Defining the Tooth Flank Temperature in High-Speed Gears

John Amendola, John Amendola III and Robert Errichello

Introduction

When calculating the total contact temperature the tooth flank temperature is as significant as the flash temperature.

$$\theta_{\text{total contact max}} = \theta_{\text{flash max}} + \theta_{\text{tooth flank temp}} \quad (1)$$

Scuffing is likely to occur when

$$\theta_{\text{total contact max}} \geq \theta_S \quad (2)$$

where:

θ_S = the mean scuffing temperature

Currently, in AGMA 925-A03, Equation 91 includes the oil supply or sump temperature θ_{oil}. If spray lubrication is employed, the oil supply temperature is multiplied by 1.2. This refers to the oil supply temperature as the oil inlet temperature to the gear unit.

Clause 6.3 of AGMA 925-A03 states, "The tooth temperature may be significantly higher than the temperature of the oil supplied to the gear mesh." This statement cites a publication by Errichello (Ref. 1), which refers to the gear tooth flank temperature measured by Akazawa (Ref. 2).

The question is whether a multiplier of 1.2 is sufficient for all speeds of gears utilizing a spray lubrication system varying from relatively slow speed gears with pitch line velocities (PLV) < 35 m/s to high-speed gears with PLV up to 200 m/s.

The field referenced examples used in 19FTM24 (Ref. 3) are high-speed units in operation in the field. See Table 1 for a summary of application data. These units provide data for assessing scuffing risk according to three methods: MAAG "63," ANSI/AGMA 6011-J14, Annex B and AGMA 925-A03. Two additional referenced documents, by Akazawa (Ref. 2) and Martinaglia (Ref. 4) report on testing results of single helical high-speed gears that both confirm the gear tooth flank temperatures increase with PLV. These results are compared in Table 4 The steeper slope with increasing PLVs from Martinaglia's paper (Ref. 4) could be caused by gears with lower helix angles and wider face width's having higher axial pumping velocities.

To fully understand the contents of this paper the reader is encouraged to refer to the earlier paper 19FTM24, a version of which also appeared in the March/April 2020 issue of *Gear Technology* (see *www.geartechnology.com/19FTM24* to download the article). The reference data in this paper is based on extensive experimental data listed in the bibliography.

The objective of this paper is to improve the methodology for determining the tooth flank temperature. Two methods are proposed for assessing scuffing risk when applying AGMA 925 for high-speed gears. Both methods provide similar results.

A Brief Review of Scuffing

- When gears are subject to highly loaded conditions and high sliding velocities, the lubricant film may not adequately separate the surfaces. This can cause localized damage to the surface of the gear tooth flanks called "scuffing." Scuffing exhibits itself as a dull matte or rough finish usually at the extreme end regions of the contact path or near the points of a single pair of teeth contact resulting in severe adhesive wear.
- Scuffing is not a fatigue phenomenon and it can occur instantaneously. The risk of scuffing damage varies with the material of the gear, the lubricant being used, the viscosity of the lubricant, the surface roughness of the tooth flanks, the sliding velocity of the mating gear teeth under load and the geometry of the gear teeth.
- Any changes in any of these factors can alter scuffing risk.

Calculation Methods for Determining Tooth Flank Temperature θ_M

The calculation methods for θ_M given herein were each derived from the DIN 3990-4 Standard.

The original calculation for determining θ_M, given in DIN 3990-4, is based on test stand gearboxes in the FZG laboratory. PLV was reportedly limited to 15 m/s.

DIN 3990-4 (flash temp. method)

$$\theta_M = X_S(\theta_{oil} + 0.47\,\theta_{flmax}) \quad (3)$$

where:

$X_S = k_{sump}$ is 1.2 for spray bar lubrication

The equation can be rewritten:

$$\theta_M = k_{sump}(\theta_{oil} + 0.47\,\theta_{flmax}) \quad (4)$$

Note: $\theta_{Bmax} = \theta_M + \theta_{flmax}$

ISO 6336-20

ISO adopted a modified version of the DIN formula as follows:

$$\theta_M = \theta_{oil} + 0.47\,(X_S)\,(X_{mp})\,(\theta_{flm}) \quad (5)$$

where:

θ_M is Tooth flank temperature
θ_{oil} is Oil inlet temperature
X_S is 1.2 for spray lubrication
X_{mp} is 1 for single mesh gears
θ_{flm} is the average flash temperature (SAP–EAP)

Note: SAP = start of active profile; EAP = end of active profile.

This resulted in:

$$\theta_M = \theta_{oil} + 0.564\,(\theta_{flm}) \quad (6)$$

AGMA 925-A03 (Ref. 5)

AGMA 925-A03 had applied the DIN 3990-4 formula with a single value for k_{sump} and multiplied through the equation, which

Table 1 Data table, Field Referenced Inputs

Ref.	Est hrs	helical	type	*a* (mm)	*b* (mm)	*v'* (m/s)	kW	input (rpm)	output (rpm)	module	Z_1/Z_2	*b/d*	*β*
1	>200k	single	increaser	400	236	142.0	10,515	4,831	11,406	6.5	36/85	1.07	10°
2	120	double	decreaser	360	228	112.0	7,915	8,476	4,573	5.5	41/76	0.90	26°30′
3	175k	single	increaser	250	120	118.3	4,096	6,840	13,310	4.5	37/72	0.71	10°
4	160k	single	decreaser	580	502	109.3	37,286	4,670	2,927	6.25	47/75	1.12	10°
5	180k	single	increaser	520	352	142.1	22,670	3,428	10,933	6.5	37/118	1.42	11°
6	200k	single	increaser	780	255	123.0	13,500	1,775	9,951	7.0	33/185	1.08	10°30′
7	150k	double	increaser	610	370	92.7	16,406	1,800	7,636	6.0	33/140	1.59	31°20′
8	150k	single	increaser	509	323	72.6	12,304	1,800	5,606	6.9	35/109	1.31	10°
9	120k	single	increaser	600	270	88.1	9,694	1,800	7,582	5.9	37/163	1.22	10°
10	200k	double	increaser	270	140	43.7	570	1,782	11,616	3.4166	19/124	1.95	24°
11	120K	single	increaser	500	347	175.3	31,905	4,786	11,100	6.3	46/107	1.15	13°30′

Note: $\theta_{Bmax} = \theta_M + \theta_{flm}$ where θ_{Bmax} is maximum contact temperature.

Table 2 AGMA-925-A03 Preset Input Parameters

Oil Type:	Mineral VG-32
FZG Load stage:	fail 6
Scuffing temperature θ_s:	177°C
Oil Temperature:	49°C
surface roughness R_a:	0.50 µm
LSF (load sharing factor):	smooth meshing/with profile modification
Thermal Coefficient of Contact for Steel B_m :	13.796 N/[mm s0.5 K]

fixed the multiplier variable for θ_{flm} to 0.56

$$\theta_M = k_{sump}(\theta_{oil}) + 0.56\,\theta_{flmax} \quad (7)$$

where:

θ_{flmax} is maximum flash temperature along (SAP–EAP)

k_{sump} is 1.2 for spray lubrication

This resulted in:

$$\theta_M = 1.2\,(\theta_{oil}) + 0.56\,\theta_{flmax} \quad (8)$$

The equation should have been rewritten:

$$\theta_M = k_{sump}(\theta_{oil} + 0.47\,\theta_{flmax}) \quad (9)$$

However, if k_{sump} is to be treated as a variable then the original DIN formula needs to be applied as shown in Equation 9. The authors consider Equation 91 in AGMA 925-A03 is only valid when $k_{sump} = 1.2$.

Establishing the Oil Inlet Temperature θ_{oil} Using a Variable Multiplying Factor k_{sump}

The $k_{sump} = 1.2$ was reportedly developed using small test stand gears limited to 15 m/s PLV in a laboratory environment. For an inlet temperature of $\theta_{oil} = 49°C$ the multiplying factor of 1.2 results in a supply temperature of $\theta_{oil} = 59°C$ delivered to the tooth flank. This is considerably less when using MAAG and AGMA 6011 Annex B which fixed the tooth flank temperature at 100°C. To equate the use of the DIN 3990/AGMA 925 equation, a $k_{sump} > 1.2$ is required in order to raise the supply temperature to 70°C. This would generally result in a tooth flank temperature of 100°C which is consistent with MAAG & ANSI/AGMA 6011 Annex B. Assessing scuffing risk for high speed gears using AGMA 925 with the current 1.2 multiplier would result in a false assessment of safety.

AGMA 925-A03 applies the k_{sump} factor as a multiplier of the oil inlet temperature $\theta_{oi\,1}$, whereas ISO 6336-20 does not.

For pitch line velocities less than 35 m/s the ISO approach seems logical as it is expected the gear elements would be supported with antifriction bearings. However, above 35 m/s most gear units are installed with hydrodynamic bearings which are lower in efficiency and contribute heat to the housing structure and in turn add heat to the oil supply temperature θ_{oil}. Therefore, for high-speed gears this document uses the original DIN 3990-4 equation.

This document includes data from the field inspections (Ref. 3) shown in Table 1, and instrumented test gears (Refs. 2, 4) shown in Tables 4a and 4c.

Referenced Gears

Test Gear (Ref. 2)

25,000 HP speed increaser 7656/18689 rpm Single Helical
a: 506.25 mm *b*: 250 mm *v'*: 200 m/s
Temperature measurements using embedded thermocouples in the pinion/gear teeth.

Test Gears (Ref. 4)

Various 21-62 MW speed reducers/increasers Single Helical
21 MW 3000/7625 varying speeds Single Helical
a: 360 mm *b*: 300 mm *v'*: 137 m/s – 148 m/s
Temperature measurements using embedded thermocouples in the pinion
62 MW 2988/1000 Single Helical
a: 1750 mm *b*: 802 mm *v'*: 137 m/s
Temperature measurements using imbedded thermocouples in the pinion

All gearsets described in this document are of a single or double helical configuration. Spur gears have not been considered.

Field References (Ref. 3)

Table 1 is a summary of the inspected gear units in field operation with applied data in assessing scuffing risk.

Table 2 lists preset input parameters for the calculations listed in Table 3.

The values of θ_M in Table 3 differ from those given in 19FTM24 (Ref. 3) for the same field references. The values in 19FTM24 (Ref. 3) applied a fixed value for $k_{sump} = 1.2$ using a very high oil supply temperature of 70°C, whereas Equation 10 in this document employs a variable value for k_{sump} with normal oil inlet temperature of 49°C.

The actual measured tooth flank temperatures listed in Tables 4a and 4b are taken from test data (Refs. 2, 4). They indicate k_{sump} increases with increasing PLV. The Table 4a and 4b values were compared to the field references of similar pitch line velocities and a value for k_{sump} was applied to the examples in Table 3 to match the measured values in test data (Refs. 2, 4). The calculated tooth flank temperatures θ_M listed in Table 3 are summarized in Table 4c for comparison with full-size test gears (Refs. 2, 4). The comparison shows comparable θ_M values. They are grouped in stepped values of k_{sump} as follows:

$k_{sump} = 1.0$ for splash lube

$= 1.2$ for spray lube with gears utilizing antifriction bearings

$= 1.35$ for PLV 35–50 m/s

$= 1.38$ for PLV 50–90 m/s

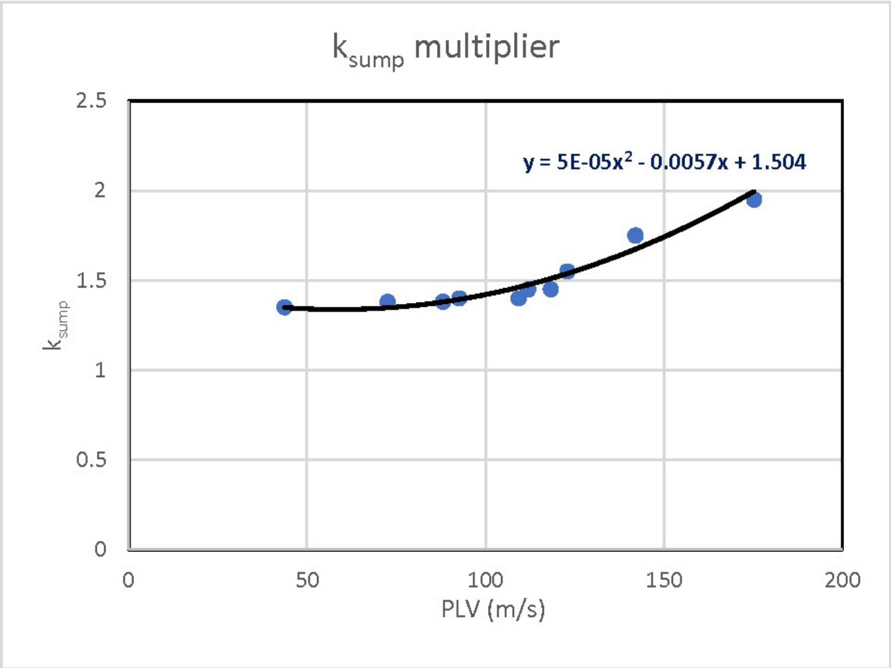

Figure 1

Table 3

v′ range (m/s)	Case	v′ (m/s)	Scuffing risk	Risk	Tooth Temp (°C) θ_m	Flash Temp (°C)	Contact Temp (°C)
k_{sump} = 1.35 (DIN)*							
35≥50	10	43.7	5.0%	low	75.3	14.5	91.5
k_{sump} = 1.38 (DIN)*							
50≥90	8	72.6	5.0%	low	89.1	33.1	122.3
	9	88.1	5.0%	low	80.3	19.6	99.9
k_{sump} = 1.40 (DIN)*							
90≥110	4	109.3	5.0%	low	90.4	33.1	123.5
	7	92.7	5.0%	low	92.9	37.0	129.9
k_{sump} = 1.45 (DIN)*							
110120	2	112.0	5.1%	moderate	96.8	37.8	134.6
	3	118.3	5.0%	low	75.6	6.7	82.3
k_{sump} = 1.55 (DIN)*							
120≥130l	6	123.0	5.0%	low	92.0	22.0	108.1
k_{sump} = 1.75 (DIN)*							
130≥145	1	142.0	5.0%	low	99.2	16.3	115.5
	5	142.1	5.0%	low	108.2	27.4	132.6
k_{sump} = 1.95 (DIN)*							
>170	11	175.3	23.7%	moderate	120.0	26.3	158.3

* k_{sump} calculated per DIN 3990-4 (flash temp. method) per Equation 4

Table 4a

v′ (m/s)	θ_m (°C)
100	80
110	85
120	90
130	95
140	100
150	105
160	110
170	115
180	120
190	125
200	130

4a Note: Measured test gear values (Ref. 2)

Table 4b

v′ (m/s)	θ_m (°C)
100	70
115	85
134	101
145	111
151	117
160	125

4b Note: Measured test gear values (Ref. 4)

Table 4c

Ex. Ref.	v′ (m/s)	DIN (X_s) k_{sump}	θ_m (°C)
10	43.7	1.35	75.3
8	72.6	1.38	89.1
9	88.1	1.38	80.3
4	109.3	1.40	90.4
7	92.7	1.40	92.9
2	112.0	1.45	96.8
3	118.3	1.45	75.6
6	123.0	1.55	92.0
1	142.0	1.75	99.2
5	142.1	1.75	108.2
11	175.3	1.95	120.0

4c Note: field (calculated values) (Ref. 3)

= 1.40 for PLV 90–110 m/s
= 1.45 for PLV 110–120 m/s
= 1.55 for PLV 120–130 m/s
= 1.75 for PLV 130–145 m/s

Values above 145 m/s should be based on field experience or applying the curve in Figure 1.

A plot for k_{sump} versus PLV can be applied as an option to a table as shown in Figure 1.

This curve is based on the references listed in Table 4c resulting in the following equation:

$$k_{sump} = 0.00005(v')^2 - 0.0057(v') + 1.504 \quad (10)$$

Verification of the Calculated Values to Measured Test Values

For further verification, measured values for pinion tooth flank temperatures from Tables 4a and 4b and the calculated values from Table 3 are plotted against PLV in Figure 2. By plotting all values the following averaging relationship can be defined as follows:

$$\theta_M = 0.0021(v')^2 - 0.1188(v') + 77.088 \quad (11)$$

Similar adjustment can be applied to C_w in the formulation used in Annex B of ANSI/AGMA 6011-J14 (Ref. 7).

Table 5 compares the results from Equation 11 for θ_M with those calculated with k_{sump} listed in Table 4c.

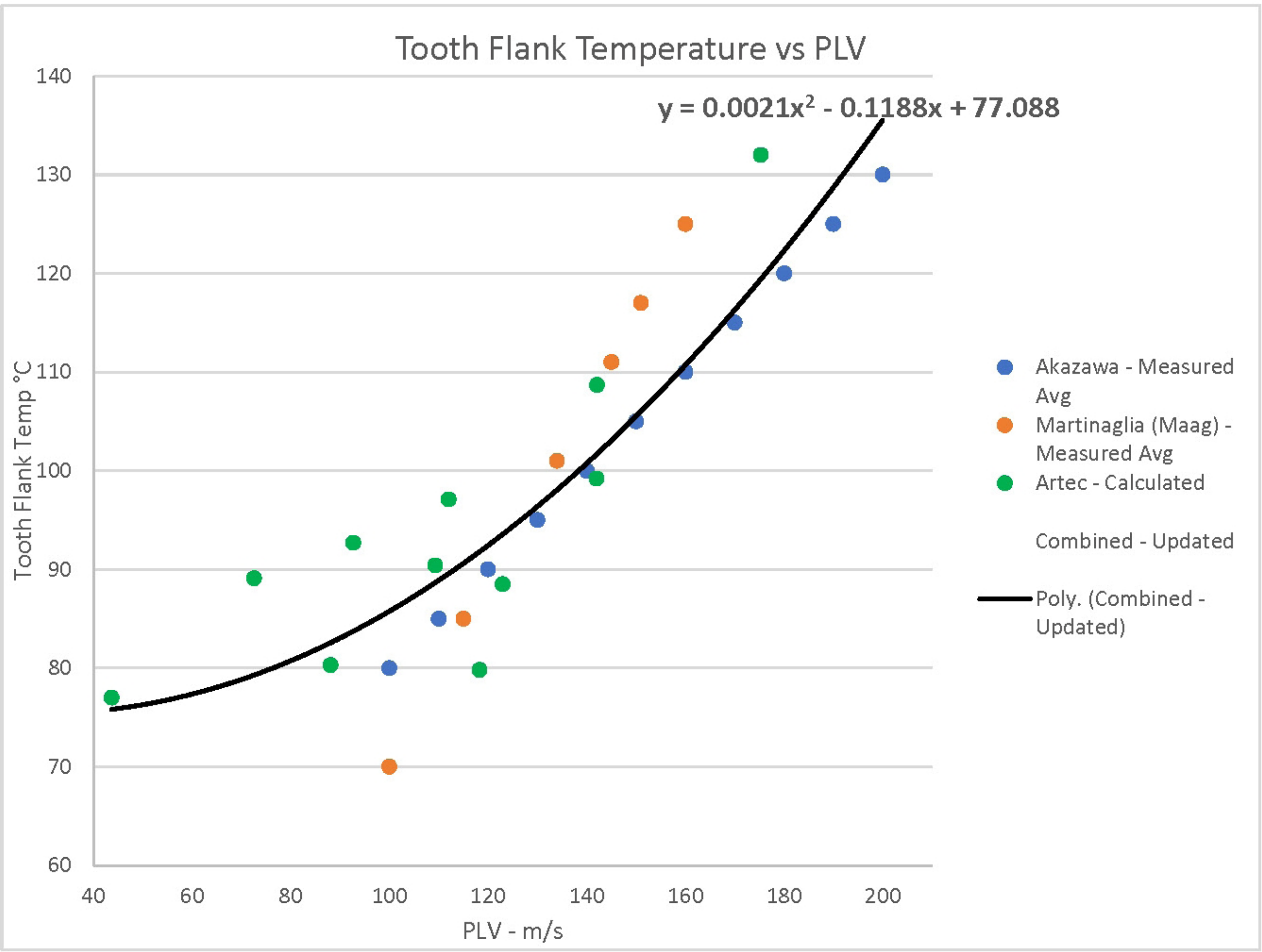

Figure 2

Table 5

Field Ex. Ref.	v' (m/s)	DIN(X_s) for k_{sump}	Equation 10 θ_m (°C)	Equation 11 θ_m (°C)
10	43.7	1.35	75.3	75.9
8	72.6	1.38	89.1	79.5
9	88.1	1.38	80.3	82.9
4	109.3	1.40	90.4	89.2
7	92.7	1.40	92.9	84.1
2	112.0	1.45	97.1	90.1
3	118.3	1.45	75.6	92.4
6	123.0	1.55	92.0	94.2
1	142.0	1.75	99.2	102.6
5	142.1	1.75	108.2	102.6
11	175.3	1.95	120.0	120.8

Table 5a

Field Ex. Ref.	v' (m/s)	ISO 6336-20 X_s	Equation 10 θ_m (°C)	Equation 11 θ_m (°C)
10	43.7	3.88	75.3	75.9
8	72.6	2.58	89.1	79.5
9	88.1	3.41	80.3	82.9
7	92.7	2.54	92.7	84.1
4	109.3	2.66	90.4	89.2
2	112	2.68	97.1	90.1
3	118.3	8.48	75.6	92.4
6	123	4.15	92	94.2
1	142	6.53	99.2	102.6
5	142.1	4.61	108.2	102.6
11	175.3	10.42	120	120.8

These values for θ_M are reasonably consistent. References 2, 6 and 11 are all references where tooth surface distress had been evident. Corrective action was required to arrest the problem.

Note ISO 6336-20 Equation 5 differs significantly from DIN 3990-4 Equation 3 because the oil supply temperature θ_{oil} is not adjusted by X_S values as proposed in Table 3. Therefore, a different set of X_S values described by PLV levels will be required for application with Equation 4. However, in using ISO 6336-20, Equation 11 is applicable.

Using ISO 6336-20 Equation 5 the values for X_S are adjusted for use of the equation.

Equation 5 from ISO 6336-20 produces a scattering of values for X_S versus PLV levels which cannot result in a curve similar to Equation 10. Equation 5 from ISO 6336-20 produces a scattering of values for X_S versus PLV levels which cannot result in a curve similar to Equation 10.

Determining Value for θ_M

Equations 10 and 11 are both suitable equations to calculate a value for θ_M in AGMA 925.

Method A

The value for k_{sump} obtained from Equation 10 can be applied in Equation 9 to obtain a value for θ_M.

Method B

Equation 11 directly calculates θ_M. It should be noted when using this method the applied data is based on oil supply temperatures over a limited range from 40˚C–70˚C. Most of the Table 3 applications had a supply oil temperature of 43°C–55°C. Therefore, the reliability of Method B where a lube oil supply temperature is beyond this range may be somewhat compromised. Furthermore, Method B should only be applied with gears utilizing hydrodynamic bearings. The Table 3 gears employed sump pans to prevent windage affecting the outflow of oil through the discharge port(s). Additional shrouding of the gear rotors that can mitigate tooth flank temperatures is not considered here. Tooth flank temperatures with shrouded gears should be based on field individual field experience.

Factors that Influence Tooth Flank Temperature

In all the high-speed examples discussed, the gears employed hydrodynamic bearings. These bearings are less efficient than roller bearings used in FZG testing. The heat generated in hydrodynamic bearings is significant. Martinaglia (Ref. 4) reported measured values of approximately 30% of the gear power losses was in the bearings. Temperature range as measured in journal bearing RTDs are typically in the range of 70°–90°C. Consequently, the bearing journals absorb heat. The question is, does the energy absorbed by the journals, particularly higher in the pinion, contribute to the tooth flank temperature. During the early Nineties, MAAG developed special turbo gears whereby the gears operated in a near-vacuum. Tests were conducted on a full-sized 65 MW turbo gear (Ref. 8). Temperature measurements in the gearing were recorded for both conventional and near-vacuum modes. The temperature difference was reportedly approximately 40°C lower in the vacuum mode.

It can be stated the requirements to increase the k_{sump} factor in high-speed gears is primarily the result of the operating windage. Martinaglia had suggested, “in especially fast running gears, the frictional heat developed in the bearings also passes via the shaft stub into the pinion body proper.” Furthermore, the MAAG HET test results have shown this to be a significant influence. More recently there have been some high-capacity gears designed with a shroud that closely surrounds the gear set. The shroud is externally cooled, thereby minimizing the oil flow required in the gear mesh for lubricating purposes only. This in turn reduces the pumping losses in the mesh resulting in an increase in operating efficiency. This also mitigates the adjustment in the lead modification to compensate for thermal deformation.

There are some variable factors that result in minor differences in the tooth flank temperatures plotted in Figure 2. Length of the tooth face width, size of the module, helix angle of the gear and internal housing dimensions can influence the windage behavior. Test Gear 4 temperature plots are steeper than Test Gear 2 temperature plots most likely due to lower helix angles. These differences have a minor influence on the variations in tooth flank temperatures. There are infinite combinations of these parameters making it difficult to assess their influence on the values of θ_M. This is shown by the varied plots of the field references where these parameters are all from different gearboxes. Nevertheless, PLV has the single largest influence on operating tooth flank temperatures.

However, where windage is low, the number for k_{sump} is lower. The gear References 2 and 4 and “Field Reference 3" indicate there are small changes for the k_{sump} number. For PLV < 35 m/s, k_{sump} may not be less than 1.35. It is not in the scope of this document to evaluate values for k_{sump} where PLV < 35 m/s. The AGMA threshold for high-speed gears applies for pitch line velocity above 35 m/s. The determination of k_{sump} requires additional research where operating PLVs are between 15–35 m/s. Nevertheless, it seems improbable there could be a significant change between $k_{sump} = 1.2$ up to 15 m/s and $k_{sump} = 1.35$ up to 35 m/s.

References 2 and 4 and the Table 3 applications were equipped with hydrodynamic bearings whereas the FZG test gears employed antifriction bearings. Power losses in gears with hydrodynamic bearings may influence the gear tooth flank temperatures from heat absorbed by the bearing journals and transmitted into the main body of the gear elements. Therefore, suggested values for k_{sump} are:

$k_{sump} = 1.35$ for gears where PLVs are < 35 m/s when equipped with hydrodynamic bearings.

$k_{sump} = 1.20$ for gears where PLVs are < 35 m/s when equipped with antifriction bearings.

Note: ANSI/AGMA 6011-J14 references high-speed gears with hydrodynamic bearings. Roller bearings are occasionally used in special cases.

Conclusions

1. AGMA 925-A03 Equation 91 should be limited to PLV < 35 m/s for gears equipped with antifriction bearings.
2. Method A for calculating k_{sump} in Equation 10 should be used to calculate θ_M in Equation 9 and added to AGMA 925.

3. Method B for calculating θ_M using Equation 11 should be added to AGMA 925.
4. The fixed k_{sump} value in AGMA 925-A03 is not suitable for assessing scuffing risk for high-speed gears and will lead to an erroneous value for safe scuffing assessment.

References

1. Errichello, R. *Friction, Lubrication and Wear of Gears,* ASM Handbook, Vol. 18, Oct. 1992 pp. 535–545.
2. Akazawa, M. *Full Scale Test of High Speed, High Powered Gear Unit:* August 18, 1980.
3. Amendola, Amendola, Errichello; *19FTM24 – Calculated Scuffing Risk: correlating AGMA 925-A03, AGMA 6011-J14 and Original MAAG Gear Predictions:* October, 2019.
4. Martinaglia, L. *Thermal Behavior of High-Speed Gears and Tooth Corrections for Such Gears*; Oct. 11, 1972.
5. AGMA 925-A03, *Effect of Lubrication on Gear Surface Distress*; March 13, 2003.
6. API 613-5, *Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Units*. Sec. 2.8.2; December, 2002.
7. ANSI/AGMA 6011-J14, *Specification for High Speed Helical Gear Units*; Annex B, December 19, 2019.
8. Deeg, Thomas; The MAAG High Efficiency HET Turbo gears; 1996.

John B. Amendola is an executive officer and chairman of the board and Chairman of the Board of Artec Machine Systems where he has been working for 50 years. Prior employment was with Western Gear, Texaco & Boeing Co. where he operated a full load four square locked torque test stand for helicopter gears. He is currently an active member of AGMA Helical Gear Rating & Lubrication Committees, active chairman of AGMA Enclosed High Speed Units Committee, a member of the US TAG to ISO TC60 and a member of the American Petroleum Institute 613-6 Standards Committee. John holds a Bachelor of Mechanical Engineering from Villanova University and a Master of Science degree in Mechanical Engineering from (NYU) Brooklyn Polytechnic Institute. Amendola is a recipient of the AGMA TDEC Award.

John B. Amendola III, is the president of Artec Machine Systems where he has been working for 29 years. Prior experiences include: 1 year at Maag Gear, Zurich Switzerland, with assembly, testing, field services group for high speed gear applications; and 2 years at SUNY Maritime College, Assistant Professor Engineering Department training of engineering undergrads in Marine/Mechanical operations. He is an active member AGMA Helical Gear Rating Committee and holds a Bachelor of Science in Marine Engineering from Maine Maritime and a Professional Engineering License for Mechanical Engineering. John III also maintains a USCG unlimited license as a senior Marine Engineering Officer and Commander Retired US Navy Reserve.

Robert Errichello heads his own gear consulting firm, GEARTECH, and is founder of GEARTECH Software, Inc. He has over 57 years of industrial experience. He has been a consultant to the gear industry for the past 44 years and to over 50 wind turbine manufacturers, purchasers, operators, and researchers. A graduate of the University of California at Berkeley, Errichello holds BS and MS degrees in mechanical engineering and a Master of Engineering degree in structural dynamics. He is a member of several AGMA Committees, including the AGMA Gear Rating Committee, AGMA/AWEA Wind Turbine Committee, ASM International, ASME Power Transmission and Gearing Committee, STLE, NREL GRC, and the Montana Society of Engineers. He is technical editor for GEAR TECHNOLOGY and STLE Tribology Transactions. Errichello is recipient of the AGMA TDEC Award, the AGMA E.P. Connell Award, the AGMA Lifetime Achievement Award, the STLE Wilbur Deutch Memorial Award, the 2015 STLE Edmond E. Bisson Award, and the AWEA Technical Achievement Award.