Marine Main Propulsion Gears – A Classification Society Review

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SYNOPSIS

The first part of this paper examines the survey data collected by Lloyd's Register of Shipping for marine main propulsion gears over the last ten years. These data have been analysed and the results are presented and discussed. Perhaps most interesting are the conclusions regarding the relative reliability of main propulsion reduction gear units during this period and whether any trends have been detectable over the years. A necessary contribution to reliability is made by the detailed appraisal of new designs. This has become an accepted procedure in the marine gear industry, and the second part of the paper deals with the single most important aspect of this appraisal, namely the tooth loading calculations. The International Standards Organisation (ISO) Draft International Standard (DIS) 6336 'Principles for the Calculation of Load Capacity of Spur and Helical Gears' seems now to have been generally accepted by all parties concerned and experiences with parts I, II, III and V applied to some of the latest main propulsion gearing designs are given, including notes on the principal factors involved. The aim has been wherever possible and without significantly affecting accuracy to condense the calculation methods specifically for this application. It is intended that the Lloyd's Register of Shipping gearing rules will be revised along these lines. Finally, a few aspects of the survey and inspection procedures during manufacture, which again are considered to make a necessary contribution to reliability, are mentioned and a possible development based on reliability concepts is outlined.

INTRODUCTION

It is nearly twelve years since a paper relating exclusively to marine reduction gearing was presented to the Institute of Marine Engineers by Lloyd's Register of Shipping (LRS). At that time, Toms[1] reviewed the service experience of gears built since Rules for gearing were introduced in 1946 and described the development of the Rules up to the major revision which became effective in 1974. The background to the 1974 Rules, which remain current with minor revisions, was described in depth. It is now opportune to review the service experience of gears built under the latest Rules, to consider the changes introduced by the gear industry and to outline the future revisions in the Rules which are being developed.

The major change which has occurred in the design of merchant ships in the intervening period is the almost total demise of steam turbine ships. This is reflected in this paper by limiting consideration to reduction gears in oil engine installations. The dominance of the oil engine has now been established, so that during 1985 only two new steam ships were completed,[2] although certain special categories will probably remain the preserve of geared steam turbine propulsion. At the end of 1985 there were 225 geared steam turbine installations currently classed by LRS in ships built since 1960 compared with 444 at the end of 1972. In the same period the number of geared oil engine installations in class over 500 bhp has increased from 1629 in 1972 to 4201 in 1985. This situation leads the authors to consider that it is reasonable to restrict the scope of this paper to oil engine reduction gears.

A classification society design appraisal and drawing approval for a marine main propulsion gear unit is primarily concerned with the tooth load capacity. The power rating of such a gear unit is then normally set by the applicable classification society Rules, *eg* Ref. 3, which limit tooth flank surface and tooth root bending stress levels.

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It is now generally accepted, not least in the interests of standardization, that such Rules should be based on the ISO DIS 6336 calculation methods for involute gears.[4] This DIS, parts I to V, is intended to cover the whole range of parallel spur and helical involute gears and applications. They are therefore by their nature complex and lengthy. Indeed to cater for differing applications and sensitivities various levels of calculation accuracy must be offered and alternative methods (A to D, with complexity decreasing in that order) are quoted for the factors affected. It remains then to select the most appropriate formulae for these factors and to list specifically values for other factors.

RECENT SERVICE EXPERIENCE

The operational reliability of oil engine reduction gearing has been assessed by reference to the records of LRS. These predominantly consist of reports by surveyors but some reference has also been made to information contained in shipowners' planned maintenance scheme records. The main study is based on an evaluation of damages recorded in survey reports relating to geared installations with a gearbox rating of 220 kW and greater: gearboxes of a lower rating do not require plan approval in accordance with the Rules for gearing. All ships built during the period from the beginning of 1975 to the end of 1985 and classed by LRS are considered together with all defects reported up to the end of June 1986. The service history of the population relates to the current Rules introduced in 1974 and will be compared with the data cited by Toms for earlier post-war periods. Although some gear units are known to have suffered systematic failures, the causes of which have since been established and eliminated from future installations, the data are presented without removal of any records.

In order to place the reliability of gearing in perspective the study has included the evaluation of defects in the major elements of the propulsion systems: engines, shafting and propellers. The number of components considered, categorized by gearbox power and input speed, is shown in Table I. Some small discrepancies appear between, for instance, the number of shafts and the number of gearboxes because of replacements. The total service life of the 3427 gearing units is 18429 years. It is interesting to note that despite considerable interest a few years ago there have not been many geared low-speed oil engine installations built, presumably as a result of the development of ultra-long-stroke engines.

The subdivision of the total number of gearing units by arrangement is given in Table II, clearly showing the four predominant types to be single and double reduction non-epicyclic gears with or without reversing capability. These four categories constitute 98% of the total study population and since no defects were found relating specifically to epicyclic components no separation on account of physical arrangement has been made in the ensuing discussion.

Before presenting the data relating to defects found in service it is useful to consider the performance requirement for defining successful operation and the various levels of failure definition. Reduction gears are expected to operate under a wide range of loads with very little invasive maintenance throughout their intended life. Unlike many mechanical devices there is little scope for an intended plan for progressive component renewal to counter wear-out. When basic design limits are exceeded, for whatever reason, failure generally follows in a short period of time. However, many gears continue to perform satisfactorily in spite of minor damage. Failure can be construed as a total or partial loss of power transmission capability, although when considering service reliability cognizance must be taken of defects which cause resort to rectification or replacement in situations where failure is considered to be a likely consequence of continued operation.

All failure information is related to the existing operational environment. If steam turbine gearing is examined some consideration must be given to the effect of the adverse trading conditions of recent years, particularly slow steaming and lay-up. In the case of oil engine gearing it is not anticipated that the information is other than typical for normally intended operational patterns.

Consider first the incidence rate of defect reports for the major elements of the propulsion system. This information is shown in Table III. However, many minor stoppages will be repaired without resort to the classification society. Additionally, and conversely, some of the defects recorded can not be considered to be failures, requiring either minor rectification to restore satisfactory condition or regular observation to ensure that further degradation does not occur. By comparison, Table IV shows the frequency of stoppages, or breakdowns, recorded by a shipowner in respect of five ships. Those ships, two being twin screw, are classed by LRS and each propulsion system comprised a geared medium speed engine of 6000–6500 kW. As would be expected, the reported breakdown frequency is significantly higher than the defect incidence rate recorded by the classification society. This suggests that the gearbox constitutes about 1% of the total number of propulsion system breakdowns. In fact none of the reported defects affecting the gearbox are recorded in the information used in the preparation of Table III; all are minor in nature. On any basis gearing is highly reliable compared with other elements in the total system.

Returning to the study of classification records of reported defects, Table V shows the distribution of damage by location within the gearbox. Nearly half of the defects concern

Table I: Number of propulsion system components in study population

Engine Speed (rev/min)	*Gearbox Rating (kW)*	*Number at Risk*				
		Main Engine	*Gear-box*	*Shafting*	*CPP*	*FPP*
0–299	220 +	5	3	3	2	1
300–999	220–4999	2045	1893	1891	1051	842
300–999	5000–999	331	265	268	176	92
300–999	10000 +	141	84	83	58	26
1000 +	220–2499	1179	1132	1140	123	1011
1000 +	2500 +	55	50	50	34	16
All	220 +	3756	3427	3435	1444	1988

Table II: Distribution of gear arrangements in study population

Engine speed (rev/min)			0–299	300–999			1000 +	
Gearbox rating (kW)			220 +	220–4999	5000–9999	10000+	220–2499	2500+
Reduction stages	*Rever-sing*	*Epi-cyclic*						
Single	No	No	3	1094	243	73	131	35
	No	Yes		5	5	2	2	2
	Yes	No		527	6	1	900	10
	Yes	Yes		4	1		28	2
Double	No	No		90	7	5	3	1
	No	Yes		1	2			
	Yes	No		172			52	
	Yes	Yes			1		16	

Table III: Defect incidence rate based on survey reports

Input speed (rev/min)	*Gearbox rating* (kW)	*Defect incidence rate per set year*				
		Main Engine	*Gear box*	*Shafting*	*CPP*	*FPP*
0–299	220+	0	0	0	0	0
300–999	220–4999	0.135	0.033	0.072	0.085	0.078
300–999	5000–9999	0.314	0.052	0.097	0.135	0.072
300–999	10000 +	0.308	0.060	0.108	0.133	0.021
1000 +	220–2499	0.047	0.024	0.068	0.049	0.047
1000 +	2500 +	0.246	0.052	0.071	0.049	0
All	220 +	0.141	0.034	0.074	0.091	0.064

flexible couplings and clutches. Bearing damage accounts for a further 16% of the total. Although many aspects of gear construction and condition are considered by the classification surveyor during building and in service, the Rules relating to design are limited to assessment of the teeth with respect to tooth bending strength and tooth surface loading. This strategy is based on the prevention of failure, as discussed above, of the gearbox in terms of loss of power tranmission capability. The remainder of this section will be concerned exclusively with the service experience of gear teeth.

Table IV: Propulsion system breakdowns recorded in planned maintenance scheme

	Number at risk	*Time at risk (years)*	*Number of break downs*	*Break down rate per set year*
Main engine	7	34.4	634	18.42
Gearbox	7	31.3	7 (1)	0.22
CP propeller & shafting	4	17.4	24 (2)	1.38
Fixed pitch propeller & shafting	3	15.3	4 (2)	0.26

1. These breakdowns are identified by cause as follows:
   leakage 2
   obstruction 1
   overheating 1
   undefined 3
2. No defects in relation to propellers are reported

Table V: Distirbution of defect locations by component part affected

Part affected	*Number of defects*	*% of total defects*
Clutches	149	24.1
Flexible couplings	140	22.7
Casings & foundations	22	3.6
Pinions	124 (1)	20.1
Wheels	57 (2)	9.2
Thrust bearings	12	1.9
Lubrication system	50	8.1
Power take-off	43	7.0
Reversing gear	8	1.3
Miscellaneous	13	2.1

1. 76 reports relate to bearings, remainder concern pinion
2. 15 reports relate to bearings, remainder concern wheel
Source - survey records

Table VI: Incidence of tooth defects and consequent renewals

Engine speed (rev/min)	*Gearbox rating (kW)*	*Number of sets affected*	*Number of of sets renewed*	*Percentage of sets renewed per year*
0–299	220 +	0	0	0.0
300–999	220–4999	42	30	0.27
300–999	5000–9999	10	3	0.18
300–999	10000 +	2	0	0.0
1000 +	220–2499	4	4	0.08
1000 +	2500 +	5	3	1.43
All	220 +	63	40	0.22

Source - survey records

The records based on classification survey reports are believed to be reasonably complete in relation to gear tooth damage, particularly where severe damage or replacement is involved. For the study population the number of gear sets which have suffered tooth damage is shown in Table VI, together with the consequent renewals involving some or all of the gear elements. The overall renewal rate of 0.22% per set year can be compared with similar information cited by Toms[1] (his Table III). Toms gives for:

1. Ships built after 1946 and in class at the end of 1965 for service over 1961–1965, 0.10% per set year.

2. Ships built after 1959 and in class at the end of 1972 for service over 1966–1972, 0.15% per set year.

There are several reasons why direct comparisons between the data given in Table VI and those quoted by Toms should be made with extreme caution. Toms produced his results from the old manual records system whereas the data presented here have been prepared from the current computer-held database which is likely to produce more complete retrieval reports. Furthermore, it appears that Toms only considered damage recorded as being caused by pitting, scuffing and tooth fracture. However, apart from certain differences which result from the exact definition of the data extraction exercise the fundamental change is that in the data prepared for this paper all defects reported after date of build are included. Toms prepared data for which some ship service had existed before the study period, thereby eliminating some of the early failures, which tends to suppress the calculated defect incidence rates. In addition, since the values cited are referred to an 'elapsed time' base, changes in the utilisation rate are important: this has obviously increased significantly since 1965.

By examining the defect description, shown in Table VII, it appears that the incidence rate of renewal because of pitting and other surface damage (flaking, spalling *etc.*) has doubled and that caused by tooth breakage has increased slightly. In several cases, as indicated in Table VII, damage is recorded as being consequent on another incident and not directly attributable to the gear design. It is significant that of the sets affected the current study shows a higher percentage leading to renewal (63% as opposed to 31% reported by Toms). This may show a trend towards lower damage tolerance in highly loaded gears. Gears built in accordance with the Rules do indeed perform satisfactorily in service and the renewal rate (neglecting cases caused by extraneous material passing through the mesh, propeller impact and corrosion during lay up) is about 0.19% per set year.

Taking cognizance of the comments made above, the service performance of gears built since the introduction of the 1974 Rule revision is as good as that reported by Toms for earlier versions of the Rules. In terms of damages which result in total loss of power transmission capability and a shut-down of the propulsion installation at sea, the records indicate eight cases or 0.04% per set year, which must be regarded as highly satisfactory. (In other words, this represents one total failure every 2300 set years of operation.)

DESIGN APPRAISAL OF TOOTH LOADING

Symbols and terms are listed in Appendix I.

Tooth loading calculations are accepted as the single most important aspect of a gearing design appraisal. The present LRS gearing rules are based, in this respect, on the ISO TC60 WG.6 (1967) document no. 75E, which has subsequently been revised and further developed through the DP (draft proposal) stage into the ISO DIS 6336 parts I, II and III (1978). A study of this DIS 6336, referred to simply today as the ISO calculation methods, with a view to a corresponding revision of the LRS gearing rules, was a logical consequence and is now nearing completion. The opportunity is taken here to outline the resulting condensed calculation methods together with some initial experience of their application to recently submitted designs.

It must be stressed that the following is still subject to possible change and should therefore not be construed as a final draft LRS Rule proposal.

Where alternative calculation methods are offered by ISO these have been evaluated and the most appropriate selected; the overall aim is to reduce the complexity and length of the calculations so far as practical. Also, as already mentioned, the ISO calculation methods have been developed to cover the whole range of parallel spur and helical involute gears and it is therefore possible to reduce further the length of calculations where it is feasible at the onset to define limiting parameters for a particular application.

Merchant marine main propulsion gearing today and in the immediate future has, for reasons given above, been taken in this paper to mean reduction gearing for oil engines. From a study of designs manufactured during the last two years the following limiting parameters would seem reasonable for this application.

1. Linear speed at pitch circle $v < 50$ m/s.
2. Pinion facewidth to diameter ratio $b/d_1 < 1.5$.
3. Number of teeth in pinion $z_1 > 20$.
4. Accuracy of manufacture: ISO 1328 (1975)[5] Q grade 6 or better.
5. Normal pressure angle $\alpha_n = 20°$.
6. Pinions symmetrically positioned between bearings.
7. Helix angle $\beta \leq 30°$.
8. Nominal tangential tooth load per unit facewidth $F_t/b >$ 150 N/mm.

In addition, it can be said that the vast majority of designs are of single helical, carburised, hardened and profile ground gear elements.

Traditionally, design appraisal of tooth loading has involved calculations of tooth surface and root bending stresses and the ISO methods for these are discussed below.

Scuffing is a very complex phenomena for which it has proved extremely difficult over the years to develop a reliable calculation method. The incidence of scuffing on marine main propulsion gears has also declined over the last few years, no doubt in part because of the increasing role played by oil engine reduction gears with their relatively low pitchline speeds. The low incidence of scuffing defects in comparison with surface damage and breakage is clearly shown in Table VII. Whilst some experience has been gained with both the 'flash temperature' and 'integral temperature' methods, these have not been included. It is perhaps worth mentioning, however, that the calculation method based on the 'integral temperature' requires test data [T_1 (Nm) scuffing load torque from the FZG-Test A/8, 3/90] which are not included in the draft standard.

K loading factors

Application factor K_A

This factor represents external influences from the driving (oil engine) and driven (main propulsion shafting and propeller) machinery which act to increase the transmitted torque. The main component for the gears under consideration is the torsional vibratory torque which should be limited to one-third of the full rated transmission torque. Other components are those caused by misalignment and, where the torque path is split, unequal load sharing. Based on present practice, Table VIII lists values for K_A that can initially be used.

Dynamic factor K_V

This factor represents internal influences which act to produce dynamic increases in the transmitted torque. It is thus primarily a function of pitch line speed and tooth pitch and profile deviations.

Table VII: Distribution of gear tooth damage by failure mechanism

Description of defect (1)	*Number of sets affected*	*Number of sets renewed*	*Percentage of sets renewed per year*
Scuffed	2	1	0.005
Pitted	13	6	0.033
Other surface damage	6	2	0.011
Broken, chipped, cracked	35	28	0.152
Corroded	2	1	0.005
Indented	5	2	0.011
Total	63	40	0.217

(1) Attributed failure causes include:
failure of torsional vibration damper (2 cases)
clutch failure (2 cases)
propeller fouling (1 case)
shaft misalignment due to bearing damage (3 cases)
foreign objects (4 cases)
corrosion due to lay up (2 cases)
Most are not attributed to a specific cause
Source - survey records

Table VIII: Application factor K_A

Main propulsion oil engine single and double reduction gears	*Single engine drive*	*Multi engine drive*
Hydraulic coupling or equivalent on input	1.10	1.25
High elastic coupling on input	1.30	1.45
Other couplings	1.50	1.70
Auxiliary gears		
Turbine, electrical and diesel engine drives with hydraulic coupling or equivalent on input		1.0
Diesel engine drives with high elastic coupling on input		1.20
Diesel engine drives with other couplings		1.40

For the application being considered ISO calculation method C is sufficiently accurate and an expression of the form below can be used:

$$\text{Helical gears} \quad K_V = 1 + Q^2 v z_1 \times 10^{-5} = K_{V\beta} \quad \text{with } \varepsilon_\beta \geq 1 \tag{1}$$

$$\text{Spur gears} \quad K_V = 1 + 1.8 Q^2 v z_1 \times 10^{-5} = K_{V\alpha} \tag{2}$$

$$\text{Helical gears} \quad K_V = K_{V\alpha} - \varepsilon_\beta (K_{V\alpha} - K_{V\beta}) \quad \text{with } \varepsilon_\beta < 1 \tag{3}$$

Normally $K_V < 1.3$ with operation in the sub-critical speed range. It is perhaps worth mentioning at this point that the consequence of operation in and around the main resonating speed range, as indicated in the ISO calculation method B and the new BS 436,[6] with the resulting high K_V values, does not seem to have been borne out in practise with many steam turbine reduction gears.

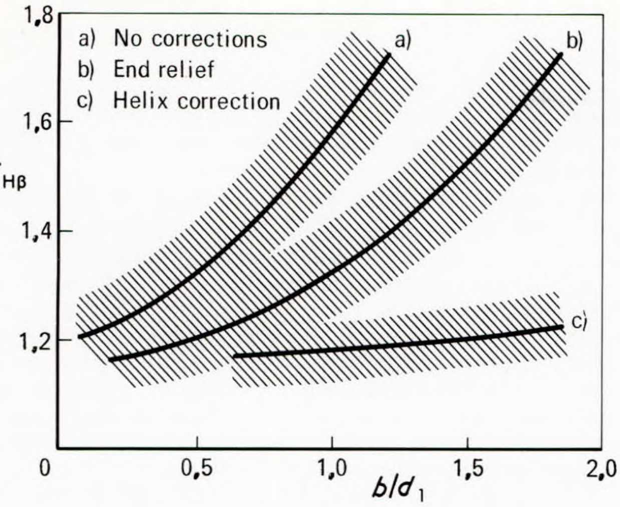


FIG. 1: Longitudinal load distribution factors. Average values for spur and helical gears

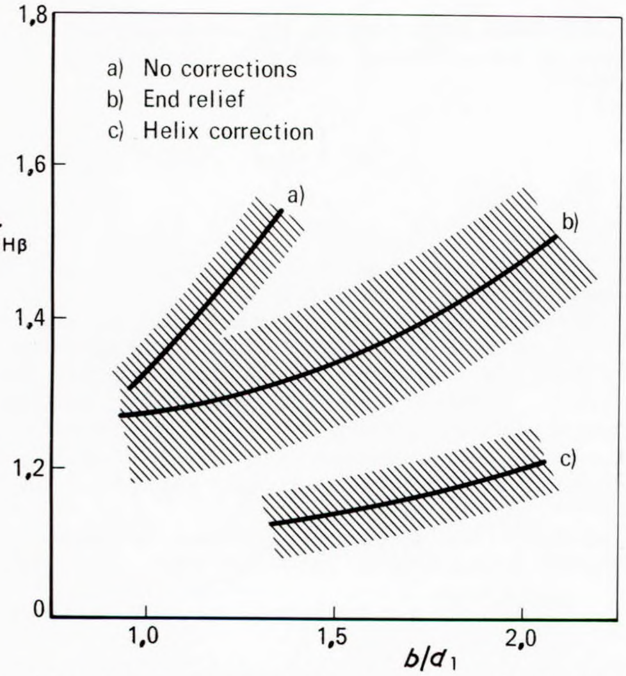


FIG. 2: Longitudinal load distribution factors. Average values for double helical gears

Longitudinal load distribution factors $K_{H\beta}$ and $K_{F\beta}$

These factors take into account non-uniform tooth load across the facewidth caused mainly in this application by:

1. Tooth manufacturing deviations.
2. Tooth mesh stiffness.
3. Tooth flank 'running in' effects.
4. Wheel body, shafts, bearings and gear case stiffnesses.
5. Bearing clearances.
6. Shaft, bearing and gear case manufacturing deviations.

It has been found that acceptable results can be obtained with ISO calculation method C which is expressed in the form below:

$$K_{H\beta} = 1 + \frac{b\, F_{\beta y}\, C_\gamma}{2\, F_t\, K_A\, K_V} \le 2 \tag{4}$$

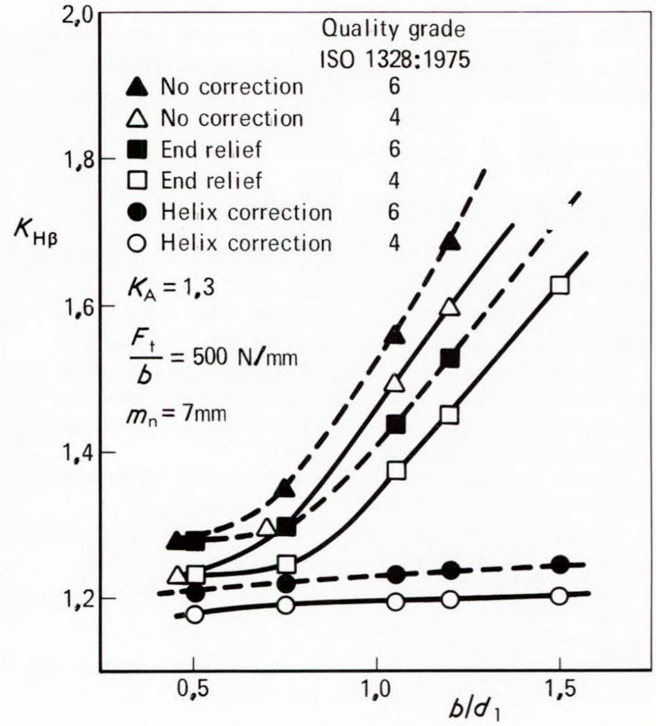


FIG. 3: Examples of load distribution factor changes with a basic single helical design

The mean value of mesh stiffness C_γ is calculated in accordance with the adopted ISO method. This calculation for the type of gears under consideration results in values of C_γ between 23 and 30 N/mm µm.

$$F_{\beta y} = F_{\beta x} - y_\beta = |f_{sh} + f_{ma}| - y_\beta \tag{5}$$

The values of f_{sh}, f_{ma} and y_β are, at the design stage, estimations and as such subject to some discussion. It therefore seems reasonable to expect that the parameter used in the estimations, namely F_β the total tooth alignment deviation, should be determined after the teeth have been finish machined.

For the curves shown in Figs 1 to 3 the following estimates have been used at the design stage:

$$f_{ma} = \frac{2}{3} F_\beta \tag{6}$$

or where helix correction has been applied

$$f_{ma} = \frac{1}{3} F_\beta \tag{7}$$

and

$$f_{sh} = f_{sho} \frac{F_t\, K_A\, K_V}{b} \tag{8}$$

The calculation of f_{sho} can be simplified in the form given below which at the same time has been slightly modified in line with experience. For gears without helix correction and without end relief $f_{sho} = 31\gamma \times 10^{-3}$ µm mm/N.

For gears without helix correction but with end relief f_{sho} = $21\gamma \times 10^{-3}$ µm mm/N with minimum values which are also applicable where helix correction has been applied:

spur gears $f_{sho} = 5 \times 10^{-3}$ µm mm/N

helical gears $f_{sho} = 13 \times 10^{-3}$ µm mm/N

In the above $\gamma = (b/d_1)^2$ for single helical and spur gears and $\gamma = 3(b/2d_1)^2$ for double helical gears.

The calculation and estimation for $K_{F\beta}$ and y_β, respectively, are taken directly from ISO DIS 6336 Part 1.

Figures 1 and 2 illustrate the average type of result from the above method, although it must be appreciated that with the number and type of variables involved in the calculation a good deal of scatter is to be expected.

Figure 3 shows specific results for a gear mesh having the following averaged particulars:

- Single helical, pinion and wheel teeth carburised hardened and profile ground
- F_t/b = 500 N/mm
- m_n = 7 mm
- C_γ = 25 N/mm µm

Points have been plotted for quality levels (Q) 4 and 6. The actual b/d_1 ratio selected for this design was 0.75.

Transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$

These factors take into account non-uniform tooth loading down the tooth flank influenced by:

1. Tooth manufacturing deviations.
2. Tooth mesh stiffness.
3. Tooth flank 'running in' effects.
4. Tooth mesh geometery.
5. Tip and root relief.

ISO calculation method B is straightforward and taken directly from ISO DIS 6336 Part 1. This requires a value for f_{pb}, the base pitch deviation, and normally at the design stage the maximum for the quality level envisaged appropriate to the wheel should be used. When tip or tip and root relief is applied half this particular value may be taken. As with F_β, f_{pb} should be determined after the teeth have been finish machined.

For the gears under consideration, the calculated $K_{H\alpha}$ and $K_{F\alpha}$ values are invariably unity and certainly no more than 1.1.

Tooth loading for surface stress

The Hertzian contact stress σ_H at the pitch circle is calculated in accordance with the established ISO formula

$$\sigma_H = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t(u+1)}{d_1\, b\, u} K_A K_V K_{H\beta} K_{H\alpha}} \tag{9}$$

where Z_H, Z_ε and Z_β are geometric factors and Z_E is a material elasticity factor.

A maximum permissible Hertzian contact stress σ_{HP} may be calculated for this application as follows:

$$\sigma_{HP} = \frac{\sigma_{Hlim} Z_R Z_V}{S_{Hmin}} \tag{10}$$

Strictly in accordance with the ISO method, σ_{HP} is calculated separately for pinion and wheel but it can also be considered in the more traditional way for a pinion and wheel combination, thus eliminating the Z_w (work hardening) factor.

In line with previous tests and experience a larger differential is maintained where a surface hardened pinion meshes with a through hardened wheel than when both gears are through hardened.

For example, with wheel materials of σ_B = 700 N/mm² and σ_B = 1300 N/mm², increases of 23% and 13%, respectively, are appropriate when the pinion is surface hardened. ISO, with the Z_w factor, allows increases of only 15% and 5%, respectively.

The lubrication factor Z_L can be omitted as a relatively narrow range of oil viscosities are normally encountered. There is also little control of lubricating oil grades by the manufacturer or classification society in service and it seems unreasonable to give credit for a high-viscosity grade which can easily be changed later during the vessel's life.

The velocity factor Z_V can be simplified over the range in question and expressed in the form:

$$Z_V = 0.88 + 0.23\left(0.8 + \frac{32}{v}\right)^{-\frac{1}{2}} \tag{11}$$

The surface finish factor Z_R can also be simplified for the range of surface finishes appropriate and expressed in the form:

$$Z_R = \left(\frac{1}{R_a}\right)^{0.11} \tag{12}$$

When the tooth flank surface roughness of pinion and wheel differ the larger value of R_a is taken.

Acceptable limits for R_a and measurement procedures are outlined in BS 1807(1981), para 7.1.2.[7] The actual values of R_a should be determined after the teeth have been finish machined. Experience shows that with modern tooth flank grinding machines, operating wet or dry, an R_a value better than 0.8 µm can be expected. Conversely, it is unreasonable to accept claims that R_a values better than 0.3 µm can be attained.

Table IX lists acceptable σ_{Hlim} values for material heat treatment combinations of pinion and wheel encountered in

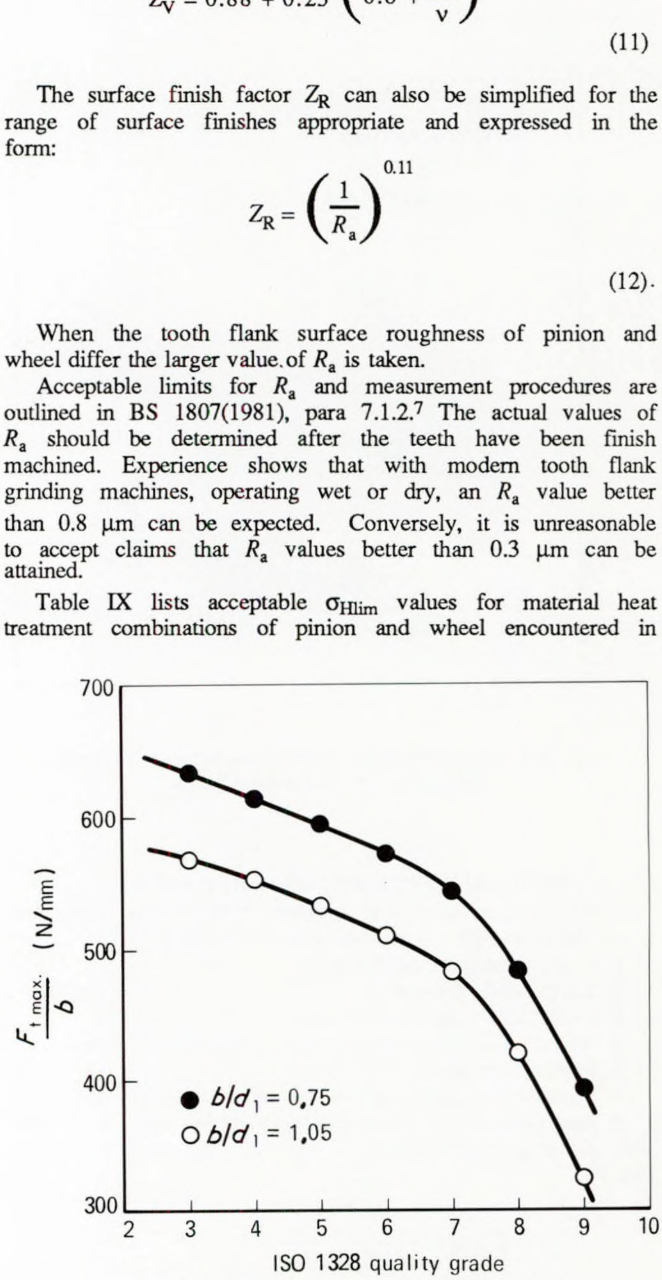


FIG. 4: Effect of quality grade on design loading

Table IX: Acceptable σ_{Hlim} values

Heat treatment		
Pinion	*Wheel*	σ_{Hlim} (N/mm²)
Through hardened	Through hardened	$0.46\,\sigma_{B2} + 255$
Surface hardened	Through hardened	$0.42\sigma_{B2} + 415$
Carburised nitrided or induction hardened	Soft bath nitrided (Tuftrided)	1000
Carburised nitrided or induction hardened	Induction hardened	$0.88\,H_{V2} + 675$
Carburised or nitrided	Nitrided	1300
Carburised	Carburised	1500

Table X: Acceptable σ_{Flim} values

Heat reatment	σ_{Flim} (N/mm²)
Through hardened carbon steel	$0.09\,\sigma_B + 150$
Through hardened alloy steel	$0.01\,\sigma_B + 185$
Soft bath nitrided (Tuftrided)	330
Induction hardened	$0.35\,H_V + 125$
Gas nitrided	390
Carburised (B)	410
Carburised (A)	450

Where (A) is applicable for Cr NiMo carburising steels and (B) is applicable for other carburising steels

Table XI: Proposed values of S_{Hmin} and S_{Fmin}

	S_{Hmin}	S_{Fmin}
Main propulsion gears	1.4	1.65
Main propulsion gears small craft single screw	1.25	1.50
Main propulsion gears small craft multiple screw	1.20	1.45
Auxiliary gears	1.15	1.40

the marine field. These values lie around the middle of the ranges quoted in ISO.

Mention should be made here of the grouping in ISO of 'Induction – flame hardened gears'. Whilst induction hardening of large wheels is accepted in the marine main propulsion field, flame hardening has not received the same recognition.

Tooth loading for root bending stress

The bending stress σ_F at the tooth root must be equal to or less than the allowable tooth root bending stress σ_{FP}.

It is considered that σ_F should be calculated with the load applied at the outermost point of single tooth contact in accordance with the established ISO formula:

$$\sigma_F = \frac{F_t}{b\,m_n} Y_F\, Y_S\, Y_\beta\, K_A\, K_V\, K_{F\beta}\, K_{F\alpha} \qquad (13)$$

Y_F, Y_S and Y_β are calculated exactly as outlined in ISO DIS 6336, Part 3.

The effect of protuberance is small, normally increasing the product $Y_F Y_S$ by no more than 3% and to be correct an estimate of the protuberance left after final machining should be used:

$$\sigma_{FP} = \frac{\sigma_{Flim}\, Y_{ST}\, Y_{\delta relT}\, Y_{RrelT}\, Y_X}{S_{Fmin}\, Y_D} \qquad (14)$$

As the ISO values for σ_{Flim} are used $Y_{ST} = 2.0$.

The calculation of $Y_{\delta relT}$ can be simplified for the range of q_s (notch parameter) involved and expressed in the form:

•for through hardened steels ·

$$Y_{\delta relT} = 1 + 0.036(q_s - 2.5)\left(1 - \frac{\sigma_y}{1200}\right) \qquad (15)$$

•for carburised and induction hardened steels

$$Y_{\delta relT} = 1 + 0.008(q_s - 2.5) \qquad (16)$$

•for nitrided steels

$$Y_{\delta relT} = 1 + 0.04(q_s - 2.5) \qquad (17)$$

$Y_{\delta relT}$ is calculated exactly as outlined in ISO DIS 6336, Part 3. For R_Z, $6R_a$ is substituted, which refers to the surface roughness in the tooth root fillet area. The same remarks apply to the tooth root surface roughness as were given for the tooth flank surface roughness.

The size factor Y_X is also calculated exactly as given in ISO.

Table X lists acceptable σ_{Flim} values for material heat treaments encountered in the marine field. These values lie again around the middle of the ranges quoted in ISO although an extra allowance is considered appropriate for the more sophisticated carburising steels.

It is appreciated that in industry generally manufacturers are taking higher values of σ_{Flim} and σ_{Hlim} near to the top of the ISO ranges. However, for the level of material testing specified with marine gears, the values listed in Tables IX and X are considered reasonable.

The additional factor Y_D is introduced to take account of the following features:

1. Idler duty.
2. Shrink fit stresses.
3. Shot peening of the tooth roots.

Where none of the above apply $Y_D = 1.0$.

Factors of Safety, S_{Hmin} and S_{Fmin}

Specific values of S_{Hmin} and S_{Fmin} are not given in ISO DIS 6336 but the values listed in Table XI are a proposal based upon present limits for marine main propulsion gears, including small craft application. For the sake of completeness essential auxiliary gears have also been included. The table is presented to give an idea of values being considered and for the present should be taken only for guidance. Small craft, defined as being normally less than 24 m overall length, have differences from steel ships which reasonably justify special consideration.

These include:

1. Difference in loading spectrum with a higher percentage of operation away from the maximum continuous rating.
2. The gear units are normally line or batch produced in standard frame sizes with service experience to hand.
3. The smaller size of gear units and standard frame size lends itself to easier procurement and fitting of spare parts when needed.

Overall effect of changing accuracy grade

To demonstrate the effect which a change in the Q grade has with the ISO calculation methods, the averaged gear from Fig. 3 is shown with the limiting (F_t/b) values plotted against the Q grades 3 to 9 in Fig. 4. It is clear how strongly the Q grade influences the design load capacity of the gears.

INSPECTION AND TESTS

Once a design has been appraised it becomes the responsibility of the manufacturer's production and quality assurance (QA) departments together with the classification society surveyor and any purchaser's inspector to ensure that the design specification and tolerances are met. It is in the interest of all parties that an adequate QA system exists and its importance cannot be over emphasised. However, for the manufacture of gears a QA system should not be expected to substitute for the necessary expertise.

Materials

ISO DIS 6336 Part 5 and DIN 3990 Draft Part 5[8] outline not only endurance limits for materials but also define quality levels (ME, MQ, ML, decreasing in that order) and list appropriate acceptance tests to ensure that the material quality and heat treatment tolerances have been met. The bonus for a higher quality level is higher design endurance limits. It is considered that at least level MQ should be specified for marine main propulsion gears.

Manufacturing

Detailed inspection and test plans and inspection check lists, an integral part of any QA system, are especially important with the number of complex procedures involved in gear manufacture.

For the gear elements it is necessary to determine that the design accuracy grade (Q) and tooth surface finish values are attained. Meshing pinions and wheels need to be carefully matched. Measurements taken should be carefully recorded or, preferably, equipment used that produces a 'print out' of results.

Tooth flank traces, in the radial direction to check profile deviations and in the longitudinal direction to check alignment or helix angle deviation, can readily demonstrate not only the Q grade appropriate for those measurements but also any designed flank corrections.

Where gears are surface hardened, tooth flank hardness measurements should be taken after finish machining and the hardened zone thickness determined by an established method. In this respect it is necessary to have a record of the amount of metal removed from the flanks during final machining so that an actual case depth can be determined.

RELIABILITY CONCEPTS

The previous sections of this paper have discussed the reliability of marine reduction gears by relating the historical records of failure events. The basis for approval of gears for classification purposes, now and in the forseeable future, has been described. Compliance with these Rules should lead to a level of reliability in service which is consistent with the requirement for safe operation. Reliable operation has been achieved as borne out by the infrequent occurence of failures. These Rules, along with most other design methods and standards, are based on a pragmatic, deterministic approach.

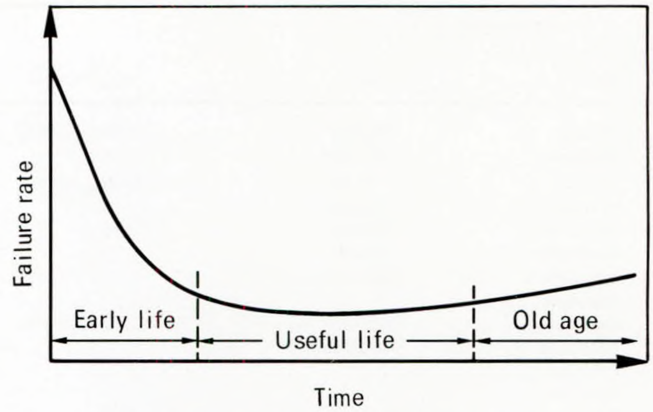
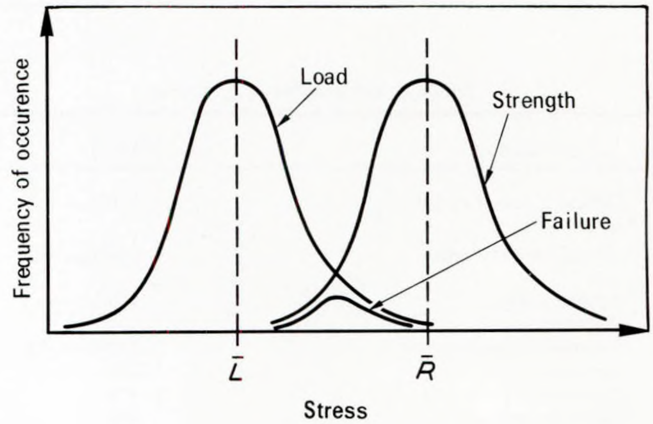


FIG. 5: Temporal variation of failure rate



$$\text{Safety margin} = \frac{\bar{R} - \bar{L}}{\sqrt{\sigma_R^2 + \sigma_L^2}}$$

$$\text{Loading roughness} = \frac{\sigma_L}{\sqrt{\sigma_R^2 + \sigma_L^2}}$$

FIG. 6: Basic failure model

In common with many standard-making organisations, LRS has taken a keen interest in the development of design Rules based upon a direct evaluation of structural reliability.[9,10] The modelling of loads and strengths for machinery items such as gears is complex and presents many problems but it is appropriate to present some of the basic ideas as an indicator of possible future developments.

Before progressing with the application of structural reliability concepts to gearing, it is worthwhile emphasising that it is not only the overall reliability that is of concern. The temporal variation over the working life must also be considered. Typically the failure pattern will approximate to the well known bath-tub curve shown in Fig. 5. In the early life the failure rate is relatively high: an examination of the records previously described suggests that many failures of marine gears which result in replacement fall into this category, being caused by installation or production inadequacies. Once the running-in period has eliminated the items which are sub-standard, there is a period of relatively constant, low failure rate which constitutes the useful life.

In the case of gears, wear-out is not apparently a serious problem, since the design is based on the appropriate fatigue limit, and the final phase of the bath-tub curve need not be considered. The evidence suggests that failures which occur during the useful-life phase are due to foreign object damage, for instance, or to external system changes leading to gross overloads and, effectively, a new 'infant mortality' phase under changed operating conditions (for example clutch damage or misalignment). If the deviation from design intent, poor design and manufacturing or assembly errors are reduced by appropriate checks and procedures, it becomes evident that the overall reliability becomes a function of the random processes occuring during the useful-life phase.

For the useful-life situation it is possible to base design rules on a statistical interference model of loads and strengths, where for gears the strength model is concerned with surface or bending fatigue. It is argued that a model of loads could be constructed by utilising the results from measurement or from calculation using a realistic range of off-design conditions. From the resulting model on the assumption of normal distribution, Fig. 6, the safety margin can be derived by standard methods. It is not intended to elaborate on the procedures involved but to discuss the implications of using such an approach.

First the load model can be constructed to give a realistic evaluation of the maximum load which will occur in service, to a given confidence level. This avoids the need to use pragmatic load factors in an arbitrary cumulative manner. In combination with a model of the appropriate strength, the evaluation gives a direct estimate of reliability. However, the difficulty of establishing the load models and the requisite level of reliability should not be underestimated.

Secondly, account should be taken of the ideas of intrinsic reliability and loading roughness proposed by Carter.[11] The idea of a useful life period which is essentially failure-free is very appealing when considering gears. Loading roughness is a measure of the severity of loading and, like the safety margin, is a function of the statistical representation of the loads and strengths. The theoretical failure rate is related to both the loading roughness and the safety margin. At some value of safety margin, for a given loading roughness, the failure rate falls to a low value, effectively equal to zero, corresponding to a condition of intinsic reliability.

Some basic work has been carried out to establish whether it is possible to employ such a basis for gearing rules and the results have been encouraging.

CONCLUDING REMARKS

The standardisation of calculation methods for the determination of gear load capacity can only be of advantage to all parties concerned. It is, however, to be expected that both manufacturers and regulating bodies will develop calculation methods for individual factors in line with their own experience or tests where they consider this justifiable. The ISO calculation methods lend themselves to this approach particularly well and it is expected that their adoption will increase. Indeed for the application under discussion the International Association of Classification Societies (IACS) are considering the development of 'unified' gearing rules based on the ISO DIS 6336.

Turning to manufacture, the importance of a strong link between design and production through a QA department has already been emphasized. Machine tool equipment available today enables high accuracy to be attained and monitored during manufacture and where specified to be measured and recorded on completion. Economics, however, place constraints which must be recognised by all parties concerned. It is therefore necessary to agree and specify what tolerance grade is applicable and what is to be measured for each particular order. In this respect, namely the grouping and tolerance grading of measurements determined by importance to an application, the concept of tolerance families is a helpful development. An example already available is in the DIN standard 3961 (1978).[12] It is expected that a revised ISO 1328 will also contain a section dealing with tolerance families.

Looking to the future, there is the development of fully integrated CAD/CAM systems for the machining and measuring of gears with topological tooth flank modifications. Such developments can only lead to higher guaranteed accuracy with less direct involvement of inspection but more precise and meaningful records. Further, an increase in reliability may be achieved or higher tooth load capacity justified.

Throughout all of these considerations the reliability in service must reflect the requirement of the industry. The historic performance has been good and this must be maintained when changes are introduced. Analysis of information derived from service experience is essential for the elimination of problems and for the evaluation of the performance of analysis procedures.

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APPENDIX I

Symbol (based on ISO 701–1986)

Symbol	Description	SI unit
a	Centre distance	mm
b	Facewidth	mm
d	Reference diameter	mm
suffix	1 refers to the pinion	
suffix	2 refers to the wheel	
u	Gear ratio (no. teeth in wheel/no. teeth in pinion)	
F_t	Nominal tangential tooth load	N/mm
z	Number of teeth	
m_n	Normal module	mm
α_n	Normal pressure angle at reference diameter	degree
β	Helix angle at reference diameter	degree
v	Linear speed at pitch circle	m/s
ε_α	Transverse contact ratio	
ε_β	Overlap ratio	
σ_y	Yield or 0.2% proof stress	N/mm^2
σ_B	Ultimate tensile strength	N/mm^2
Q	Accuracy grade from ISO 1328(1975)	
C_γ	Tooth mesh stiffness (mean total mesh stiffness per unit facewidth)	N/mm µm
R_a	Surface roughness - arithmetrical mean deviation (CLA) as determined by an instrument having a minimum wavelength cut-off of 0.8mm and for a sampling length of 2.5mm	µm
K_A	Application factor	
K_V	Dynamic factor $K_{V\beta}$ for helical gears and $K_{V\alpha}$ for spur gears	
$K_{H\beta}$,$K_{F\beta}$	Longitudinal load distribution factors	
$F_{\beta y}$	Actual longitudinal tooth flank deviation after running in	µm
$F_{\beta x}$	Actual longitudinal tooth flank deviation before running in	µm
y_β	Running in allowance	µm
f_{sh}	Tooth flank misalignment due to wheel and pinion deflections	µm
f_{ma}	Tooth flank misalignment due to manufacturing errors	µm

Symbol (based on ISO 701–1986)

Symbol	Description	SI unit
f_{sho},γ	Intermediary factors for the determination of f_{sh}	
$K_{H\alpha}$,$K_{F\alpha}$	Transverse load distribution factors	
f_{pb}	Maximum base pitch deviation of wheel	µm
y_α	Running in allowance	µm
σ_H	Hertzian contact stress at the pitch circle	N/mm^2
σ_{HP}	Allowable Hertzian contact stress	N/mm^2
Z_H	Zone factor	
Z_E	Material elasticity factor	
Z_ε	Contact ratio factor	
Z_β	Helix Angle factor	
σ_{Hlim}	Endurance limit for Hertzian contact stress	N/mm^2
Z_R	Surface finish factor	
Z_V	Velocity factor	
S_{Hmin}	Minimum factor of safety for Hertzian contact stress	
F_β	Total tooth alignment deviation (maximum value specified)	µm
σ_F	Bending stress at tooth root	N/mm^2
σ_{FP}	Allowable bending stress at the tooth root	N/mm^2
Y_F	Tooth form factor	
Y_S	Stress concentration factor	
q_S	Notch parameter	
Y_β	Helix angle factor	
Y_{ST}	Stress correction factor	
$Y_{\delta relT}$	Relative notch sensitivity factor	
Y_{RrelT}	Relative surface finish factor	
Y_X	Size factor	
Y_D	Design factor	
S_{Fmin}	Minimum factor of safety for bending stress	
σ_{Flim}	Endurance limit for bending stress	N/mm^2
H_V	Vickers hardness	