Practical Analysis of Highly-Loaded Gears by Using the Modified-Scoring Index Calculation Method

by
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Introduction

The power of high speed gears for use in the petrochemical industry and power stations is always increasing. Today gears with ratings of up to 70,000 kW are already in service. For such gears, the failure mode of scoring can become the limiting constraint. The validity of an analytical method to predict scoring resistance is, therefore, becoming increasingly important.

A simplified calculation procedure suitable for high power, high speed gearing and based on the Winter and Michaelis integral temperature method is presented.

Scoring itself can be described as a momentary flashing of the oil film because of high loads and high speeds. Flashing of the oil film results in metal-to-metal contact and instantaneous welding in small local spots. These small welds are torn apart as fast as they are formed. The sliding action of the teeth causes the torn-out weld to be dragged across the mating surface, creating a gouge or score mark; hence, the name failure mode scoring. The typical appearance of a scored test gear is shown in Fig. 1.

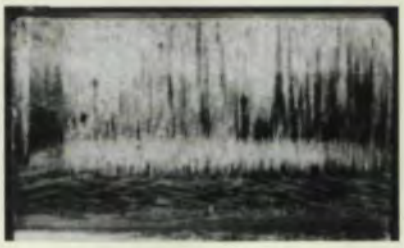

Fig. 1 – Carburized test gear with scoring failure.

The pitting resistance of gear teeth increases by the square of the hardness. In a given gear set, a change in hardness from 300 Brinell Hardness Number to Rockwell C 57 increases the pitting resistance by over 200%. Pitting failures, therefore, are unlikely in RC 57 gearing.

In contrast to this, the resistance to breakage of a gear tooth increases with the first power of the hardness. Therefore, a gear set of 300 BHN that has balanced pitting characteristics and bending strength becomes unbalanced at RC 57.

The size of the teeth must be increased to gain the additional strength required in bending. For this reason, hardened and ground gears characteristically have larger teeth (lower tooth numbers) than their through-hardened predecessors.

The scoring resistance of the gear teeth is not significantly affected by hardness. Sliding velocity, surface finish, oil film anti-weld agents (copper and silver plating, or extreme pressure (EP) additives in the oil) have a much greater influence on this mode of failure.

Gear diameter and tooth size affect the tangential and sliding velocities during the tooth engagement cycle. Tangential velocity, in turn, affects the thickness of the elastohydrodynamic oil film, while sliding velocity affects the heat generated therein. Good gear tooth action results when the peripheral velocity is high enough to produce an oil film thick enough to keep the asperities of the tooth surfaces from touching, and when, at the same time, the sliding velocities are low enough to avoid the generation of excessive heat and flashing of the oil. Therefore, the rise with regard to scoring is of greater importance in using case carburized gears than in using through-hardened gears.

Scoring Calculation Methods

Two early calculation methods were established by J.O. Almen and J.C. Straub in the mid 1930's. They used the PV and PVT factors. P, V and T represent contact stress, sliding

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velocity and distance.

During the late 1930's, H. Blok of Delft Technical Institute developed a flash temperature theory which was based on the conversion of friction energy to heat and, in turn, to a local peak temperature.

Later, Bruce W. Kelley, Caterpillar Tractor Company, described certain modifications to the Blok approach required to correlate test and field experience, notably the surface roughness. This information was presented in AGMA 219.04, 1953.

An allowable specific load intensity expressed in terms of the tangential load per unit of face per unit of diameter was developed by G. Niemann, Technical University, Munich, in 1960. While this criterion had certain usefulness in low speed gears, it was too pessimistic for gears with pitch line velocities over 20 meters per second (m/s).

In 1962, Darle W. Dudley[5] presented an equation for a scoring criterion number above which scoring might be encountered and suggested more elaborate analyses be made. AGMA Standard 217.01,[2] 1965, contains an adaptation of the Kelley/Blok work, especially tailored to aerospace spur and helical gears, and also contains the Dudley scoring index as a simplified quick check for aerospace type gears.

EHD film thickness criteria were used for several years, but because they were designed mainly for mineral oils with a high portion of additives, they do not work correctly. Also, the real oil-pressure, temperature and roughness in the oil film is not known.

More recently, at the Technical University of Munich, analytical and experimental work under the direction of H. Winter has resulted in further refinements of the Kelley/Blok work. This information was presented in AGMA Technical Paper P 219.17,[8] 1983, and is called the integral temperature method. This method is based on a mean integrated tooth flank temperature in contrast to a local peak temperature as used by Blok. Additional refinements include consideration of tooth geometry effects (profile and addendum modification, gear ratio, length of the line of action and tooth size), surface coatings, EP additives, material effect, surface roughness, etc. Since the integral temperature method is generalized approach applicable to many types of gears (spur, bevel, single and double helical) and to a broad range of sizes and speeds, a large number of influencing factors have been included in the calculations. This improves the accuracy of the method, but also makes it relatively complicated to work with.

Integral Temperature Method

The integral temperature method developed by Winter and Michaelis has been adopted as DIN/ISO 3990, Part 4.[4] With this method, a mean integrated tooth flank temperature is determined and compared to an allowable value established from gear tests using the specific lubricant involved. Mean values are used for the coefficient of friction and for the load distribution.

The mean flash temperature calculated is multiplied by empirical factors and added to the bulk temperature, T_M.

$$T_I = T_M + 1.5\ T_{Flm} \tag{1}$$

This mean tooth flank temperature, T_I (also called the integral temperature), corresponds to a measurable tooth flank temperature which could be verified by thermocouples.

The bulk temperature, T_M, can, in turn, be determined from the oil inlet temperature and is added to a percentage of the mean flash temperature.

$$T_M = 1.2\ T_{oil} + 0.\ 84\ T_{Flm} \tag{2}$$

The factors 1,5, 1,2, and/or 0, 84 result from the adaptation of the measured flank and/or bulk temperature to the calculation method and are to be considered constant parameters. Thus, the mean tooth flank temperature becomes:

$$T_I = 1.2\ T_{oil} + 2.34\ T_{Flm} \tag{3}$$

The mean flash temperature, T_{Flm}, is dependent on the geometry, the material, the speed, and the load parameters.

$$\left[T_{Flm} + \mu_B\ \frac{X_M X_{BE}}{X_{Ca} X_Q}\, X_\epsilon\, \frac{W^{3/4}\ V^{1/2}}{a^{1/4}}\right] \tag{4}$$

where

- μ_B = mean coefficient of friction
- X_M = material factor
- X_{BE} = factor for geometry, Hertzian pressure and sliding velocity at the pinion tip
- X_{Ca} = factor for tip relief
- X_ϵ = contact ratio factor
- X_Q = rotation factor.

Mean values for these influencing factors, diagrams, and/or calculation formulae are given in DIN/ISO 3990.

The factor of safety against scoring, S_{SI}, then results in

$$S_{SI} = \frac{T_{SI}}{T_I} \tag{5}$$

These temperature calculations are based on the flash temperature T_{FL} according to Blok. The contact temperature then is determined by

$$T_C = T_M + T_{FL} \tag{6}$$

and the flash temperature is given by

$$T_{Fl} = 2.52\ (\mu)\left[\frac{F_{bt}}{b}\right]^{3/4}\left[\frac{n}{60}\right]^{1/2}\left[(R_{(1)})^{1/2} - (\frac{R_2}{u})^{1/2}\right]/R^{1/4} \tag{7}$$

- μ = gear ratio
- n = pinion speed, rpm
- F_{bt} = tangential tooth load, N
- b = face width, mm
- μ = local friction coefficient
- R_1, R_2 = radii of curvature, mm
- R = $\frac{R_1\ R_2}{R_1 + R_2}$ relative radius of curvature, mm.

Utilizing this equation, contact temperature curves, T_C, can be plotted as a function of the line of contact (Fig. 2). For the integral temperature calculation method, a large

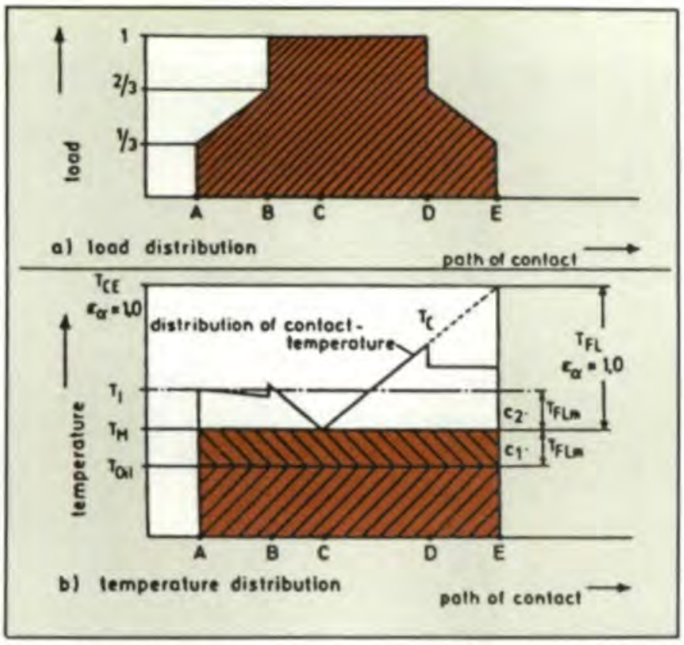

Fig. 2 – Assumptions for integral temperature calculation: load distribution and temperature distribution at contact line.

number of gear tests were used to determine the allowable temperature T_{SI} according to formula 5.

Modified-Scoring-Index Calculation

Based on the integral temperature calculation, a method was developed which can be used for typical applications such as high-speed units in turbomachinery[1] or as shown below for turbine-marine gearing. For the range of applications certain constants were introduced.

Equation 4, as developed by Winter and Michaelis and as adopted by DIN/ISO, is relatively complex and is usually handled in a computer program. One of the attractive aspect of the well-known Dudley scoring index, or AGMA 217.01, is its simplicity and the ease with which it can be determined with a hand-held calculator.

The range of the variables in Equation 4 when applied to high speed turbogearing, is shown first. The more or less standard practices followed in the design, manufacture and operation of high quality turbogearing with respect to tooth geometry, contact ratio, profile modifications, surface finish and lubricant permit assigning constant values to the following variables:

μ_B	coefficient of friction	$\mu_B = 0.03$
X_M	material factor	$X_M = 50$
X_{Ca}	tip relief factor	$X_{Ca} = 1.15$
X_Q	rotation factor	$X_Q = 1.00$

The resulting simplification to Equation 4 is an easy-to-use calculation procedure expressly applicable to high speed gearing. The product of the terms X_{BE} and X_ε and is equated to a geometry factor X_{GEO}, accounting for the number of teeth, contact ratio, radii of curvature, and ratio, and constant values for the terms μ_B, X_M, X_{Ca}, and X_Q are substituted in Equations 3 and 4. The result is:

$$T_{IM} = 1.2\ T_{oil} + X_{GEO}\ SI_M \tag{8}$$

SI_M = Modified Scoring Index

$$SI_M = \frac{3\ w^{3/4}\ v^{1/2}}{a^{1/4}} \tag{9}$$

The load, w, is the tangential force per unit face width.

$w = F_{bt}/b$, N/mm
v = circumferential speed, m/s
a = center distance, mm

For example, the load, w, should include all overload effects from non-uniform load distributions and/or from actual operating conditions.

As can be easily verified, the modified scoring index, SI_M, Equation 9, is similar in form to the AGMA 217 scoring index. For example:

$$\text{AGMA Scoring Index} = \frac{W^{3/4} n_p^{1/2}}{P_d^{1/4}} \tag{10}$$

W = tangential force per unit face width
n_p = pinion speed, rpm
P_d = diametral pitch.

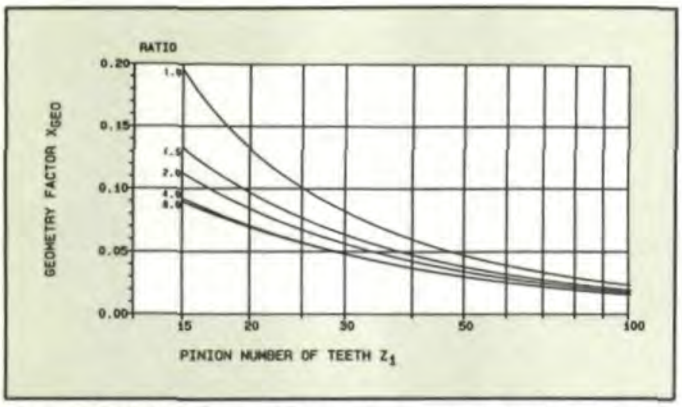

Fig. 3 – Modified Scoring Index Method: Geometry factor for helix angle of 27.5° and addendum and modification factor $X_1 = 0$.

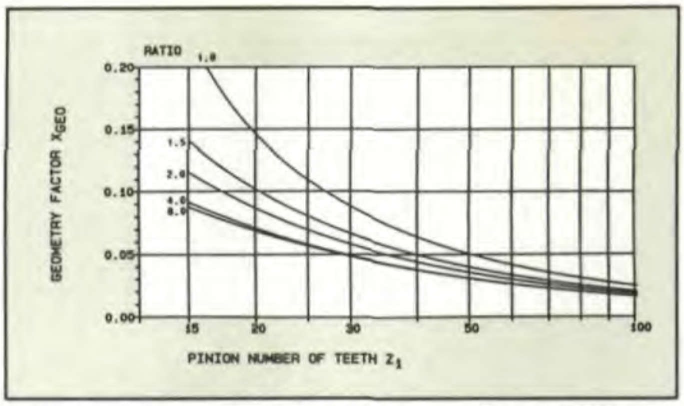

Fig. 4 – Modified Scoring Index Method: Geometry factor for helix angle of 27.5° and addendum modification factor $X_1 = 0.2$.

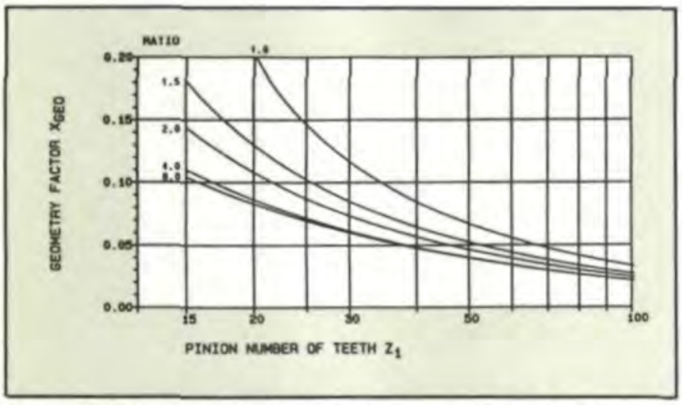

Fig. 5 – Modified Scoring Index Method: Geometry factor for helix angle of 27.5° and addendum modification factor $X_1 = 0.5$.

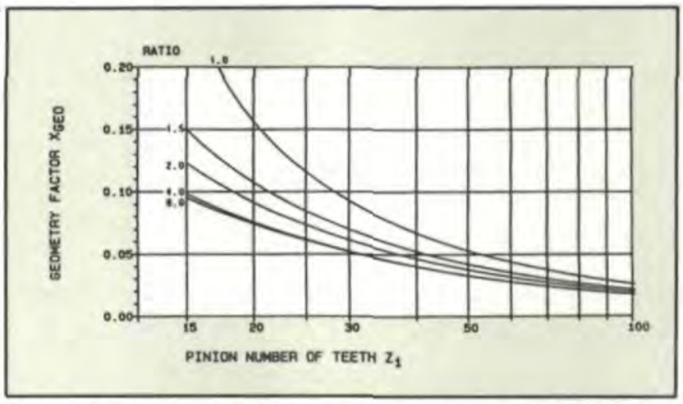

Fig. 6 – Modified Scoring Index Method: geometry factor for helix angle of 10° and addendum modification factor $X_1 = 0$.

The geometry factor, X_{GEO}, in Equation 8 considers the influence of tooth numbers, ratio, and addendum proportions in addition to tooth size. Geometry effects in Equation 10 are limited to tooth size only. Diagrams of X_{GEO} for teeth with zero-addendum modification or zero sum of addendum modification are shown in Figs. 3 through 8.

These diagrams are calculated for the typical high speed or marine turbine gearing:

a) single helical with helix angle of appr. 10°,
b) double helical with helix angle of appr. 27.5°,

and "normal" addendum modification factors 0, 0.2 and 0.5 at the pinion.

Furthermore, common European practice is to have the sum of modification factors of pinion and gear near zero. Similarly to the integral temperature calculation, the safety margin for the modified method can be found by:

$$S_{SIM} = \frac{T_{IS}}{T_{IM}} \tag{11}$$

Using Equation 8 and allowable values of T_{IS} from Equation 4, or Table 1, the safety margin for scoring resistance can be determined.

An adequate margin of safety exists when the S_{SIM} value of Eequation 11 is 1,6 or greater. As a quick design check, the permissible modified scoring index, SI_{MP}, can be used from Table 1. The values shown include normal safety factors and are based on average gear geometry.

Applications of Modified Scoring Index Method

The following will show how the different constant figures for this modified method were determined with regard to high speed and turbine marine gearing.

Coefficients of Friction

A curve of the coefficient of friction as a function of the pitch line velocity is shown in Fig. 9. Surface roughness, Ra, must be 0.5 μm or better (after running-in). The curves shown are believed to be flat beyond 70 m/s. A value of $\mu_B = 0.03$ has been taken for high speed gears. For pitch line velocities below 30 m/s, a slightly larger value should be used. The points in Fig. 9 are marking typical applications for high speed and marine gearing.

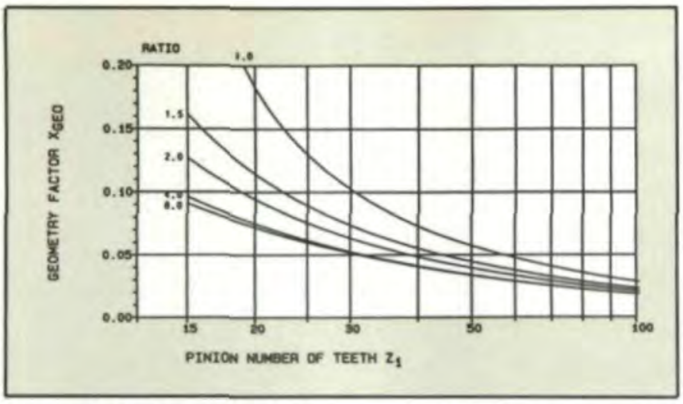

Fig. 7 – Modified Scoring Index Method: geometry factor for helix angle of 10° and addendum modification factor $X_1 = 0.2$.

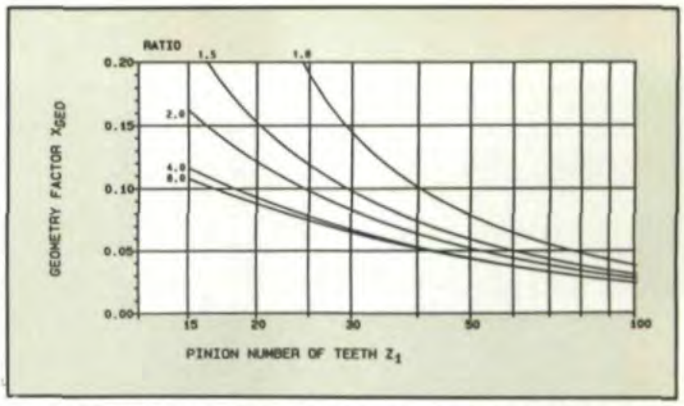

Fig. 8 – Modified Scoring Index Method: geometry factor for helix angle of 10° and addendum modification factor $X_1 = 0.5$.

Table 1 Allowable Temperature T_{IS}, Loads and Indices for Several Commonly Used Oils

Oil Type	ISO Viscosity grade SAE Viscosity grade	FZG-Test Load-stage	Ryder Gear Test, ppi app.	T_{IS}	Permissible Modified Scoring Index SI_{MP}
Turbine-Oil	ISO VG 46	6	1,500	160°C	900
Special Turbine Oil	ISO VG 46	7*	2,200	180°C	1,150
Turbine-Oil	ISO VG 100	7*	2,200	180°C	1,150
Motor-Oil	SAE 30	>8	>3,000	>200°C	>1,400
		9	4,000	235°C	1,850

*If pinion and/or wheel copper plated one to two load-stages higher

Tip relief factor.

With optimal profile modification and normal contact ratios, tip relief factor X_{Ca} can be assigned a value of 1.15.

Addendum Proportions.

When addendum proportions are kept within the range of standard to 50% long and short, the rotation factor, X_Q, can be taken as unity.

A standard addendum set is defined as having mating pinion and gear with addendums equal to 100% X Module, mm. The maximum departure from the standard addendum is 150% x Module (driver) mating with 50% x Module (driven). The summation addendum lengths shorter and longer than standard should equal 100%.

Material Factor, X_M, is considered a constant of 50 for normal steel gears.

Lubrication.

Pressure fed lubrication by spray jet is mandatory.

Typical Examples.

The following three groups of practical examples will be analysed by the exact Integral Temperature Method and the Modified Scoring Index Method. Tables 2 to 4 show the main technical data of these three gear groups, which are

a) High speed gears for powers of 7,300 kW to 70,000 kW, Table 2,

Table 2 Technical data, load and oil quality of typical examples of high speed gears

Gearing No.	Power P kW	Ratio i	Pitch line velocity v m/s	Unit load K $\cdot(F_{bt}/b)$ N/mm	Oil type	ISO Viscosity grade (SAE)	FZG-Test DIN 51 354 Load stage	Plant
1	70,000	1.2	137	1,860	Special Turbine-Oil	ISO VG 46	(7) 9	Gas Turbine/Generator Pinion copper plated
2	19,500	5.8	116	800	Turbine-Oil	ISO VG 46	6	Motor/Compressor
3	7,300	1.9	139	435	Turbine-Oil	ISO VG 46	6	Motor/Compressor

Table 3 Technical data, load and oil quality of typical examples of marine turbine gears

Gearing No.	Power P kW	Ratio i	Pitch line velocity v m/s	Unit load K $\cdot(F_{bt}/b)$ N/mm	Oil type	ISO Viscosity grade (SAE)	FZG-Test DIN 51 354 Load stage	Plant
4	3,500	2.6	71	630	Turbine-Oil	ISO VG 46	6	Steam Turbine/Propeller 1. Stage
5	3,500	5.1	44	640	Turbine-Oil	ISO VG 46	6	Steam Turbine/Propeller 2. Stage
6	19,000	4.6	30	2,770	Turbine-Oil	ISO VG 100	7	Gas Turbine/Propeller
7	3,800	1.1	56	740	Turbine-Oil	ISO VG 100	7	Gas Turbine/Propeller
8	10,250	8.2	35	1,720	Turbine-Oil	ISO VG 100	7	Gas Turbine/Propeller

Table 4 Technical data, load and oil quality of typical examples of marine diesel drives

Gearing No.	Power P kW	Ratio i	Pitch line velocity v m/s	Unit load K $\cdot(F_{bt}/b)$ N/mm	Oil type	ISO Viscosity grade (SAE)	FZG-Test DIN 51 354 Load stage	Plant
9	3,750	1.0	24	1,300	Turbine-Oil	ISO VG 100	7	Diesel-Engine/Propeller
10	7,400	5.3	39	1,500	Motor-Oil	SAE 30	8	Diesel-Engine/Propeller

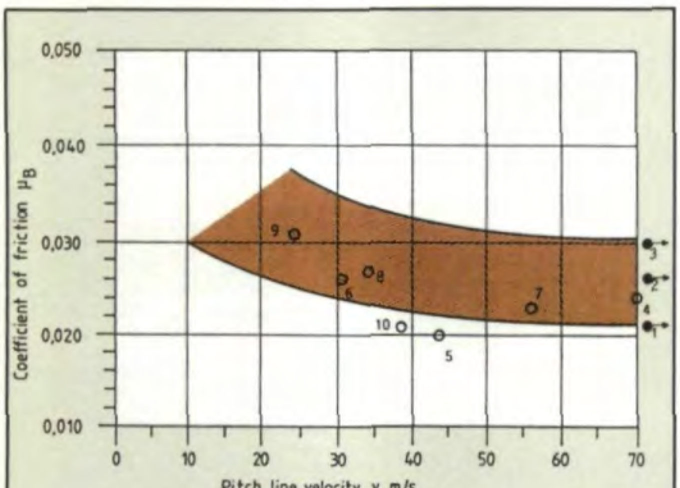

Fig. 9 – Typical calculated coefficients of friction and scattering field for marine and high speed applications.

b) Marine turbine gears for powers of 3,500 to 19,000 kW, Table 3,

c) Marine diesel gears for powers of 3,700 to 7,400 kW, Table 4.

Besides the main geometrical data, loading data such as unit load are also given. For these gears the factor of safety against scoring according to the modified scoring index method versus the safety factor according to integral temperature method is shown in Fig. 10. A good relation can be recognized which proves that the simplied method is working satisfactorily.

Additionally, Fig. 11 gives a relation between the scoring load stage (FZG) and the permissible modified scoring index together with the practical modified scoring index used in the gears. One can see that a certain safety margin exists between the data points and the limit line. All these gears have been

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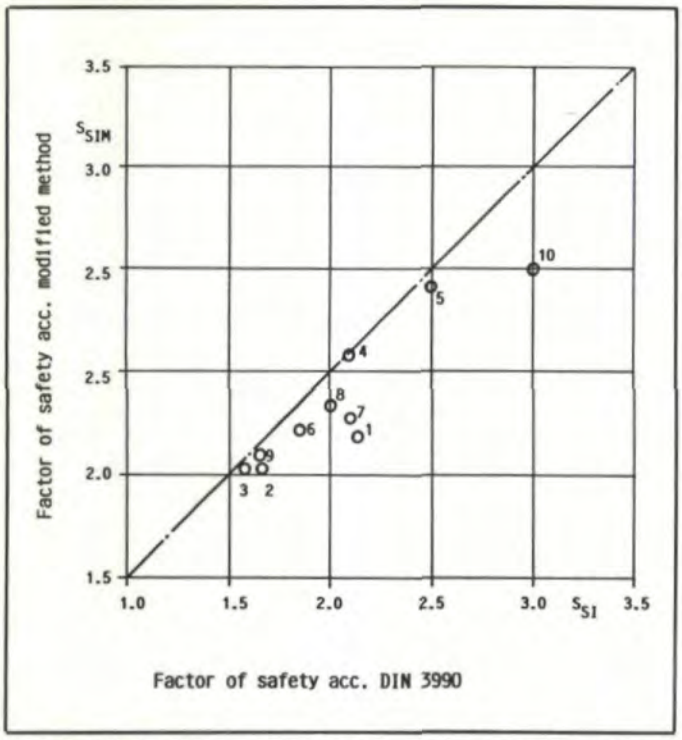

Fig. 10 – Calculated scoring safety factors according Integral Temperature Method versus scoring safety factors according Modified Scoring Index Method.

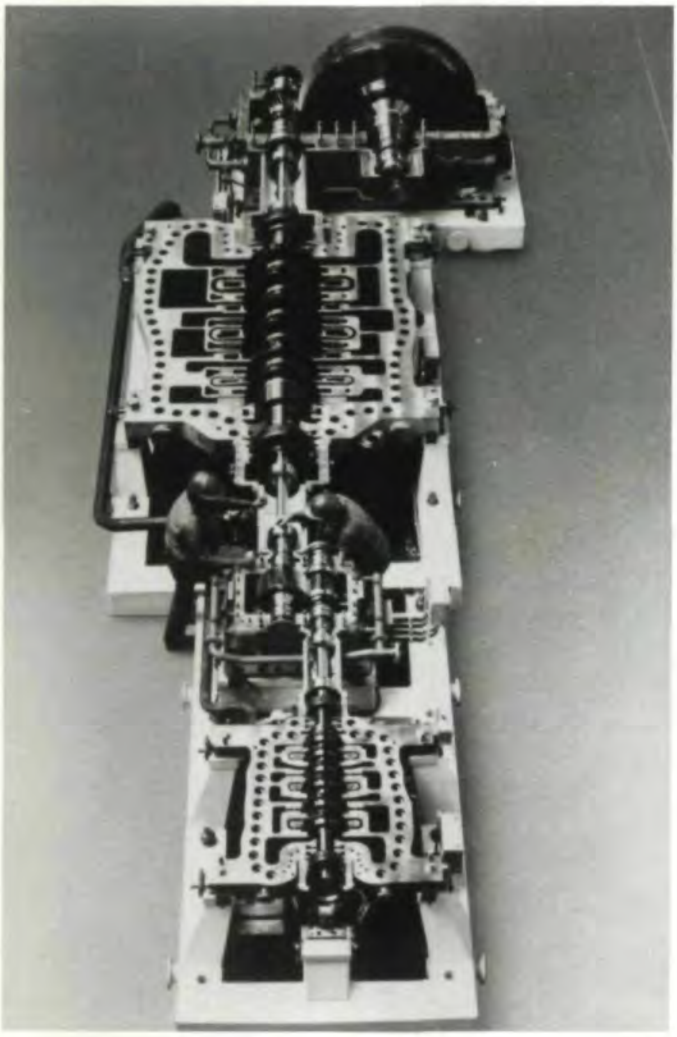

Fig. 12 – Total view of a high- and low-pressure gas compressor unit equipped with high speed gears, Ref. Nos. 2 and 3.

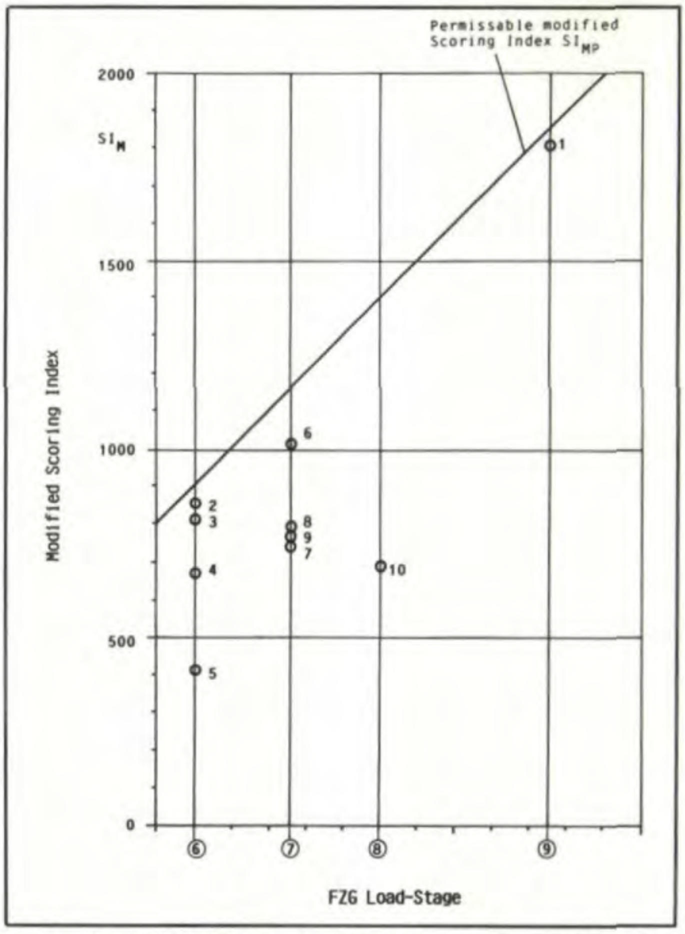

Fig. 11 – Practical Modified Scoring Index values SI_M for typical applications versus allowable SI_M values and FZG load stages.

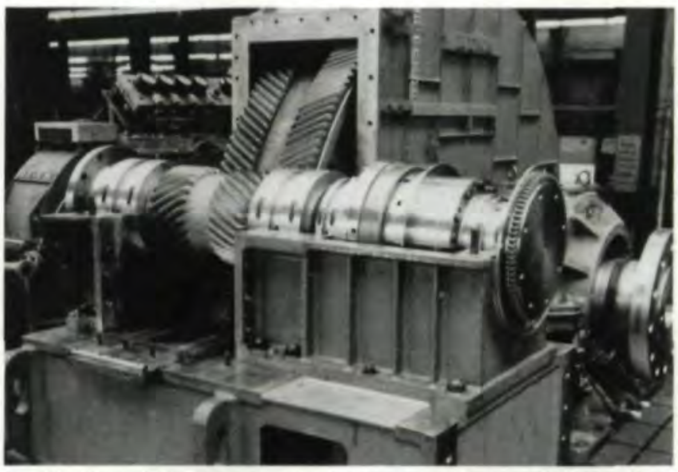

Fig. 13 – Typical Navy gear, main pinion at output stage, Ref. No. 8.

operating for years without scoring problems. Fig. 12 shows a photograph of the gears mentioned in Table 2, reference numbers 2 and 3. Fig. 13 provides an impression of a modern marine gear for a navy ship. As mentioned, these gears proved high scoring capacity under load. Therefore, the application of the modified scoring index seems to be acceptable to calculate the real scoring resistance of high speed and turbine marine gears.

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IMPROVED GEAR LIFE . . .

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PRACTICAL ANALYSIS OF HIGHLY LOADED . . .

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Conclusion

Based on the DIN/ISO formulae for scoring capacity, a simplified method adopting a modified scoring index has been developed. As can be seen from typical applications, this method works with sufficient accuracy.

The calculation of scoring capacity will become more and more important in parallel with an increasing demand in transmitted power per gear volume. The practical experience with highly-loaded gears with regard to scoring will give more safety in the application of this calculation method and will possibly permit a reduction of the safety margins used today.

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GEARS FOR NONPARALLEL SHAFTS . . .

(continued from page 62)

sacrificed to obtain a large mechanical advantage. Typical applications are standby pumps, large valves, and gates.

2. *Intermittent, manual* operations requiring a *large mechanical advantage*, such as in steering mechanisms and opening and closing of valves and gates by means of handwheels (Fig. 14).
3. *Motorized, nearly continuous* operations where worm gearing competes with gear reduction units. When space is at a premium, as in machine tools, packaged, motor-driven worm reduction units are used in preference to gear reducers (Fig. 13). Depending on size and application, the unit may be self-contained or built integrally with an electric motor. Because of silent operation, such units are preferred in machine tools and also in elevators. These units all require multithreaded worms and ratios not exceeding 1:18. Larger ratios are achieved by connecting two units in series.

Design Detail of Worm Gearing

The unit shown in Fig. 15 is a typical, medium-size worm gear speed reducer. Smaller units of this type usually have housings of cast aluminum alloys for maximum thermal rating. For larger units the preferred material is cast iron. The worm is case-hardened and ground alloy steel of integral shaft design. The gear is cast bronze with generated teeth and keyed to the output shaft. Larger worm gears are often composed of a ring of bronze mounted on a center or hub of less expensive material. A common design utilizes a flanged rim mounted on the hub by means of shear bolts (Fig. 16*a*). Equally common is mounting by means of a press fit (Fig. 16*b*) assisted by a pin connection. The output shaft is high-quality, medium-carbon steel, ground to close tolerances. The worms and output shafts are frequently mounted on roller bearings. All shaft extensions are equipped with lip style, synthetic oil seals.

Lubrication

Generally, oil is contained within the housing and directed by splash to the bearings and to the zone of tooth and thread contact. Natural splash may be augmented by flingers, scrapers, and cups attached to the gear. Channels or ribs may be furnished inside the housing to help direct oil to the bearings.

Summary

Despite higher initial cost, gears for nonparallel shafts are justified because they often save space and lead to a better design. Kinematically, these gears all perform the very difficult task of changing the plane of rotation. With the exception of crossed helical gears, all have reached a high degree of perfection and a long, useful life of transmitting power. Hypoid gears for automotive differentials, for instance, rarely fail during the life of a car. The versatility of worm gearing is due to the inverse relationship of efficiency to torque and reduction ratio. Table 1 summarizes comparative characteristics of speed reducer gear families.

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