors and a complete revision of the dynamic factor. It would require investigation of over 100 gear sets to be certain that one could not be accused of undue bias. However, taking an average contact ratio and surface roughness as quoted earlier, the following very general conclusions might be drawn. For turbine gears with through hardened materials in both gears without post hobbing or correction would have slightly reduced K values whilst, with these refinements, the K values would be increased. With surface hardened gears the K values would be higher than formerly whilst for hard on soft they would be less but still slightly greater than manufacturers had proposed to date. Since the oil engine gears had been placed on the same basis as turbine gears, those with through hardened elements in both gears would have somewhat lower K values than previously whereas the corrected gears with post-hobbing having surface hardened materials in both elements of the pair would be generally similar to pre-1973 loadings.

The effect on pitch was more difficult to assess because the whole basis had been changed but very generally there would be little change in turbine gearing but a number of oil engine gears would require larger pitches if they were not normally limited in loadings by the K values. It had to be stressed that for both K value and tooth pitch these were very broad conclusions and would be affected by variations from the average e.g. the lower end of the contact ratio range.

All statistics were open to doubt and the author had only drawn broad conclusions from them. However, as far as the periods at risk were concerned it was thought that factors affecting loadings would manifest themselves within the first six years of service. Troubles experienced after this period were likely to be caused by particular problems in service rather than high average loadings.

By comparison with Fig. 4, the values in Fig. 5 were appropriate to a B/d ratio of 2·25 corresponding to a secondary pinion in which the torque input was on the aft helix.

Mr. Wolstencroft was correct in stating the range in Tables II and III to be too wide for detailed statistical analysis. Having dealt with the teeth of the gears at this time, the manner of construction, gear case stiffness, etc, were aspects to be added to those of manufacturing and installation on which emphasis now had to be concentrated.

The author was pleased to have Mr. Yates' backing on the need for quality assurance procedures and concurred with all his remarks on epicyclic gearing. He agreed the knowledge that the gears had undergone a full load test satisfactorily provided a datum, but this was not possible with all types of gears before full sea trials.

Mr. Leggatt had correctly stated that calculations were not a guarantee of success. One could only continue to improve the calculations based on experience which may leave some of the factors obscured for a considerable time. Again it had to be admitted that unless one could formulate the factor easily or was sufficiently confident to hazard a guess at reasonable values it was liable to be included in the overall ignorance fctor. The overlap ratio was not modified by the end relief in the application of the Rules and the contact ratio was only modified when the tip relief was in excess of British Standard recommendations.

The calculated values for the load distribution factors quoted by Mr. Myers were referred to in the paper. However, note had been taken of other published work as well and average values, which seemed in line with the few measurements known, were plotted. This would be reconsidered as the results of more measurements became available.

With regard to Mr. Hobson's question of the number of tooth fractures in the statistics initiated by pitting, none had been reported. Although a few of the gears had also pitted, no fracture had its origin at a pit. The author agreed that pitting and scuffing were very subjective phenomena but they were also time dependent. Many cases stabilized, whether as result of running-in, reduced power or remedial measures carried out on the installation, and as long as others ran reasonably, i.e. no excessive noise or too great a quantity of detritus appearing in the filters, the desire was to continue running rather than to renew them. With the exception of some occurring late in the last period quoted, as far as was known all the vessels were in service long enough after the incidence of pitting for renewal to have been carried out if considered necessary. Maintenance and renewal and, in fact, all service problems constituted a subject large enough for a paper on its own and had no place in this paper which specifically ended at sea trials. The only hope that could be given was that steps had been taken in the right direction to minimize the incidence of troubles. If pitting or scuffing did occur one could attempt remedial methods according to its position on the teeth but otherwise records had to be kept to determine if it were progressing and, if possible, estimate the rate.

As Mr. Dawson had stated, the effect of oil film thickness and surface roughness had not been quantified in terms of permitted Hertzian stress in his papers but, unfortunately, this was what the Rules had to attempt. All factors involving lubrication were of necessity complicated and there was the problem of the ability of the material to run in. There was a suspicion that surface hardened gears would have different characteristics from through hardened materials and further investigation of this would be required before one could go further than had been accomplished in the paper. Z_V in Fig. 2 was devised from published empirical results including those reported by ZGF, Munich, which formed the basis of the I.S.O. proposal given in ISO/TC60/WG6/Doc.102. Little information existed on the effect of surface roughness between the comparatively rough and high grade finishes. For Z_R in Fig. 9 an attempt had been made to correlate the loadings for hobbed and post-hobbed gears occurring in the statistics with the accuracy grades but it was not claimed that this was an exact relationship. The Rules prior to 1973 allowed an increase in K value of 25 per cent and 10 per cent in bending strength for what amounted to Grade 3 (approximately 0·25 micro metres CLA per gear) over Grade 7 to 8 (approximately 2·5 micro metres CLA per gear) and this ratio had been retained in Fig. 9. The Rules had a cut off at the surface roughness considered appropriate to Grade 3.

The absence of a factor reflecting the possible effects of oil viscosity was due to the fact that, at best, the choice of oil had to be a compromise between that appropriate to low and high speed gears and it could not be guaranteed that the same type would be used throughout the life of the gears particularly where one desired to use one oil for all lubrication on the ship. Again ZGF, Munich reported only a few per cent increase for through hardened materials and none at all for surface hardened gears with an increase in viscosity from 100 to 300 centistrokes at 50°C.

Mr. Volcy was well known for his papers on the alignment of shafting and deformations of the hull structure. Others, including members of Lloyd's Register, had been and were still carrying out work successfully in this field. He was sure Mr. Volcy would agree that a single bearing situated approximately in the middle of the forward intermediate shaft would satisfy the statement in the paper regarding the positioning of the bearings. Whilst Mr. Volcy was apparently content to leave things as they were the author, mindful that a palliative for one case could be harmful for another, was concerned to progress towards the stage when the gearbox would not rely upon outside influences to prevent the unloading of a main wheel shaft bearing. Dr. Pinnekamp[6] had indicated this approach in the attempt to obtain symmetrical movements at the bearings by correct design of the support. That reference had given simple formulae for separating out the effect of misalignment although these could eventually be found to be over simplified. Where possible, measurements were being taken in an attempt to correlate the load distribution across the teeth with the influence of external effects. When these were sufficiently numerous it might be possible to devise a sophisticated formula to introduce some correction factors into the loading formula.

Meanwhile, based purely on calculations and thus subject to revision as a result of measurements, a very rough guide to the anicipated increase in loading based on the misalignment appropriate to the grades of accuracy given in British Standard Specifications would be $(1 + n/20)$, where n was the grade number, approximating to that given by Harrison and Mudd[7].