Calculated Scuffing Risk: Correlating AGMA 925-A03, AGMA 6011-J14 and Original MAAG Gear Predictions

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Introduction

This paper relates specifically to gears that are finish ground and considered high speed per ANSI/AGMA 6011; meshing elements with PLVs (pitch line velocities) in excess of 35 m/s or rotational speeds greater than 4,500 rpm (Ref. 1).

Current application standards by AGMA & ISO both provide methods for rating gears for macropitting and bending fatigue. However, neither have a standard to calculate scuffing risk. Both have provided documents; AGMA 925-A03 and ISO/TS 6336-20 (formally ISO/TR 13989-1) as informative suggestions to consider scuffing risk in gear design. Nevertheless, scuffing requires the same consideration as macropitting and bending in rating gears — especially in high-speed applications with high sliding velocities.

Scuffing is not a fatigue phenomenon and may occur instantly, often during early stages of operation. This failure is directly related to the lubricant oil film, which does not adequately separate the surfaces. Figure 1 is a classic example of gear tooth scuffing. One or more of the following are typical root causes for scuffing:

- The threshold for scuffing resistance may not have been considered when designing the geometry of the gearset. For example, excessive sliding velocities occurring between the meshing elements.
- The lubricant selected was not in accordance with the original lubricant intended for the application.
- The load distribution along the tooth flanks did not consider mechanical (elastic) deflection and thermal deformation resulting in either non-uniform load or less than full face width loading along the entire flank at full load.
- The surface roughness of the gearset was excessive.
- Operational oil inlet temperature was excessive.

Figure 1 Scuffing example.

Background

In 1937 Harmon Blok published his theory about the relationship between contact temperature and scuffing, which was presented at the Second World Petroleum Congress held in Paris, June, 1937 (Ref. 3).

This subject went largely unnoticed in the U.S. until the early 1950's (Ref. 2).

Until that time most high-speed gears produced in the U. S. were through hardened, which did not require a calculation for scuffing. While scuffing distress in case hardened gears had been recognized for a fairly long time, in earlier times it was often referred to as scoring. This was incorrect as scoring is an abrasive action with surfaces scratching each other during engagement, whereas scuffing is an adhesive action where transfer of metal from one surface to the other occurs due to welding and subsequent tearing. In 1963 MAAG identified it as "hot scoring," thus separating the nature of these actions of distress. With the publication of AGMA 925-A03, the difference was made clear that an adhesive action is referred to as scuffing.

Two schools of thought emerged; 1) the flash temperature criterion according to Blok based on the highest momentary local maximum temperature, and 2) the integral temperature criteria, which is based on the integrated temperature along the path of contact arriving at a steady average gear tooth temperature. Blok's flash temperature criteria has been the adopted practice in AGMA standards, much in parallel with MAAG's work first recorded in the early 1960's.

While the scuffing mechanism is now clearly understood, what remains uncertain is how to assess risk on a given gear unit. Aside from operating environmental changes such as deterioration of good flank load distribution, quality of lubricant in service, operational changes such as load and speed, assessment of scuffing risk for a new installation has been the widely favored evaluation method using the maximum flash temperature as the critical factor. This document presents examples of assessing scuffing risk for high-speed gears applying three methods of calculating the scuffing criteria. The comparable results demonstrate all three methods are effectively credible for calculating scuffing risk.

Blok's theory is based on what is termed "flash temperature." Professor Blok's definition states scuffing will occur when the local temperature at the point of contact rapidly rises (flashes) due to friction at the tooth contact point. This momentarily raises the lubricant temperature above the lubricant's ability to maintain its viscosity.

Blok's criteria that limit sliding velocity are directly related to the local temperature occurring in the contact area of the gear teeth. This temperature — termed as the "flash temperature" — can be calculated for any local point on the involute along the path of contact. The maximum flash temperature calculated becomes the design point to which a gearset is rated for its maximum power transmission capability against scuffing. Under operational conditions, scuffing is influenced by a combination of the film thickness and the coefficient of friction, dependent on surface pressure and sliding velocity. High-speed gears are subject to higher sliding velocities. The highest velocity occurs at the SAP (beginning — Start of Active Profile) and EAP (end — End of Active Profile) of the tooth contact path.

According to Professor Blok the governing temperature peak is obtained by adding the flash temperature to the mean temperature of the tooth flanks before entering the contact zone. This value is called the total contact temperature. Therefore, the total contact temperature is the sum of two components: the flash temperature and the tooth body temperature (Ref. 13).

$$\theta_{\text{total contact max}} = \theta_{\text{flash max}} + \theta_{\text{tooth temp}} \tag{1}$$

Where

- $\theta_{\text{total contact max}}$ is total contact temperature;
- $\theta_{\text{flash max}}$ is flash temperature is the instantaneous temperature rise over and above the steady tooth flank temperature;
- $\theta_{\text{tooth temp}}$ is tooth temperature (non-loaded flank temperature) can be defined as the steady surface temperature of the contacting body.

This so-called flash temperature should not be confused with the actual flash temperature of a lubricant. The flash point is the lowest temperature to which a lubricant must be heated before its vapor, when mixed with air, will ignite but not continue to burn. The flash point of most ISO VG 32 R&O mineral oils is in the 370°–390°F (188°–199°C) range (Ref. 10).

Stribeck curve. The Stribeck curve, originally developed for journal bearings (Fig. 2), expresses how the coefficient of friction varies over the entire range of lubrication.

For high-speed gears, the sliding velocity becomes a greater factor when compared to speed overall, including the hydrodynamic, mixed, and boundary lubrication regimes. These regimes are noted by the vertical dotted lines and are described as regimes of lubrication 1, 2, 3 that directly affect the fatigue life of the gear teeth relative to surface distress. The effects of this phenomenon were introduced by Richard Stribeck while conducting research in the field of Tribology (Ref. 11).

The three regimes of lubrication are as follows:

Regime I: Only boundary lubrication exists with essentially no EHL film, and contact of the asperities of gear flanks in motion relative to one another is pronounced.

Regime II: Partial EHL oil film is developed and there is occasional contact of the asperities of gear flanks in motion relative to one another.

Regime III: Full EHL oil film is developed and separates the asperities of gear flanks in motion relative to one another (Ref. 2).

NOTE: For gears, beyond the boundary between Regime II and Regime III, the friction coefficient effectively decreases.

Development of Application Methodology

The original formula derived by Professor Blok for the "flash temperature" is derived from the assumption that the heat source, having a uniform distribution inside a defined contact area (semi-width of the Hertzian contact band), moves along the tooth flank with a constant speed. In that way, the tooth flank receives the entire source heat flow. The original expression by Professor Blok (Ref. 3) was published by Dr. Wydler as follows: (Ref. 13).

$$\theta_{fl(i)} = 0.83 \frac{f_{(i)} P_1 \times [V_1 - V_2]}{[B_{m1} V_1^{0.5} + B_{m2} V_2^{0.5}] \times R^{0.5}} \tag{2}$$

where

- $\theta_{fl(i)}$ is flash temperature, °C;
- $f_{(i)}$ is instantaneous coefficient of friction;
- P_1 is specific loading, kg/cm;
- V_1, V_2 Flank speeds perpendicular to the line of action, cm/sec;
- B_{m1}, B_{m2} Heat dissipation characteristics — thermal coefficient of steel;
- $R^{0.5}$ Semi-width of Hertzian contact band, cm.

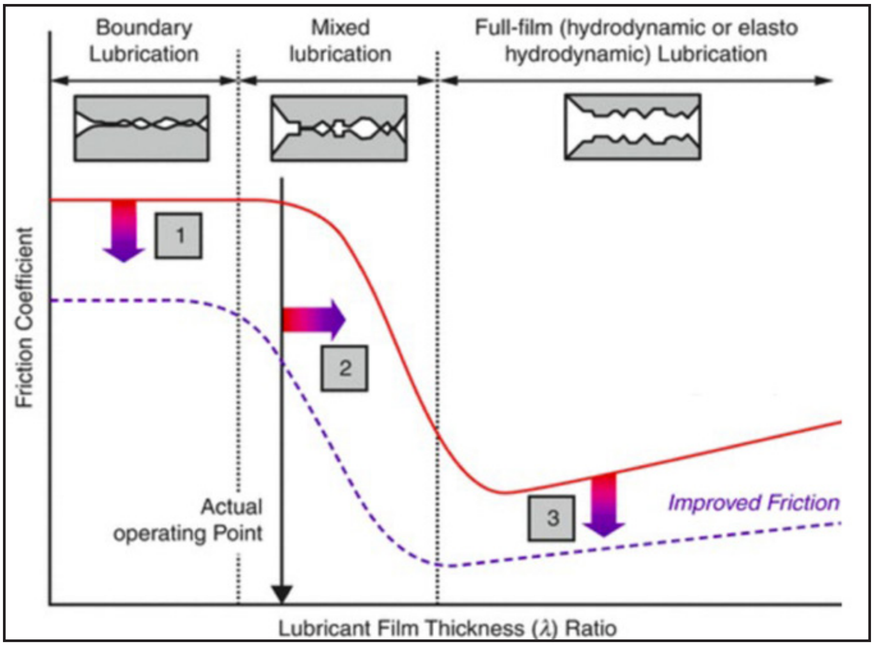

Figure 2 **Stribeck curve. (Source: Wang Q.J., Chung YW, "Encyclopedia of Tribology." (2013) Springer, Boston, MA.)**

Kelley later modified Blok's flash temperature equation by considering the influence of surface roughness (Ref. 11), and Dudley modified Kelley's equation by considering the loading actually applied on the gear tooth flank (Ref. 4). This modification was later adopted by AGMA-925-A03 in the current form:

$$\theta_{fl(i)} = 31.62\,K\mu_{m(i)} \frac{X_{\Gamma(i)}\,w_n}{\left(b_{H(i)}\right)^{0.5}} \times \frac{\left|v_{r1(i)} - v_{r2(i)}\right|}{B_{m1}\left(v_{r1(i)}\right)^{0.5} + B_{m2}\left(v_{r2(i)}\right)^{0.5}} \tag{3}$$

where

$\theta_{fl(i)}$ is flash temperature, °C;
K is 0.80, numerical factor valid for a semi-elliptic (Hertzian) distribution of frictional heat over the instantaneous width, 2 b_H, of the rectangular contact band;
μ_{mi} is coefficient of friction;
$X_{\Gamma(i)}$ is load sharing factor;
w_n is normal unit load, N/mm;
$b_{H(i)}$ is semi-width of Hertzian contact band, mm;
$v_{r1(i)}$ is rolling tangential velocity of the pinion, m/s;
$v_{r2(i)}$ is rolling tangential velocity of the wheel, m/s;
B_{m1} is thermal contact coefficient, pinion, N/[mm s$^{0.5}$K];
B_{m2} is thermal contact coefficient, wheel, N/[mm s$^{0.5}$K].

NOTE: subscript "*i*" = denotes a point on the line of action (Ref. 11).

Numerous methods of determining the propensity for scuffing have evolved and led us into today's methodology, as found in AGMA 925 and ISO/TS 6336-20. The evolution of some of these methods can be summarized in the following sections.

MAAG method 1963. In 1963 MAAG developed a simplified procedure to assess scuffing risk according to the flash temperature criteria based on the following relationship:

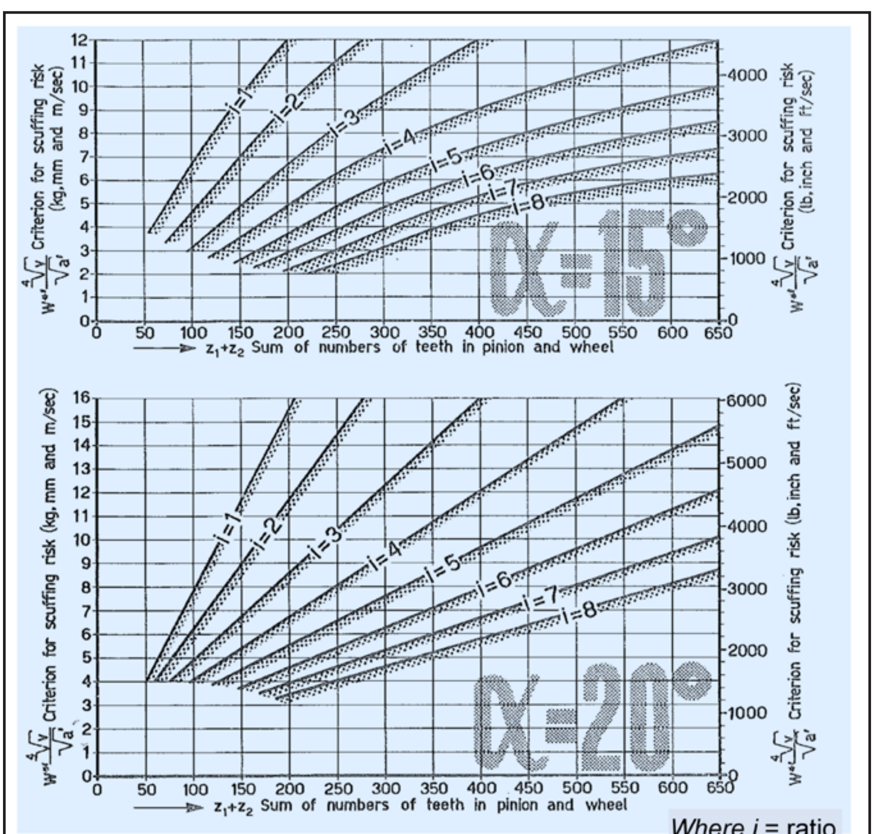

Figure 3 MAAG 19634 Handbook assessing scuffing.

$$F_{load63} = w^{*\prime}\left(\frac{\sqrt[4]{v'}}{\sqrt{a'}}\right) \tag{4}$$

Where

F_{load63} is permissible operating load, kg/mm;
$w^{*\prime}$ is unit loading, kg/mm;
v' is pitch line velocity, m/s;
a' is center distance, mm.

Formulation for application using SI units $w^{*\prime} = w'/9.81$ to be used with Figure 3. Figure 3 was published by MAAG 1963 handbook for general use in the industry, primarily in assessing scuffing risk for high-speed gears (Ref. 8).

Wydler method — 1972. On the strength of experience gained in high-capacity gear design, the following relationship between the scuffing limit, load and speed had been determined (Ref. 13).

$$\theta_{\beta fl(lim)} \approx Power \cdot \sqrt[4]{v} \tag{5}$$

Where

$\theta_{\beta fl(lim)}$ is flash temperature at scuffing limit, °C;
v is pitch line velocity, m/s.

It is difficult to predict the coefficient of friction for high-speed applications, but it is a proportionate function to speed and load. In high-speed gears Dr. R. Wydler investigated tests conducted by Hughes and Waight (Ref. 5). From this it was determined that at higher speeds, due to the improved load carrying capacity with full hydrodynamic lubrication, the influence of speed on scuffing was lessened. It was also determined that the proportional relationship with speed and load influenced the coefficient of friction in a proportional manner according to:

$$\sqrt[4]{\frac{load}{speed}} \tag{6}$$

At lower loads and speeds the frictional relationship varies (Ref. 13).

For this calculation method it is assumed that the highest flash temperature occurs at the tooth tip for an unmodified profile.

Based on the limited experience at the time and using Ra = 0.48, Wydler set up conservative permissible scuffing limits $\theta_{\beta(lim)} = 140$°C for a typical ISO VG 32 oil, and $\theta_{\beta(lim)} = 156$°C for ISO VG 46 (Ref. 13).

In summary, this method contains a considerable, but not precisely known, safety margin, for it embraces the practical conditions of high-speed, high-capacity gear drives.

Integral temperature method 1983. AGMA 925-A03 refers to Winter and Michaelis, who published a paper on scuffing where they introduced a method to assess scuffing by the integral temperature method. In AGMA 925-A03 the integral temperature method was mentioned as an alternate method to the flash temperature method. This method involves the calculation of a scuffing load basically independent of speed, but controlled by gear geometry. Application of the method requires comparison of the proposed gearset based on a test rig result to a known test rig gearset and tested oil. A comparison of the flash temperature method and integral temperature method has shown the following: Blok's method and the integral temperature method give essentially the same assessment of scuffing risk for most gearsets; Blok's method and the integral temperature method give different assessments of scuffing risk for those cases where there are local temperature peaks. These cases usually occur in gearsets that have low contact ratio, contact near the base circle, or other sensitive geometries. Blok's method is sensitive to local temperature peaks because it is concerned with the maximum instantaneous temperature, whereas the integral temperature method is insensitive to these peaks since it averages the temperature distribution (Ref. 2).

An application of the integral temperature method can now be found in ISO/TS 6336-21, based on a rig testing gears running at pitch line velocities less than 80 m/s. The integral temperature method was first presented in ISO/TR 13989-2. It averages the flash temperature and supplements empirical influence factors to the hidden load sharing factor. The resulting value approximates the maximum contact temperature, thus yielding about the same assessment of scuffing risk as the flash temperature method of this part of ISO/TR 13989, now ISO/TS 6336-20.

The equations might be used for gears which run at higher speeds, but with increasing uncertainty as speed increases. The integral temperature method is less sensitive for those cases where there are local temperature peaks, usually in gearsets that have low contact ratio or contact near the base circle or other sensitive geometries (Ref. 6). Furthermore, ISO/TS-6336-20 states the methodology is based on tested gears operated at pitch line velocities less than 80 m/s, with increased uncertainty at speeds in excess of experimental limits.

This paper reaffirms the flash temperature method is preferred for assessing scuffing risk for high-speed gears.

Current Comparative Methods of Determining the Propensity of Scuffing

ANSI/AGMA-6011 Annex B (***MAAG method 1963/1983 mod***) ***— simplified scuffing criterion method.*** With twenty years of experience, MAAG broadened the approach to verifying scuffing resistance. Instead of approximating the probability of risk using a set of curves, MAAG developed a simplified calculation procedure that compares an applied load function F_{load} against a calculated value F_{geom}, according to the calculation procedure in the MAAG gear book 1963 establishing a rule:

$$F_{load} < F_{geom} \tag{7}$$

Where F_{load} assumed a straight mineral oil of viscosity ISO VG 32, the definition was expanded to apply to oils with other viscosities in centistokes. Therefore:

$$F_{load} = w'(\sqrt[4]{v'})\left[\sqrt[6]{\frac{46}{\nu}}\right] \tag{8}$$

Where

w' is specific tooth load on the operating pitch circle, N/mm;

v' is pitch line velocity, m/s;

ν is viscosity, cSt @ 40°C;

F_{geom} is based on the gear size, numbers of teeth, and rotor size based on center distance; the coefficient of friction assumes an Ra = 0.4.

$$F_{geom} = \frac{(50 + z_1 + z_2)(a)^{0.5}}{A}[Cu] \tag{9}$$

Where

z_1 is number of teeth of the pinion;

z_2 is number of teeth of the gear;

a is center distance, mm;

A is taken from Table 1;

C_u is taken from Table 1.

Table 1 Values A and Cu for calculating F (geometric)

α	A	C_u at $1 \le u < 3$	C_u at $3 \le u \le 10$
15	350	$95 + 28.6\,(3-u)$	$130-10\,[112.5-(13-u)^2]^{0.5}$
17.5	300	$90 + 30\,(3-u)$	$120-10\,[90-(12-u)^2]^{0.5}$
20	300	$100 + 33.3\,(3-u)$	$130-10\,[109-(13-u)^2]^{0.5}$
22.5	250	$95 + 28.5\,(3-u)$	$130-10\,[112.5-(13-u)^2]^{0.5}$
25	250	$105 + 31.4\,(3-u)$	$140-10\,[133.5-(14-u)^2]^{0.5}$

Notes:
α is normal pressure angle, degrees;
u is gear ratio (z_2/z_1)

This method was later refined with additional experience of 15 years, allowances for toothing accuracies, experience with flank modifications, increased knowledge of thermal influences on the tooth flanks, and improved quality of lubricants. Calculations as described in MAAG Handbook 1983 added a rating factor in the formula for F_{load} to permit higher F_{geom} values. These were first published in ANSI/AGMA 6011-H98 (Ref. 1).

$$F_{load} = \frac{w'}{C_w}(\sqrt[4]{v'})\left[\sqrt[6]{\frac{46}{\nu}}\right] \tag{10}$$

Where, C_w is risk factor:

1.10 - Conservative value;

1.15 - Nominal value;

1.20 - Maximum value.

AGMA 925-A03/MAAG method 1983 — differentiated calculation procedure. There is uncertainty concerning the estimation of the "bulk" tooth temperature, coefficient of friction, and allowable temperatures as speeds exceeded the range with experimental background. High-speed gears are defined in AGMA as gear operating above 35 m/s or pinion speed above 4,500 rpm. As an example, 80 m/s is significantly greater than the AGMA threshold, but it is well below high-speed applications that are commonly up to 175 m/s or greater. A reliable method applied for all the high-speed applications is an important consideration for calculating scuffing risk.

NOTE: 80 m/s is referenced in the following section as a starting threshold for an increased heat source in the gear mesh.

The derivation of absolute allowable flash temperature limits has been established with FZG gear test rigs. However, their determination has been accumulated with the use of relatively small gears. These gears will develop heat characteristics unlike real-world high-speed gears that are larger in size. To apply a base for scuffing assessment that reflects real gears rather than test gears, the total contact temperature requires a value for the base tooth body temperature. With this method it was realized that the calculation of the steady tooth bulk temperature is complex and is therefore beyond a routine calculation procedure. Thus, MAAG applied the integral temperature criteria establishing a base temperature among all real high-speed gears of 100°C, this can be allowed for by substituting the value of the dynamic viscosity η_M at this temperature for determination of

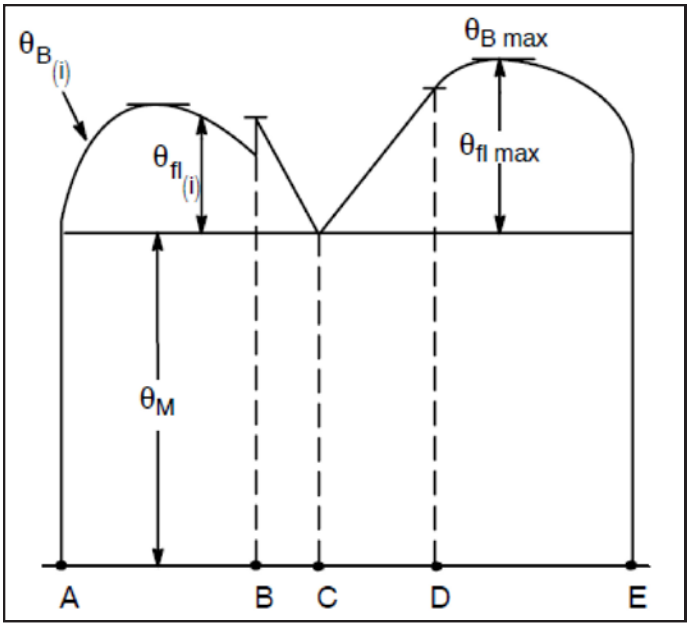

Figure 5 **Contact temperature along the line of action.**

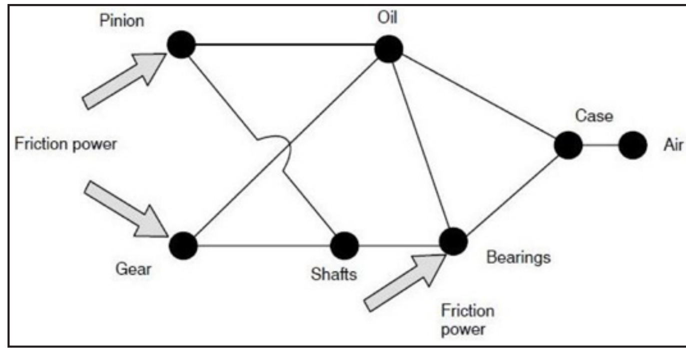

Figure 4 **Identifying the thermal network in a gearbox.**

the coefficient of friction (Ref.9).

Sometimes the tooth temperature is referred to as the tooth bulk temperature or body temperature. This can be misleading. The surface or flank temperature of the non-drive flank better defines the tooth temperature.

Carlos Wink conducted an experimental study to predict scuffing risk applying AGMA 925-A03 in terms of calculation of the tooth temperature. The results indicated that AGMA 925-A03 is indeed effective and consistent in predicting scuffing risk. Tooth flank temperature was estimated (Ref.12):

$$\theta_{tooth\ flank} = k_{sump}\,\theta_{oil} + 0.56\,\theta_{flmax} \tag{11}$$

where

$\theta_{tooth\ flank}$ is θ_M from Figure 5;

k_{sump} is 1 for splash lube and 1.2 for spray lube;

θ_{oil} is oil supply or sump temperature in °C;

θ_{flmax} is the maximum flash temperature found over all line–of–contact points.

Recalling the contact temperature is the sum of the flash temperature and the tooth body temperature:

$$\theta_{Btotal\ contact\ max} = \theta_{flash\ max} + \theta_{tooth\ flank} \tag{12}$$

At no point in the contact zone must the flash temperature be allowed to exceed a maximum permissible value (Ref.8).

MAAG's differentiated calculation procedure does not provide criteria for risk assessment. Variables that are unique to each gear unit manufacturer have significant impact on risk assessment. These include size (small gears heat up more than large gears), flank surface roughness, thermal deformations of the gears during operation (changing the load distribution across the face width and between several teeth).

Therefore, MAAG suggests experience with statistical investigation is required to develop reliable limit values for practical use (Ref.9).

AGMA 925-A03. This document addresses several forms of gear tooth surface distress applying various methods and provides an elementary risk assessment omitted in the MAAG differentiated calculation procedure.

AGMA 925 permits the applicant or designer use of specific values that are variable in assessing scuffing risk. This provides a more precise answer to determining the risk assessment. AGMA 925 can be applied to gears of variable sizes, speeds, and configuration. ANSI/AGMA 6011 is specific for high-speed gears and many of the parameters are fixed for assessing risk in high-speed applications.

AGMA 925 addresses the importance of determining the tooth temperature with analysis of the heat flow balance in the gearbox. As stated, there are several sources of frictional heat, of which the most important ones are the tooth friction and the bearing friction; other heat sources, like seals and oil flow, may also contribute. For gear pitch line velocities above 80 m/s, churning loss, expulsion of oil between meshing teeth, and windage loss become important heat sources that should be considered. Heat is conducted and transferred to the environment by conduction, convection and radiation (Ref.2).

An example of identifying the thermal network in a gearbox is depicted in Figure 4.

Table 2 **Parametric Comparison**

	AGMA-6011 (MAAG Method 1963/1983 modified) Simplified Scuffing Criterion	AGMA 925-A03 modified MAAG 83 Differentiated Calculation Procedure	AGMA 925-A03
Oil Type	Mineral	Mineral	Variable †
Oil Viscosity	Variable	Variable	Variable †
Oil inlet temperature	Fixed	Fixed	Variable †
Surface Roughness	Fixed	Variable	Variable †
Combination of trapezoidal loading	Fixed (Smooth meshing)	Fixed (Smooth meshing)	Calculated †
Coefficient of friction*	Fixed	Calculated	Calculated
Tooth flank surface (bulk) temperature**	100 °C	100 °C	Calculated
Thermal Coefficient of Contact for Steel***	Fixed	Fixed	Fixed
FZG Oil Load Stage	Fail 6	Fail 6	Fail 6

*This value accounts for changing tooth geometries resulting from modules fitted with number of teeth and center distance.
**This is derived from the tooth temperature which is a sum of the entire environment.
***Thermal Coefficient of Contact for Steel Bm :13.796 N/[mm $s^{0.5}$ K]
† Refer to Table 2

Table 3 AGMA-925-A03 preset input parameters	
Oil Type:	Mineral
Oil Viscosity:	VG 32
Inlet Oil Temperature:	70 °C
Surface roughness, Ra:	0.48 μm
LSF (load sharing factor):	smooth meshing/with profile modification
Thermal Coefficient of Contact for Steel B	13.796 N/[mm s$^{0.5}$ K]

AGMA 925 defines the contact temperature as described earlier by Professor Blok and repeated in MAAG methodology.

The scuffing risk is calculated to very similar criteria as described in MAAG 1983 for C_w:

Probability of scuffing risk:

<10% Low (for this paper authors consider <5% as low threshold)

10 to 30% Moderate

>30% High (Ref. 2)

Setting the Parameters

A comparison of the risk assessment is as follows:

AGMA-925-A03: In order to create a comparatively close alignment of the subsequent example calculations it was necessary to predefine those variables with the following fixed parameters. Parameters fixed for the comparison of different operating units are listed in Table 3. The data input for each installation is listed in Table 4.

Example Reference Calculations

NOTE: For added evaluation in assessing risk, a margin of risk has been created for Table 5 through Table 7 and is defined as follows:

$$MR\,(\%) = \left[\frac{F_G - F_L}{F_G}\right] \times 100 \tag{12}$$

Where

MR is margin of risk %;

F_G is allowable geometry load;

F_L is rated transmitted load.

Discussion

This presentation showing nine examples and applying the three methods of calculating scuffing risk are in general agreement; this supports their mutual credibility. These examples, however, involved applications specific to high-speed gears. Conditions vary for other applications and the results may not produce consistent results.

AGMA 925 –A03 has been calculated applying two procedures. Results by the differentiated calculation procedure are summarized in Table 8 and the empirical calculation procedure in Table 9. The calculation of the steady bulk temperature is complicated and beyond a routine calculation procedure (Ref. 9). So the

Table 4 Data table — inputs

Ex. Ref.	est. op hrs	Helical	Type	CD	FW	PLV	Kw	Input (rpm)	Output (rpm)	Module	Z_1	Z_2
1	>200k	single	increaser	400	236	142.0	10,515	4,831	11,406	6.5000	36	85
2	120k	double	decreaser	360	228	112.0	7,915	8,476	4,573	5.5000	41	76
3	175k	single	increaser	250	120	118.3	4,096	6,840	13,310	4.5000	37	72
4	160k	single	decreaser	580	502	109.3	37,286	4,670	2,927	6.2500	47	75
5	180k	single	increaser	520	352	142.1	22,670	3,428	10,933	6.5000	37	118
6	200k	single	increaser	780	255	123.0	13,500	1,775	9,951	7.0000	33	185
7	150k	double	increaser	610	370	92.7	16,406	1,800	7,636	6.0000	33	140
8	150k	single	increaser	509	323	72.6	12,304	1,800	5,606	6.9000	35	109
9	120k	single	increaser	600	270	88.1	9,694	1,800	7,582	5.9000	37	163

Table 5 Output data

Ref.	Sliding Velocities (m/s) SAP V_{s1A}	LPSTC V_{s1B}	HPSTC V_{s1D}	EAP V_{s1E}	max. Hertzian contact stress N/mm^2	Max coefficient of friction* $\mu_{(max)}$	Min. specific film thickness λ_{2bH} (μm)
1	24.78	2.06	8.30	31.02	585.71	0.0613	2.07
2	16.72	5.47	7.64	18.89	580.33	0.0561	1.87
3	21.54	2.64	6.88	25.78	683.90	0.0632	1.99
4	16.77	3.51	5.25	18.50	711.02	0.0787	1.72
5	20.40	3.91	8.94	25.43	606.78	0.0666	2.19
6	18.10	0.14	7.82	25.78	630.43	0.0723	1.59
7	12.11	4.91	7.95	15.15	723.13	0.0835	1.39
8	11.33	1.52	4.68	14.49	757.99	0.0824	1.08
9	13.35	0.38	4.81	18.54	640.00	0.0721	1.28

*calculated at the highest contact temperature
**NOTE: references, 4, 7, 8, 9 have had experiences of surface distress.

Table 5a Notes to output data

Reference	Notes:
1	
2	
3	
4	history of scuffing was corrected, modified spray bar and lead correction
5	
6	
7	history of scuffing was corrected, modified spray bar and lead correction
8	latent micropitting history no solution
9	latent micropitting history currently under investigation

NOTE: For added evaluation in assessing risk a margin of risk has been created for Table 5 through Table 7 and is defined as follows:

Table 6 Scuffing risk according to Maag 63 & Wydler

Ref.	Wydler	Maag 1963			
	(NO account for profile modification)				
	Flash Temp Pinion °C	Flash Temp Wheel °C	Allowable °C	Margin °C	Risk
1	111.0	106.0	140.0	20.7%	SAFE
2	135.0	129.0	140.0	3.6%	SAFE
3	118.0	115.0	140.0	15.7%	SAFE
4	135.0	129.0	140.0	3.6%	SAFE
5	93.0	82.0	140.0	33.6%	SAFE
6	144.0	122.0	140.0	-2.9%	At Risk!
7	144.0	119.0	140.0	-2.9%	At Risk!
8	145.0	131.0	140.0	-3.6%	At Risk!
9	122.0	108.0	140.0	12.9%	SAFE

Table 7 Scuffing risk according to Maag 83

Ref.	Nageli	Maag 1983	Simplified	Scuffing	Criterion
	Cv = 1.0 with profile modification /Nominal Risk Cv = 1.00				
	Load N/mm^2	Geometry N/mm^2	margin %	risk	tooth temp °C
1	1172.0	1382.5	15.2%	SAFE	100
2	2254.6	2026.4	-11.3%	AT RISK!	100
3	1030.8	1132.1	8.9%	SAFE	100
4	2254.6	2026.4	-11.3%	AT RISK!	100
5	1125.6	1469.4	23.4%	SAFE	100
6	1528.2	1404.2	-8.8%	AT RISK!	100
7	1603.1	1343.2	-19.3%	AT RISK!	100
8	1658.0	1406.8	-17.9%	AT RISK!	100
9	1307.2	1443.7	9.5%	SAFE	100

For historical reference tabulated results of the Maag 1963 method — Table 6 (was the basis used by Wydler Method — 1972) and Nageli — 1983 in Table 7 are summarized above.

simplified methods in MAAG "63"/"83" and ANSI/AGMA 6011-J14 (Ref. 1) have fixed or adjusted parameters for that purpose.

Variable parameters that need to be considered for all applications that possibly minimally vary with high- speed gears:

- Elasto-hydrodynamic effect of the oil film dependent on pitch line velocity
- Lubricant employed
- Surface Roughness
- Gear tooth flank temperature
  - ¤ Influence on coefficient of friction
  - ¤ Maximum contact temperature
- Rotor material
  - ¤ Modulus of elasticity
  - ¤ Poisson's ratio

This does not in any way suggest the application of these methods is not valid for other than high-speed applications. Rather, with the inclusion of parameters common with high-speed gears, all methods proved consistent results.

Table 8 AGMA-6011 MAAG method 1983 modified — simplified scuffing criterion method

Ref.	with profile modification /Nominal Risk Cv=1.15				
	Load N/mm^2	Geometry N/mm^2	Margin %	Risk	Tooth Temp °C
1	1019.1	1382.5	26.3%	SAFE	100
2	1960.5	2026.4	3.3%	SAFE	100
3	225.6	508.4	55.6%	SAFE	100
4	1960.5	2026.4	3.3%	SAFE	100
5	978.7	1469.4	33.4%	SAFE	100
6	1328.8	1404.4	5.4%	SAFE	100
7	1394.0	1343.2	-3.8%	AT RISK!	100
8	1441.7	1406.8	-2.5%	AT RISK!	100
9	1136.7	1443.7	21.3%	SAFE	100

Table 9 Scuffing risk according to AGMA 925-A03 — differentiated calculation procedure

AGMA 925-A03 (MAAG Method 1983) — Differentiated Calculation Procedure						
Ref.	per MAAG Simplified Assumption Tooth Temp. = 100°C					
	Scuffing Risk	Risk	Tooth Temp °C	Flash Temp °C	Contact Temp °C	Allowable °C
1	5.0%	Low	100	15.9	115.9	177.4
2	5.0%	Low	100	33.2	133.2	177.4
3	5.0%	Low	100	17.3	117.3	177.4
4	5.0%	Low	100	33.2	133.2	177.4
5	5.0%	Low	100	18.3	118.3	177.4
6	5.0%	Low	100	23.4	123.4	177.4
7	6.5%	Moderate	100	37.2	137.2	177.4
8	5.5%	Moderate	100	34.9	134.9	177.4
9	5.0%	Low	100	22.4	122.4	177.4

Table 10 Scuffing risk according to AGMA 925-A03 — empirical method

AGMA 925-A03						
per Benedict & Kelley Empirical Method						
Ref.	Scuffing Risk	Risk	Tooth Temp	Flash Temp	Contact Temp	Allowable
1	5.0%	Low	92.4	14.9	107.3	177.4
2	6.2%	Moderate	102.8	33.6	136.4	177.4
3	5.0%	Low	93.0	16.1	108.6	177.4
4	6.2%	Moderate	102.8	33.6	136.4	177.4
5	5.0%	Low	93.8	17.5	111.3	177.4
6	5.0%	Low	96.8	22.9	119.7	177.4
7	10.1%	Moderate	105.3	38.1	143.5	177.4
8	7.6%	Moderate	103.9	35.4	139.3	177.4
9	5.0%	Low	96.3	21.9	118.2	177.4.

In the example Table 4 the maximum coefficient of friction, $\mu_{(max)}$, calculated at the highest contact temperature, resulted in the highest values.

A percentage of risk has been created for the ANSI/AGMA 6011-J14 (Ref. 1) method. Currently it is effectively a go-no go assessment. The risk percentage provides a relationship when analyzing comparative data.

Conclusions

AGMA 925-A03 is intended for use for a wide range of gear applications in power, speeds and configuration. This includes a range of lube oil viscosity selected according to the application. The conditions for slow-speed gears — especially splash-lubricated — have an operating oil temperature that does not vary as greatly with the tooth body temperature as high-speed gears. Quality levels of the gearing are quite variable. It is clear that numerous options for assessing scuffing risk exist. Formulas for assessing such risk have been either empirically developed or based on testing. This is particularly true for determining the variable coefficient of friction, which has a large influence on the flash temperature calculation (Ref. 9). In order to apply AGMA 925-A03 for high- speed applications the inlet temperature must be carefully selected. A 70°C inlet temperature was assumed, which provides results consistent with ANSI/AGMA 6011. Lower (or higher) values for oil inlet temperatures may produce unreliable assessment of risk.

The choice of the inlet temperature is based on typical field operating conditions of the referenced examples. Oil supply temperatures for high-speed gears are normally limited to 49°C with an allowable overall temperature rise of 29°C (Ref. 9). The applied inlet temperature will therefore be somewhat less than 78°C. The supply temperature, which 925-A03 applies as the initial parameter in defining the tooth body temperature, may vary significantly. This can be a major divergence between assessing risk in comparing 925-A03 and 6011-J14. To align results consistently when applying 925-A03 to high-speed gears, an oil inlet temperature has been set to approximately 70°C. This selection is based on the thermal network of a gearbox described in Figure 6. An arbitrary choice 70°C seemed reasonable, resulting in a calculated tooth body temperature of approximately 100°C. MAAG "63"/"83" and ANSI/AGMA 6011-J14 (Ref. 1) fixed the tooth body temperature at 100°C.

ANSI/AGMA 6011-J14 is specifically intended for use with high-speed gears that are subject to higher pitch line velocities, which typically employ lubricants with a viscosity range of 32–46 centistokes at 40°C. As previously mentioned, for gear pitch line velocities above 80 m/s, churning loss, expulsion of oil between meshing teeth, and windage loss become important heat sources that should be considered (Ref. 9). Gear quality levels should be consistently high where particular attention must be paid to the influence of thermal deformations, (most notably above 100 m/s), affecting the change in load distribution across the face width and between several teeth (Ref. 9). The calculation assumes addendum modified gears usual for high-speed gearing (Ref. 9).

The calculation procedure developed by MAAG refrains from consideration of absolute flash temperature limits, since their determination is by testing with relatively small gears (such as

FZG testing). These small gears heat up quite differently than gears of larger dimensions that are ultimately employed in service (Ref. 9).

Looking ahead, surface distress due to scuffing or micropitting may be related; they share given parameters that may lead to either distress. Whereas micropitting is a fatigue phenomenon, scuffing is not, but rather occurring instantaneously. It is suggested that scuffing is a condition where the subsequent surface distress occurs when the lambda ratio is entirely in regimes 1 and 2, whereas micropitting occurs when the lambda ratio is in the borderline regimes 2 and 3. High FZG-rated lubricants provide scuffing resistance, but over time fatigue of the surface asperities may result in micropitting.

For more information.

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