Scoring Load Capacity of Gears Lubricated With EP-Oils

by
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Abstract

The Integral Temperature Method for the evaluation of the scoring load capacity of gears is described. All necessary equations for the practical application are presented. The limit scoring temperature for any oil can be obtained from a gear scoring test. For the FZG-Test A/8.3/90 acc. DIN 51 354 and the Ryder Gear Test acc. FTM STD Nr. 791, graphs for the direct evaluation of the scoring temperature as a function of oil viscosity and test scoring load are given.

The method is compared with the Total Contact Temperature Criterion acc. Blok (1)—the alternate procedure to the Integral Temperature Method as standardized in ISO DP 6336 part IV—and the Scoring Index Method acc. Dudley (2). Comparative calculations for practical gears with and without scoring damages showed good correlation with experience for the Integral Temperature Criterion.

Introduction

In different fields of application, the load carrying capacity of gears is limited by scoring damage.

In highly loaded, case carburized turbine gears, the normally used mineral oils with rust and oxidation inhibitors do not always give sufficient scoring protection. On the other hand, the necessary EP-additives adversely affect the anti-oxidation, anti-foam, etc. properties, so that the life of the oil may be reduced.

In the case of carburized marine gears with diesel engine drives, motor oils are frequently also used for the gears. These oils do not always provide sufficient scoring load capacity.

Also, in some types of locomotive drives, the same lubricant is used for the hydraulic torque converter and the gears. For high efficiency of the hydraulics, low viscosity oils have to be used. Because of then reduced film thickness between the gear flanks, EP-additives have to compensate for viscosity.

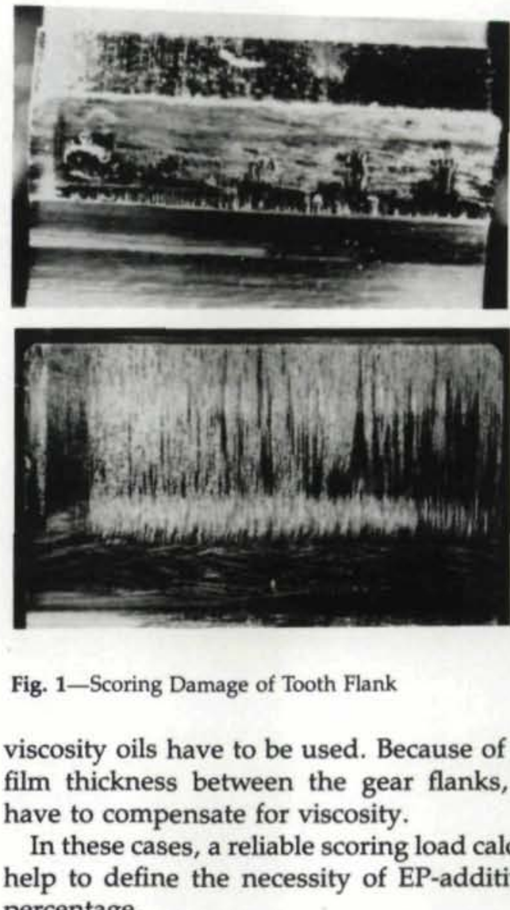

Fig. 1—Scoring Damage of Tooth Flank

In these cases, a reliable scoring load calculation could help to define the necessity of EP-additives and their percentage.

The type of damage occuring in the range of medium to high speed gears is the so called "warm" scoring (Fig. 1), which is covered by this paper. "Cold" scoring, which can be observed in the area of low speed, low quality, through hardened gears of low hardness, has to be handled with some different method.

Integral Temperature Method

Principle

Derived from hundreds of tests with different gear oils, different gear geometries, materials, operating speeds, temperatures, etc. in back-to-back gear test rigs of center distances a = 91.5; 140 and 200 mm, a mean surface temperature on the engaging flanks has been established as a governing criterion of the scoring damage.

For an assumed load distribution along the path of contact as shown in Fig. 2, the flash temperature distribution acc. Blok (1) can be calculated. The sum of the

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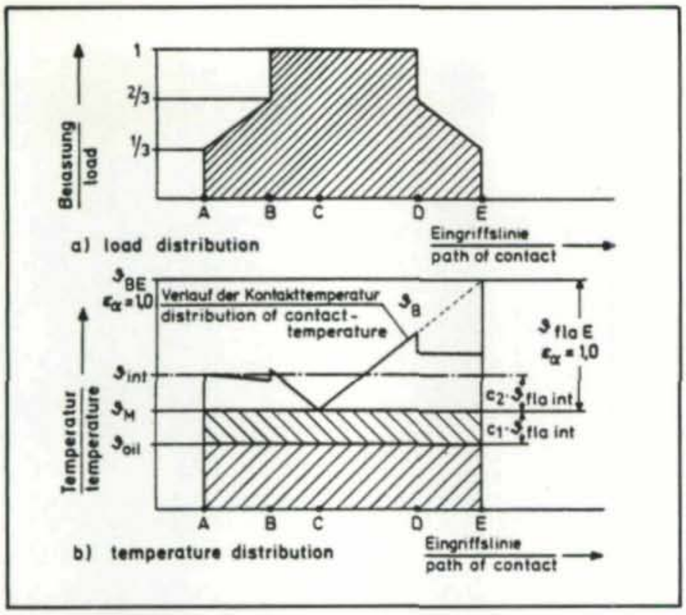

Fig. 2—Load and Temperature Distribution acc. Blok (schematic)

mean flash temperature (multiplied by a weight factor) and the gear bulk temperature is defined as the integral temperature. The weight factor accounts for the certainly differing influences of the real bulk temperature and the mathematically established mean flash temperature on the scoring damage (3).

The integral temperature of a practical gear must not exceed a critical value which is independent of the operating conditions and constant for a given material-lubricant combination. This limiting value, the scoring temperature, can be calculated according to the same set of equations introducing the parameters of any gear scoring test of the oil under consideration. If both, the actual gear and the test gear, differ in material or heat treatment, empirical correction factors have to be introduced.

Integral Temperature Rating

The integral temperature is calculated in the transverse section of the gear pair.

*) $$\vartheta_{int} = \vartheta_M + C_2 \cdot \vartheta_{fla\,int} \qquad (1)$$

*) For symbols and units see table 1.

**) For gears with internal power distribution (e.g. planetary gears) a non uniform load distribution has to be considered. In these cases $F_t\ K_A$ has to be replaced by $F_t\ K_A\ K_\gamma$.

***) The evaluation of a load quotient can be approximated by

$$S_{SL} = \frac{W_{t\,max}}{W_{t\,eff}} \approx \frac{\vartheta_{S\,int} - \vartheta_{oil}}{\vartheta_{int} - \vartheta_{oil}}$$

The weight factor, as described above, has been determined from test results $C_2 = 1.5$.

The mean flash temperature, $\vartheta_{fla\,int}$, can be approximated by the determination of the flash temperature at the tip of the pinion, $\vartheta_{fla\,E}$, for a contact ratio, $\epsilon_\alpha = 1.0$ (no load sharing) and the contact ratio factor X_ϵ (see Fig. 2).

$$\vartheta_{fla\,int} = \vartheta_{fla\,E} \quad X_\epsilon \qquad (2)$$

The nominal flash temperature, $\vartheta_{fla\,E}$, at the pinion tip is calculated acc. Blok (1)

**

$$\vartheta_{fla\,E} = \mu_B \cdot X_M \cdot X_{BE} \cdot X_{\alpha\beta} \frac{\left(\frac{F_t}{b} \cdot K_A \cdot K_{B\beta} \cdot K_{B\alpha l} \cdot K_{B\gamma}\right)^{3/4} \cdot v^{1/2}}{|a|^{1/4} \cdot X_Q \cdot X_{Ca}} \qquad (3)$$

The scoring temperature is evaluated using the same equations for the conditions of a gear oil test

$$\vartheta_{S\,int} = \vartheta_{MT} + C_2 \cdot X_{W\,rel\,T} \cdot \vartheta_{fla\,int\,T} \qquad (4)$$

The safety factor against scoring damage is defined as a temperature quotient

$$S_S = \vartheta_{S\,int} / \vartheta_{int} \quad *** \qquad (5)$$

From recalculation of practical gears, safety factors, less than unity, refer to a high risk of scoring, while safety factors over 2.0 indicate a low scoring risk. Gears with calculated safety factors between 1.0 and 2.0 are of a borderline type. They can be operated without scoring damage when a good load distribution across the face width, smoothed, run-in surfaces, etc. are obtained. In cases where, e.g., new manufactured flanks without a run-in process are operated under nominal load, scoring can occur.

Influence Factors

The coefficient of friction, μ_B, is calculated as a mean value along the path of contact. It can be approximated by introducing the parameters of the pitch point

$$\mu_B = 0.045 \left[\frac{(F_t/b) \cdot K_A \cdot K_{B\beta} \cdot K_{B\alpha}}{\cos\alpha_{wt} \cdot v_{\Sigma C} \cdot \rho_{Cn}}\right]^{0.2} \cdot \eta_\theta^{-0.05} \cdot X_R$$

$$= \mu_m \cdot (K_{B\beta} \cdot K_{B\alpha})^{0.2} \qquad \text{with } \mu_m \leq 0.2 \qquad (6)$$

$F_t/b = 150$ N/mm is introduced for $F_t/b \leq 150$ N/mm.

Eq. (6) for the evaluation of the coefficient of friction has only been introduced in the DIN standard, not yet in the ISO document. Recent investigations showed a good correlation of μ_m with practical experience and measurements of gear power loss and efficiency (Fig. 3) so that Eq. (6) can also be used for the determination of absolute frictional losses in gears (4).

The overload factors K_A, $K_{B\beta}$ and $K_{B\alpha}$ can be determined acc. ISO DP 6336 Part I for surface durability $K_{H\beta}$ and $K_{H\alpha}$.

The rolling speed on the pitch circle is

$$v_{\Sigma C} = 2 \cdot v \cdot \sin\alpha_{wt} \tag{7}$$

For the speed range v below, 1 m/s and above 50 m/s the evaluation of μ_B becomes uncertain and is no longer based on experimental data. In this range μ_B is assumed to be constant, with $v = 1.0$ m/s for $v < 1.0$ m/s and $v = 50$ m/s for $v > 50$ m/s to be introduced into Eq. (7). The radius of curvature in the normal section is

$$\rho_{Cn} = 0.5 \frac{\tan\alpha_{wt}}{\cos\beta_b} d_{b1} \frac{u}{u+1} \tag{8}$$

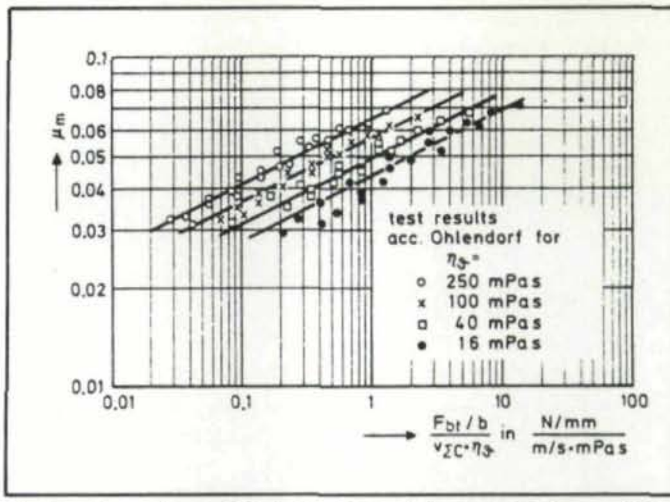

Fig. 3—Comparison of Calculated Coefficient of Friction and Test Results

Table 1: Symbols, Terms, and Units

a	*centre distance*	mm	α	*pressure angle*	°
b	facewidth	mm	β_b	base helix angle	°
c_γ	mesh stiffness	N/(μm mm)	ϵ_1	addendum contact ratio of pinion	–
c′	single stiffness	N/(μm mm)	ϵ_2	addendum contact ratio of wheel	–
C_a	amount of tip relief	μm	ϵ_α	transverse contact ratio	–
$C_{1,2}$	constants	–	ϵ_γ	total contact ratio	–
d	reference diameter	mm	ϑ_B	instantaneous contact temperature	°C
d_b	base diameter	mm	ϑ_{fla}	flash temperature	K
d_{Na}	effective tip diameter	mm	$\vartheta_{fla\,E}$	flash temperature, pinion tip	K
E	Young's modulus	N/mm^2	$\vartheta_{fla\,int}$	mean flash temperature	K
F_t	tangential force, reference circle	N	ϑ_{int}	integral temperature	°C
F_{bt}	tangential force, base circle	N	ϑ_M	bulk temperature	°C
K_A	application factor	–	ϑ_{oil}	oil temperature	°C
$K_{B\alpha}$	transverse load distribution factor	–	$\vartheta_{S\,int}$	scoring temperature	°C
$K_{B\beta}$	logitudinal load distribution factor	–	η_ϑ	dynamic oil vicosity at ϑ_{oil}	mPa s
$K_{B\gamma}$	helical load distribution factor	–	μ_B	coefficient of friction, scoring	–
K_γ	load distribution factor for more than one mesh	–	μ_m	mean coefficient of friction	–
			ν	Poisson's ratio	–
m	module	mm	ν_{40}	kinematic viscosity at 40 °C	mm^2/s
R_a	arithmetic average roughness (CLA)	μm	ρ	radius of curvature	mm
S_B	safety factor, flash temperature	–			
S_S	safety factor, integral temperature	–	Suffixes		
T_1	pinion torque	N m	b	base circle	
u	gear ratio $f_1/f_2 \geqslant 1$	–	C	pitch point	
v	linear speed at reference circle	m/s	eff	effective values	
v_Σ	rolling speed	m/s	E	pinion tip	
w_t	specific load including overload	N/mm	max	maximum	
X_{BE}	geometrical factor, pinion tip	–	n	normal section	
X_{Ca}	tip relief factor	–	t	transverse section	
X_M	thermal flash factor	$K\,N^{-3/4}s^{1/2}m^{-1/2}mm$	T	test gear	
X_Q	rotation factor	–	w	working	
X_R	roughness factor	–	1	pinion	
X_S	lubrication factor	–	2	wheel	
X_W	welding factor	–			
$X_{\alpha\beta}$	angle factor	–			
X_ϵ	contact ratio factor	–			
z	number of teeth	–			

The roughness factor accounts for surface roughness

$$X_R = 3.8\ (R_a/d_1)^{0.25} \tag{9}$$

$$\text{with } R_a = 0.5 \cdot (R_{a1} + R_{a2}) \tag{10}$$

In Eq. (10) the CLA-values of the new manufactured flank have to be introduced. An amount of normal run-in is included in Eq. (9).

The thermal flash factor X_M depends on the elastic and thermal properties of the gear materials. For gears made out of steel, mean values of conductivity $\lambda_M = 50$ N/(s.K); density $\rho_M = 7.85$ kg/dm³, specific heat capacity $c_M = 485$ N m/(kg.K); $E = 206{,}000$ N/mm², and $\nu = 0.3$ can be introduced

$$X_M = 50\ K \cdot N^{-3/4} \cdot s^{1/2} \cdot m^{-1/2} \cdot mm \tag{11}$$

For non steel materials for pinion and/or gear see ISO DP 6336, Part IV.

The geometrical factor X_{BE} takes account for the Hertzian stress and the contact time at the pinion tip E.

$$X_{BE} - 0.5\sqrt{\frac{|Z_2|}{Z_2}(u + 1)}\ \frac{\sqrt{\rho_{E1}} - \sqrt{\rho_{EZ}/u}}{(\rho_{E1} \cdot |\rho_{EZ}|)^{1/4}} \tag{12}$$

$$\text{with } \rho_{E1} = 0.5 \cdot \sqrt{d_{Na1}^2 - d_{b1}^2} \tag{13}$$

$$\text{and } \rho_{EZ} = a \cdot \sin\alpha_{wt} - \rho_{E1} \tag{14}$$

in the transverse section. Eqs. (12, 13, 14) are valid for internal and external cylindrical gears.

The angle factor $X_{\alpha\beta}$ accounts for the recalculation of the acting normal load to the circumferential load at the pitch cylinder.

$$X_{\alpha\beta} = 1.22\ \frac{\sin^{1/4}\alpha_{wt} \cdot \cos^{1/4}\beta_b}{\cos^{1/4}\alpha_t \cdot \cos^{1/2}\alpha_{wt}} \tag{15}$$

For approximate calculations and a pressure angle $\alpha = 20°$, $X_{\alpha\beta}$ can be set unity.

The helical load distribution factor, $K_{B\gamma}$, accounts for the empirical decrease of scoring load capacity for increasing total contact ratio.

$$K_{B\gamma} = 1.0 \qquad \text{for} \quad \varepsilon_\gamma \le 2.0$$

$$K_{B\gamma} = 1 + 0.2 \cdot \sqrt{(\varepsilon_\gamma - 2)(5 - \varepsilon_\gamma)} \ \text{for } 2 < \varepsilon_\gamma < 3.5 \tag{16}$$

$$K_{B\gamma} = 1.3 \qquad \text{for} \quad \varepsilon_\gamma \ge 3.5$$

The rotation factor, X_Q, considers the effect of a simultaneous load impact and high sliding at the beginning of the mesh. For gears with normal addendum modification

$$X_Q = 1.0 \text{ for } 1/1.5 < \varepsilon_1/\varepsilon_2 < 1.5 \tag{17a}$$

$$\text{with } \varepsilon_{1,2} = \frac{|z_{1,2}|}{2\pi} \cdot \sqrt{\left(\frac{d_{Na1,2}}{d_{b1,2}}\right)^2 - 1} - \tan\alpha_{wt} \tag{18}$$

In the case where the approach path of contact of the driving partner exceeds 1.5 times the recess path, X_Q is set 0.6.

$$\begin{aligned} X_Q &= 0.6 \text{ for driving pinion and } \epsilon_2 \ge 1.5\epsilon_1 \\ X_Q &= 0.6 \text{ for driving wheel and } \epsilon_1 \ge 1.5\epsilon_2 \end{aligned} \tag{17b}$$

In all other cases $X_Q = 1.0$.

The tip relief factor X_{Ca} accounts for the benefit of a profile modification in the area of high sliding (Fig. 4) acc. Lechner(5). Tip relief is only effective up to the amount where it compensates tooth deflection under load

$$X_{Ca} = 1 + 1.55 \cdot 10^{-2} \cdot \epsilon_{max}^4 \cdot C_a \tag{19}$$

with ϵ_{max} as the maximum value of ϵ_1 or ϵ_2 acc. Eq. (18)

$$\varepsilon_{max} = \max \begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \end{Bmatrix} \tag{20}$$

The effective tip relief $C_{a\,eff}$ can be approximated by

$$\begin{aligned} C_{aeff} &= F_{bt} \cdot K_A/(b \cdot c') \text{ for spur gears} \\ C_{aeff} &= F_{bt} \cdot K_A/(b \cdot c_\gamma) \text{ for helical gears} \end{aligned} \tag{21}$$

with the stiffness values c' resp. c_γ acc. to ISO DP 6336 part I.

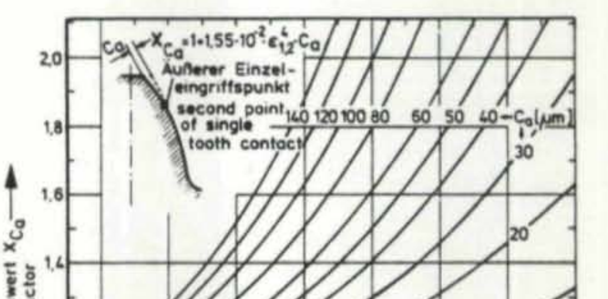

Fig. 4—Influence of Tip Relief

A profile modification increases scoring load capacity only when it is applied to the area of highest risk. In gears with normal addendum modification, the approach path with the load impact of the ingoing mesh is more dangerous. For extreme addendum modification, a tip relief has to be applied to the recess path.

For driving pinion:

$$C_a = \min\begin{Bmatrix} C_{a1} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_1 > 1.5 \cdot \varepsilon_2$$

$$C_a = \min\begin{Bmatrix} C_{a2} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_1 \leq 1.5 \cdot \varepsilon_2 \qquad (22a)$$

For driving gear:

$$C_a = \min\begin{Bmatrix} C_{a2} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_2 > 1.5 \cdot \varepsilon_1$$

$$C_a = \min\begin{Bmatrix} C_{a1} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_2 < 1.5 \cdot \varepsilon_1 \qquad (22b)$$

The contact ratio factor, X_ε, recalculates a mean flash temperature along the path of contact from the maximum temperature, $\vartheta_{fla\ E}$, at the pinion tip for $\epsilon_\alpha = 1.0$. The equations are valid for a load distribution acc. (Fig. 5) and an approximately linear increase of the flash temperature towards the tooth tip and tooth root (Fig. 2).

For $\varepsilon_\alpha < 1.0$:

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1}\left(\varepsilon_1^2 + \varepsilon_2^2\right) \qquad (23a)$$

For $1 \leq \varepsilon_\alpha < 2.0$: (23b)

ε_1 and $\varepsilon_2 < 1.0$

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1}\left[0.7(\varepsilon_1^2 + \varepsilon_2^2) - 0.22 \cdot \varepsilon_\alpha + 0.52 - 0.6\varepsilon_1 \cdot \varepsilon_2\right]$$

ε_1 or $\varepsilon_2 \geq 1.0$

$$X_\varepsilon = \frac{1}{2\varepsilon_\alpha \cdot \varepsilon_1}\left[(0.18\varepsilon_{1,2})^2 + (0.7\varepsilon_{2,1})^2 + 0.82\varepsilon_{1,2} - 0.52\varepsilon_{2,1} - 0.3\varepsilon_1\varepsilon_2\right]$$

with the first index for $\epsilon_1 \geq 1.0$ and the second index for $\epsilon_2 \geq 1.0$.

(23c)

For $2.0 \leq \varepsilon_\alpha < 3.0$ and ε_1 and ε_2 less than 2.0

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1}\left[(0.44\varepsilon_{1,2})^2 + (0.59\varepsilon_{2,1})^2 + 0.3\varepsilon_{1,2} - 0.3\varepsilon_{2,1} - 0.15\varepsilon_1 \cdot \varepsilon_2\right]$$

with the first index for $\epsilon_1 > \epsilon_2$ and the second index for $\epsilon_2 > \epsilon_1$.

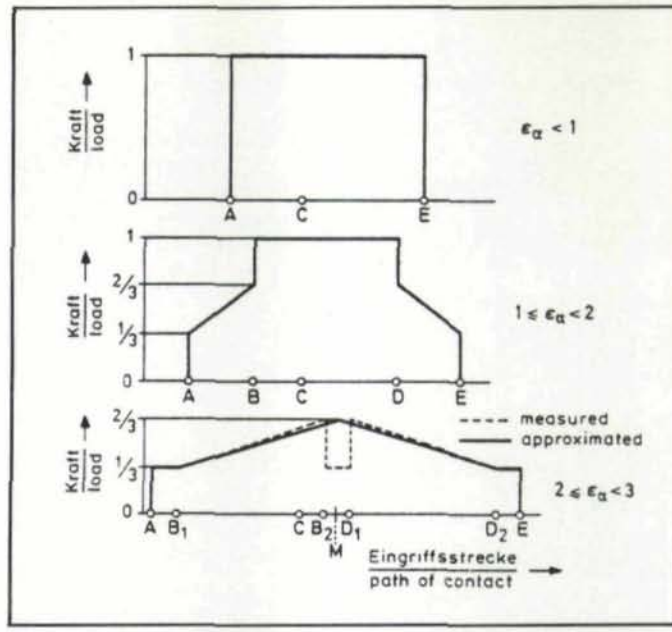

Fig. 5—Approximated Load Distribution as a Function of Contact Ratio

The gear bulk temperature, ϑ_M, is the temperature of the tooth surface before the mesh. It can be measured or calculated according to thermal network theory(6) or finite element methods.

An approximation is given by

$$\vartheta_M = (\vartheta_{oil} + C_1\vartheta_{fla\,int}) \cdot X_S \qquad (24)$$

where $C_1 = 0.7$ has been determined as a mean value from test results. For gears with more than one engagement on their circumference, higher bulk temperatures than calculated may occur.

The lubrication factor X_S accounts for the better heat transfer in splash lubricated gears compared with jet lubricated. From experience it can be assumed

$X_S = 1.0$ for splash lubrication and
$X_S = 1.2$ for jet lubrication (25)

The choice of the lubrication system, of course, has to be made due to other considerations, e.g., pitch line velocity.

$X_W = 0.45$	for austenitic steel (stainless steel)
$X_W = 0.85$	for steel with content of austenite more than average
$X_W = 1.00$	for steel with normal content of austenite
$X_W = 1.15$	for steel with content of austenite less than average
$X_W = 1.50$	for bath and gas nitrided steel
$X_W = 1.50$	for copper plated steel
$X_W = 1.25$	for phosphated steel
$X_W = 1.00$	for all other cases (e.g. through hardened steel)

Table 2—Estimation of Material Factor X_W

Scoring Temperature Evaluation

The scoring temperature, $\vartheta_{S\,int}$, can be determined according to the same set of equations (2) through (25) introducing the actual parameters of a gear oil test run with the oil under consideration. For differences between the materials or heat treatments of the test and actual gears, a relative correction factor has to be introduced.

$$\vartheta_{S\,int} = \vartheta_{MT} + C_2 \cdot X_{W\,rel\,T} \cdot \vartheta_{fla\,int\,T} \tag{26}$$

$$\text{with } X_{W\,rel\,T} = X_W / X_{WT} \tag{27}$$

Empirical data on the influence of the material resp. heat treatment are summarized in the welding factor X_W acc. table 2.

From our experience, only scoring tests on test gears can be correlated with the scoring performance in practical gears. Comparative tests with different gear oils, as well as milk and beer, have been made by Vogelpohl(7) and Wirtz(8). Different test principles are shown in Fig. 6. From the results as shown in Fig. 7, it is evident that frequently used test methods as Four Ball Test and Timken Test, do not correlate with the scoring properties in gears. Therefore, only data from oil tests on gears can be introduced into the evaluation of the scoring temperature.

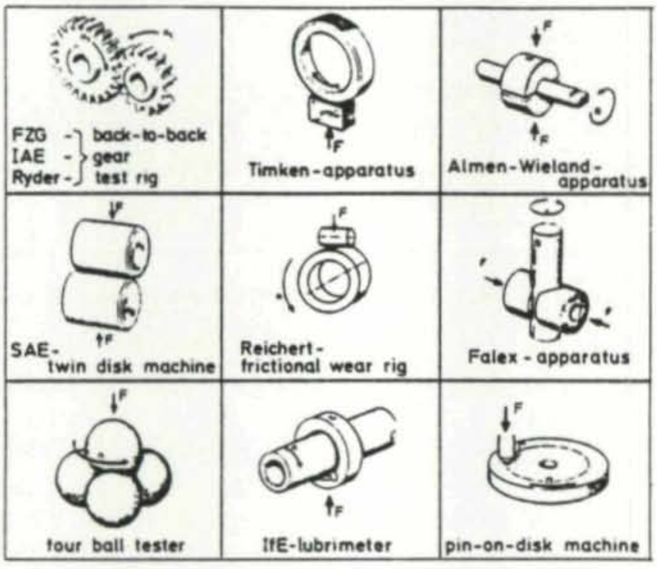

Fig. 6—Gear Oil Test Machines, Principles

An often used method is the FZG-Test A/8.3/90 as standardized in DIN 51 354 (see also AGMA 250.04). From the pinion scoring torque, T_{1T}, or the damage load stage, ϑ_{MT} and $\vartheta_{fla\,int\,T}$, can be taken from Fig. 8 for introduction into Eq. (26).

For computer calculations, the curves can be approximated by

$$\vartheta_{MT} = 80 + 0.23 \cdot T_{1T} \tag{28}$$

$$\vartheta_{fla\,int\,T} = 0.2 \cdot T_{1T} \cdot \left(\frac{100}{\nu_{40}}\right)^{0.02} \tag{29}$$

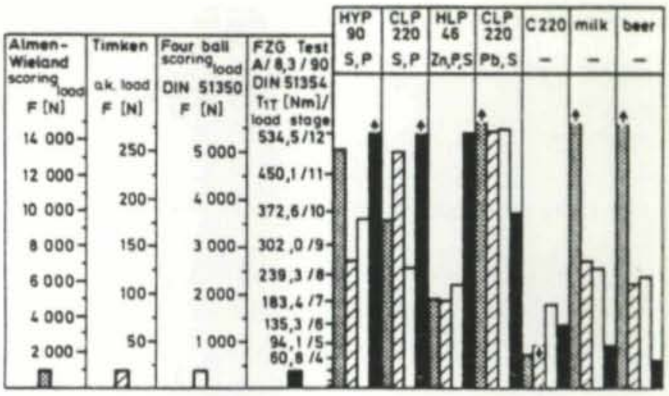

Fig. 7—Evaluation of Scoring Load in Different Test Rigs

The welding factor $X_{WT} = 1.0$ for the FZG-Test.

Starting from a Ryder Gear Test acc. FTM STD Nr. 791, and introducing the constant parameters of gear geometry and of operating conditions to the Eqs. (2) through (25), Fig. 9 is obtained.

The curves can be approximated by

$$\vartheta_{MT} = 90 + 0.0125\,(F_{bt}/b)_T \tag{30}$$

$$\vartheta_{fla\,int\,T} = 0.015 (F_{bt}/b)_T \cdot \left(\frac{100}{\nu_{40}}\right)^{0.03} \tag{31}$$

with the Ryder scoring load $(F_{bt}/b)_T$ to be introduced in Eqs. (30, 31) in ppi and the welding factor $X_{WT} = 1.0$.

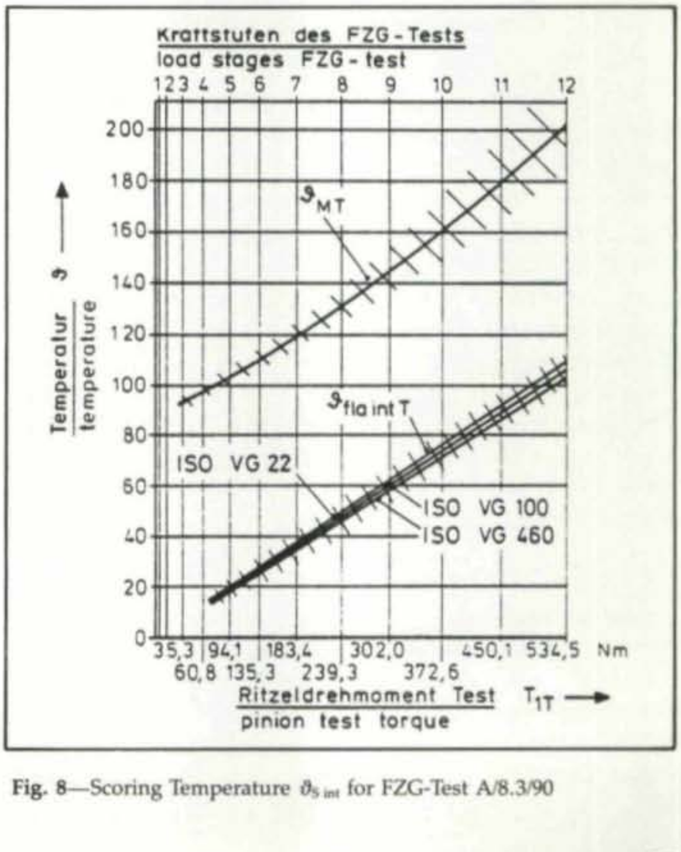

Fig. 8—Scoring Temperature $\vartheta_{S\,int}$ for FZG-Test A/8.3/90

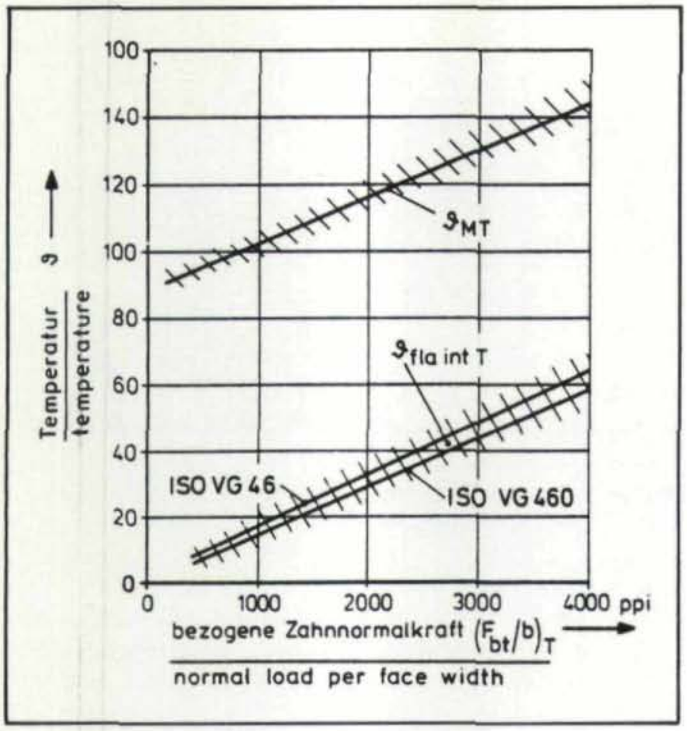

Fig. 9—Scoring Temperature $\vartheta_{S\ int}$ for Ryder-Gear-Test

Thus, test results of different test methods can be used as basic "strength" values. One of the major differences between FZG-Test and Ryder Gear Test is the pitch line velocity.

For high speed application, Ryder results obtained at v = 46 m/s and for low to medium speed application, FZG results at 8.3 m/s are somewhat closer to practical gear conditions and would be preferred, if available.

Comparison with Other Methods

General

An often used method for the evaluation of the risk of scoring damage is the Total Contact Temperature Criterion acc. Blok(1). The method predicts scoring when a maximum, local, instantaneous contact temperature, $\vartheta_{B\ max}$, exceeds a critical value, ϑ_{crit}. The contact temperature distribution along the flank is given by the sum of the constant bulk temperature and the local flash temperature (Fig. 10). The critical value is only dependent on the oil-material combination and independent of geometry and operating conditions. It can be expressed as a function of oil viscosity (Fig. 11). The total contact temperature method is also standardized in ISO DP 6336, and should be applied in parallel whenever possible. After some time of practical experience with both methods, it should be decided which one can be dropped.

The Scoring Index Method acc. Dudley(2) is derived from the Total Contact Temperature Criterion. It uses only the flash temperature part in a simplified way. Therefore, our objections against the Total Contact Temperature Criterion are also valid for the Scoring Index Method, at least to the same degree. Table 3 compares the field of application of the Total Contact

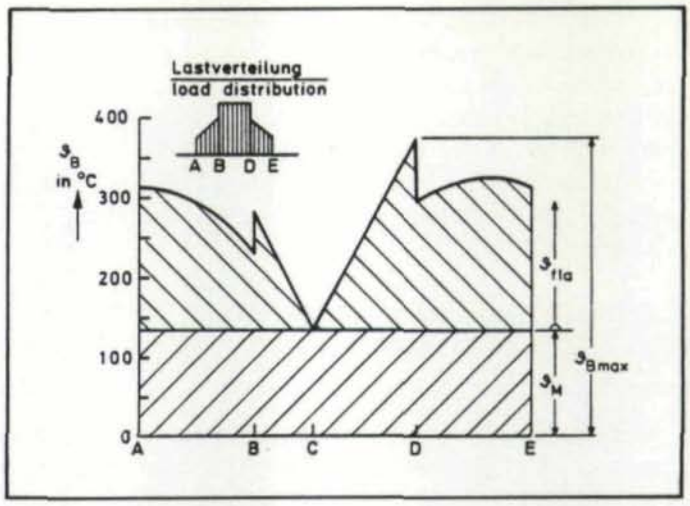

Fig. 10—Temperature Distribution along the Path of Contact acc. Blok

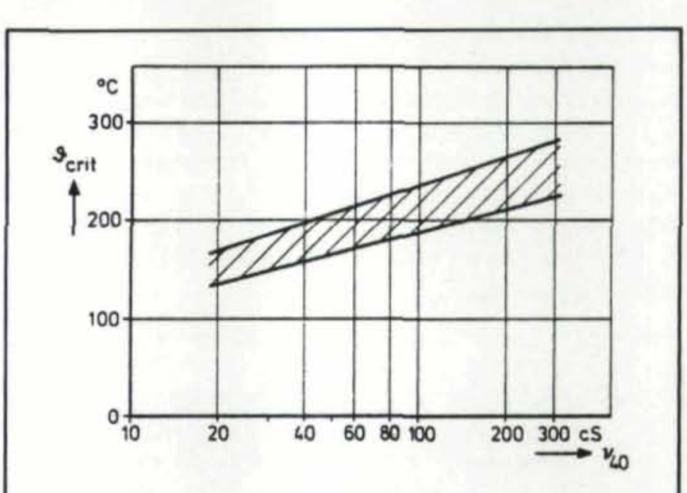

Fig. 11—Critical Contact Temperature for Flash Temperature Method

Temperature Criterion to that of the Integral Temperature Method.

In addition to the difficulties in the evaluation of local and instantaneous parameters of load—think of dynamic load distribution along the path of contact (Fig. 12)—coefficient of friction, radius of curvature under load, etc. Quite a few test results indicate that a single

	TOTAL CONTACT TEMPERATURE (SCORING INDEX)	INTEGRAL TEMPERATURE
Criterion	maximum, local, instantaneous contact temperature (simplified flash temperature)	mean, weighted flank temperature
Field of Application	straight mineral oils	straight, mild and EP mineral oils, synthetic oils
Critical Value	dependent on viscosity	from gear scoring test (e.g. FZG or Ryder test)

Table 3—Comparison of Total Contact Temperature and Integral Temperature Criterion

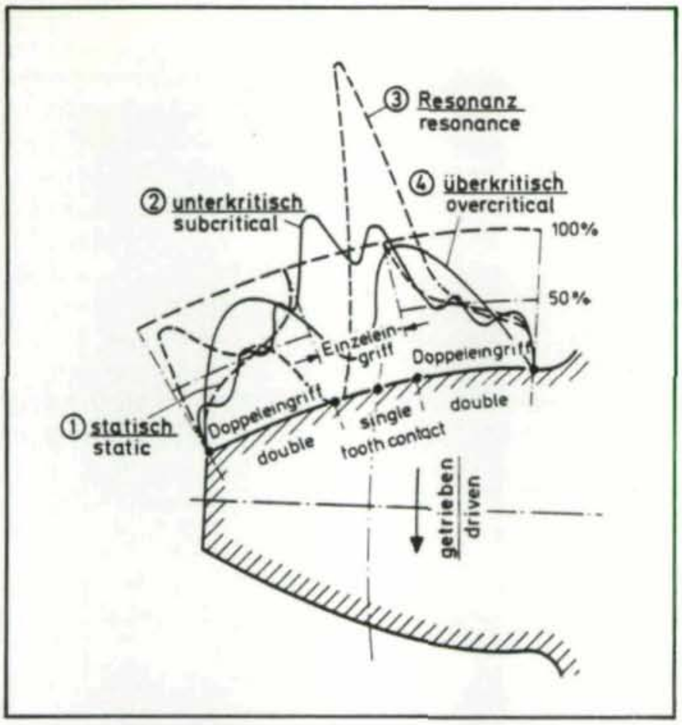

Fig. 12—Dynamic Load Distribution along the Tooth Flank acc. Rettig

temperature flash is not sufficient for a scoring catastrophy. Fig. 13 shows a tooth flank with incipient scoring of nearly the same severity, within an area of calculated contact temperatures between 320°C and 700°C. Deeper and more severe scoring and seizure would have been expected in the area of the tooth tip. This indicates the validity of a mean surface temperature as a critical energy level more than a temperature flash.

Another problem arises when tip relief is applied to gears with their critical temperature in the second point of single tooth contact (Fig. 14). In these cases, the calculated maximum contact temperature is not influenced by the tip relief while a strong increase in scoring load capacity can be observed in the test(9).

A series of tests of Ishikawa(9) were evaluated with

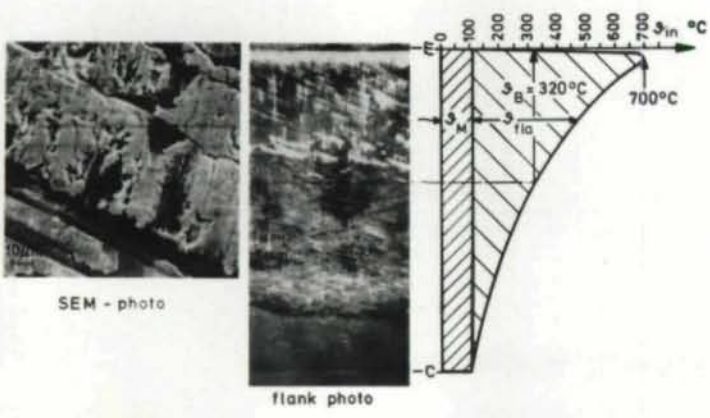

Fig. 13—Initial Scoring Damage

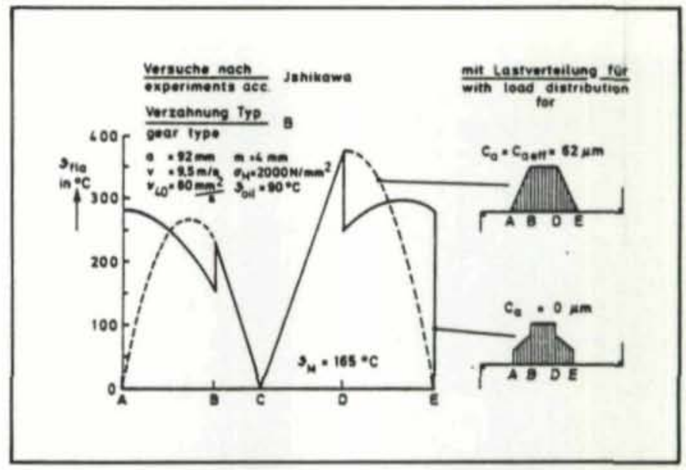

Fig. 14—Influence of Tip Relief on Flash Temperature

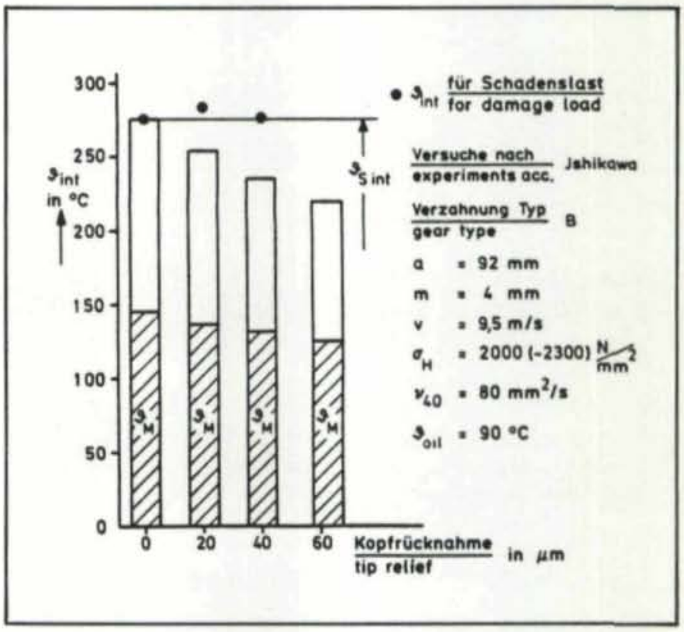

Fig. 15—Influence of Tip Relief on Integral Temperature

the Integral Temperature Method. They showed both steadily decreasing bulk and integral temperature, with increasing tip relief at constant load and a constant scoring temperature introducing the measured scoring loads (Fig. 15).

Examples

The validity of the Integral Temperature Method has been checked, with test results on different back-to-back test rigs, with center distances a = 91.5, 140 and 200 mm, with different gear geometries, different oils—straight mineral oils, compounded and EP-oils, synthetic oils of different viscosities—and different pitch line velocities up to v = 50 m/s. Fig. 16 shows the results of the calculations. For best correlation, the calculated safety factor for scoring conditions should be unity. The scattering is between about 1.0 and 1.4, which indicates a good correlation between test results and calculations, having in mind that the overload factors for the calculations have been set unity. For realistic overload factors, the calculated safety factor would somewhat decrease.

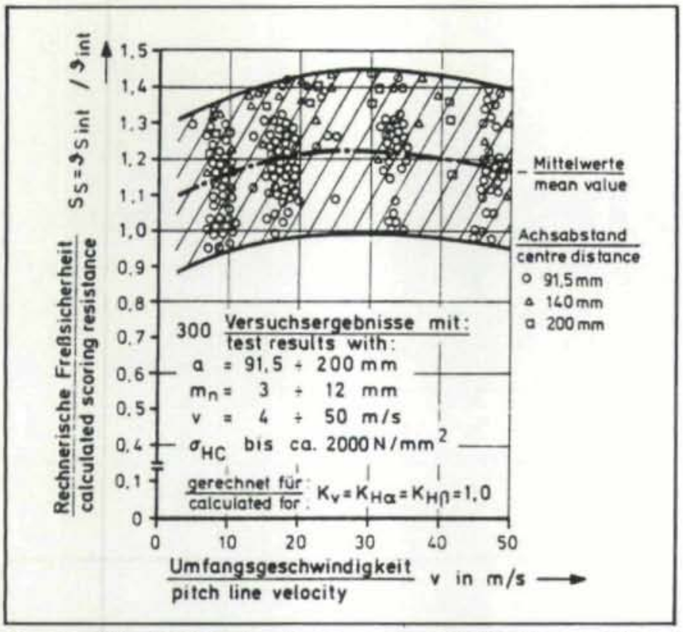

Fig. 16—Calculated Scoring Resistance for Test Results acc. Integral Temperature Method

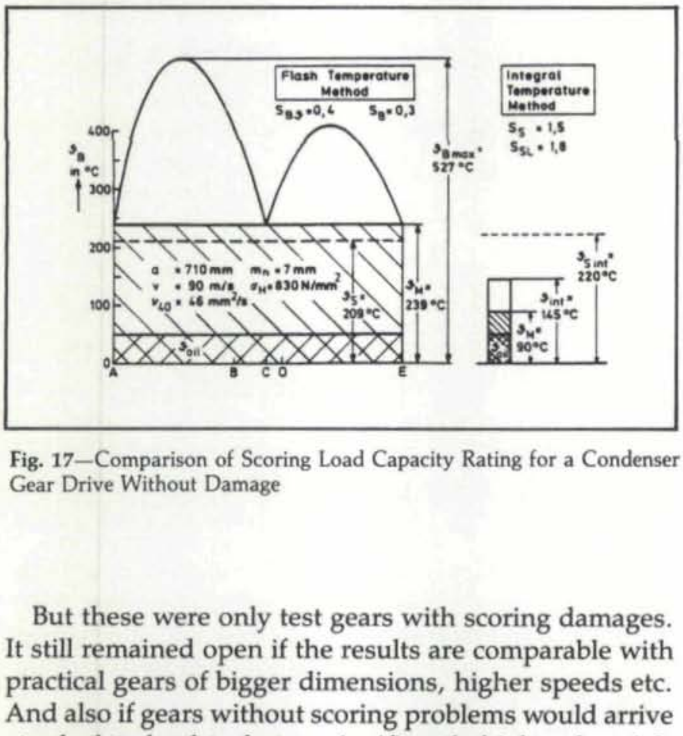

Fig. 17—Comparison of Scoring Load Capacity Rating for a Condenser Gear Drive Without Damage

But these were only test gears with scoring damages. It still remained open if the results are comparable with practical gears of bigger dimensions, higher speeds etc. And also if gears without scoring problems would arrive at calculated safety factors significantly higher than 1.0. Imagine that it is fairly easy to arrive at a value of 1.0, only extract often enough the square root of any figure and you will arrive at unity.

Therefore, we collected data from all kinds of practical gears with and without scoring damages. An example is shown in Fig. 17 for a condenser gear drive without scoring damages in service. The Total Contact Temperature Method calculates a safety factor of 0.4, the Integral Temperature Method of 1.5. A change of the unrealistic bulk temperature value, ϑ_M = 239 °C of the Total Contact Temperature Method to ϑ_M = 90 °C of the Integral Temperature, doesn't make it any better. The safety factor, S_B = 0.5, remains still far below 1.0, indicating a high scoring risk.

Similar experiences resulted when calculated safety factors of a variety of typical gears, out of more than one hundred examples were compared, with their scoring behavior in service. For the possibility of a comparison of Total Temperature resp. Scoring Index Criteria, we chose mainly gears which were lubricated with non EP-oils. The range of the operating conditions is shown in Fig. 18, and the results in Fig. 19. In cases where only the result of the Integral Temperature Method is shown, the other two criteria where not applicable because of the EP-character of the lubricant used. It is evident that the best correlation between calculated safety factors and practical experience is achieved with the Integral Temperature Method in a wide range of application.

From these recalculations, the different fields of scoring risk—high, borderline, low—as defined in *Intergal Temeprature Rating*, were established.

Achsabstand Centre distance	a	=	92 ÷ 2800	mm
Modul Module	m_n	=	3 ÷ 50	mm
Umfangsgeschwindigkeit Pitch line velocity	v	=	3 ÷ 120	m/s
Spezifisches Gleiten Specific sliding	v_{gmax}/v	=	0.1 ÷ 0.5	-
Übersetzung Gear ratio	u	=	1 ÷ 7	--
Hertzsche Pressung am Wälzkreis Hertzian stress at pitch circle	σ_{HC}	=	300 ÷ 2200	N/mm²
Unlegierte, mild und hochlegierte Mineralöle mit Viskosität straight mineral oils, mild and heavy E.P. - oils with viscosity	η_{40}	=	20 ÷ 400	mPa.s.

Fig. 18—Range of Significant Gear Paramaters for Actual Gears in Fig. 19

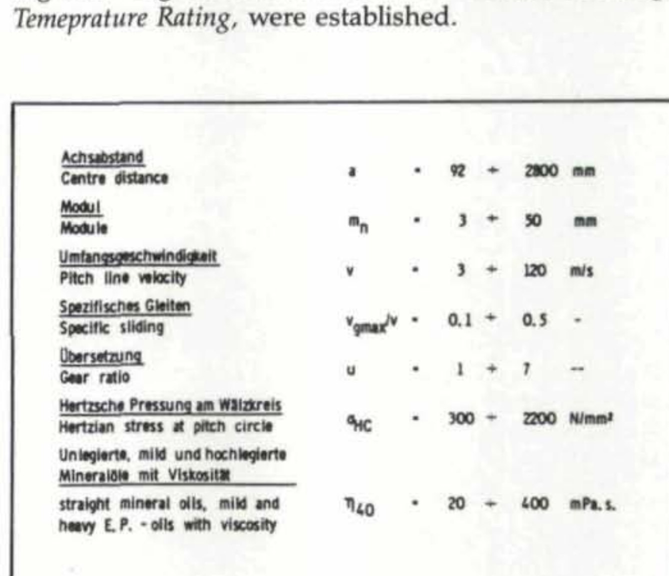

Fig. 19—Scoring Resistance of Actual Gears

(Continued on page 45)

Double Enveloping Worm Gears . . .
(Continued from Page 16)

(2) reductions at approximately 92%-93% overall efficiency or three (3) reductions at about 89%-90% efficiency. The worm gearbox with a 20:1 ratio will have about 85%-87% efficiency. A 30:1 ratio helical reducer will generally require three (3) meshes with approximately 89%-90% efficiency. The 30:1 wormgear speed reducer will have an efficiency of approximately 83%-84%. You can see the helical box *is* more efficient, but certainly not to the degree often claimed.

There are other inherent advantages in worm gearing which must be considered in evaluating the application and the type of gearing intended for that application. Double enveloping worm gearing will take a momentary overload of 300%, whereas helical gearboxes are only designed for 200%, momentary overload. Helical gearboxes restrict motor starting capacity to 200%, whereas double enveloping worm gearboxes permit 300%. Generally speaking, worm gearboxes are smaller in overall size and weight, and in terms of horsepower capacity, generally less expensive. In addition, with compactness of the double enveloping wormgear principle, double enveloping gearboxes are more compact and weigh less, horsepower for horsepower, than cylindrical gear reducers.

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Design of the Involute . . .
(Continued from page 44)

generally supposed. In other words, bearing pressures are not greatly affected by an increase in the pressure within the usual limits. This condition is graphically presented in Fig. 14. To construct this diagram, draw a line *A B* at right angles to the line of centers and tangent to both pitch circles. Then draw a line *C D* tangent to the base circles and passing through the pitch point *E*; this line representing the pressure angle. Now drop a perpendicular at any point *G* on line *A B*, passing through line *C D* at point *F*. With *E* as a center and *E F* as a radius scribe an arc. Increases in the load on the supporting bearings due to changes in pressure angle can be determined graphically by noting the changes in distance *H*, as the pressure angle changes. It is apparent that the load-increase is the ratio of lengths *E G* to *E F*, and is, therefore, proportional to the secant of the pressure angle.

The second column in Table II gives the secants of various pressure angles listed in the first column, and ranging from 14½ up to and including 30 degrees.

The last column lists in terms of percentage, the increase in the load as compared with 14½ degrees. It will be noticed that an increase in the pressure angle from 14½ to 20 degrees, results in an increased load on the supporting bearings of only 3 percent.

(Continued on the next page)

Scoring Load Capacity . . .
(Continued from page 30)

Conclusion

A new method for scoring load capacity rating, based on the calculation of a mean, weighted flank temperature, the integral temperature, has been described. The limiting temperatures necessary, for the definition of a scoring safety factor, can be obtained from any available gear oil test. The method is valid for all types of oils as straight mineral, mild and EP-oils, as well as, synthetic oils where gear scoring tests are available. The method was checked with more than 300 scoring tests on test rigs and more than 100 practical gears with and without scoring damages. A good correlation was found for the Integral Temperature Criterion, and it was obviously superior to the Total Temperature Method, as well as, to the Scoring Index Method.

The method has been modified for bevel and hypoid gears(10) and even in this field of application a good correlation between calculated scoring factors and field experience was achieved.

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