GEAR SCUFFING: POWER DISSIPATION AND MASS TEMPERATURE

J. Castro[1(*)], J. Seabra[2]
[1] Departamento de Engenharia Mecânica, Instituto Superior de Engenharia, Instituto Politécnico do Porto, Portugal (mdc@isep.ipp.pt)
[2]Departamento de Engenharia Mecânica e Gestão Industrial, Faculdade de Engenharia da Universidade do Porto, Portugal (jseabra@fe.up.pt)

ABSTRACT
Experimental gear scuffing results were obtained in the FZG test rig for wide ranges of the applied torque, tangential speed, base oil viscosity and bath oil temperature, and for FZG type A and type C gears.
A scuffing criterion for gears lubricated with base mineral oils was developed, proposing the existence of gear mass temperatures which are critical for each lubricant (viscosity grade). The scuffing criterion shows very good correlation with experimental results.
The dynamic viscosity of the oils at those critical mass temperatures is constant that permits the determination of other critical temperatures for other gear mineral oils without need additional scuffing tests.
KEY WORDS: friction, wear, scuffing, gears, mineral oils.

1.- INTRODUCTION
Traditional scuffing criteria are not able to explain many of the phenomena observed, for instance, how the different geometries of the type A and type C FZG gears and how the oil bath temperature affect the scuffing load carrying capacity of FZG gears lubricated with additive free base mineral oils.
For restricted and particular operating conditions all scuffing criteria seem to be valid, however no one seemed appropriated for the wide ranges of operating conditions considered. A preliminary analysis of the experimental results shows that the oil bath temperature and the friction coefficient between gear teeth have a very significant influence on the scuffing load carrying capacity of the FZG gears.
An expression for the friction coefficient, μ_{FZG} (1) in mixed contact lubrication regime that generally occurs between FZG gear teeth was developed taking into account the results published by Hohn e al [1] obtained from experimental measurements of the power loss in FZG type C gears lubricated with mineral oils [2].

$$\mu_{FZG} = 0{,}0257\frac{P_0^{0,18}}{V_R^{0,26}}\eta_0^{-0,05}\left(\frac{Ra}{a}\right)^{0,25} \qquad (1)$$

This friction coefficient, is obtained at each contact point of the gear meshing line, function of the maximum hertzien pressure, P_0, the rolling speed, V_R, the oil dynamic viscosity, η_0, at the bath oil temperature, T_0. The surfaces composed average roughness, Ra and the centre distance, a.

2.- EXPERIMENTAL SCUFFING RESULTS EQUATIONS

Table 1 shows the results of scuffing tests performed in a FZG test machine [3]. Two FZG gear geometries were used: type A gears, 10 mm and 20 mm wide (referenced as A10 and A20 FZG gears, respectively) and type C gears, 14 mm wide, (referenced as C14 FZG gears) [3]. Six base oils, with 4 different viscosity grades, were tested. The most important characteristics of the gears and oils used are presented in appendix 1.

Several different types of tests were performed: Variable torque tests (at constant oil bath temperature and rotating speed), variable temperature tests (at constant torque and rotating speed), variable speed tests (at constant torque and oil bath temperature) and also some standard load carrying capacity tests, according to DIN 51354. In all tests K_{FGZ} is the FZG load (torque) stage, n is the rotational speed of the wheel (rpm) and T_0 is the oil bath temperature (ºC). There is almost total compatibility between these different test procedures, which is very good for the results analysis. The gear geometry and the lubricant properties are presented in appendix I.

Table 1 – Gear scuffing conditions / results.

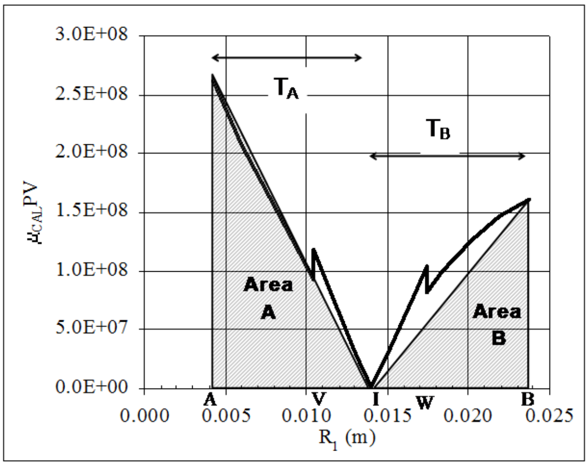
Test Nº	Oil Ref.	Gear type	Temp (ºC)	n (rpm)	K_{FZG}
1	68I	A20	100	750	10
2	68I	A20	100	1500	6
3	68II	A10	75	2500	7
4	68II	A10	90	2500	5
5	68II	C14	100	3000	5
6	68II	C14	100	1500	8
7	150II	A10	85	2000	9
8	150II	A10	90	2000	8
9	150II	A10	95	2500	6
10	150II	A20	100	2000	10
11	150II	C14	100	3000	7
12	150II	A20	105	1500	9
13	150II	A20	105	2250	7
14	150II	A10	105	1500	7
15	150II	A10	105	1750	6
16	150II	A10	105	2500	6
17	150II	C14	110	3000	6
18	150II	C14	110	1500	10
19	220I	A20	110	3000	8
20	220I	A20	118	2250	7
21	220I	A20	119	1500	8
22	220I	A20	120	750	12
23	220I	A20	120	1500	7
24	220I	C14	120	1500	10
25	220I	C14	120	3000	7
26	220II	A10	100	3000	8
27	220II	A10	110	2000	9
28	220II	C14	120	3000	8
29	680I	A20	143	1500	12
30	680I	A20	150	1500	9
31	68I	A20	75	3000	5
32	68I	A20	88	3000	3
33	68I	A20	100	3000	3
34	68I	A20	113	1500	4
35	68I	A20	130	1500	3
36	68I	A20	130	3000	2
37	150II	A20	105	3000	5
38	150II	A20	120	1500	8
39	220I	A20	120	2250	3
40	220I	A20	120	3000	5

3.- THE FRICTION POWER INTENSITY AND GEAR GEOMETRY

A scuffing criterion for FZG gears (FZG: Gear Research Centre at Technical University Munich) lubricated with base oils is proposed based on several different criteria: the traditional PVT scuffing criteria (Alman criteria) [4] is adequate when the aim is to compare the performance of different gear geometries; the FPI (Friction Power

Intensity) scuffing criteria, together with the friction coefficient developed, is adequate to evaluate the influence of the torque (or contact pressure) and of the tangential velocity in scuffing [5]. The criteria proposed adopted the integral concept from integral temperature criteria applied to FPI, resulting in critical oil bath temperatures or gear mass temperatures, above which the lubricant has no longer the ability to generate an hydrodynamic film and scuffing may occur for non-severe conditions, in the case of base oils without anti-wear or extreme pressure additives.
Figure 1 shows the FPI (μ_{FZG}-P_0V_S) applied for gear type A and C. In scuffing conditions, the area A of figure 1 is equal to area B of figure 2, which means the integral of FPI is very important for scuffing analysis and permits to compare scuffing in different gear geometries.

Another important aspect is the zone where the scuffing occur is also predicted by this parameter, that is in the beginning of tooth contact in type C gears and the end of tooth contact in type A gears.



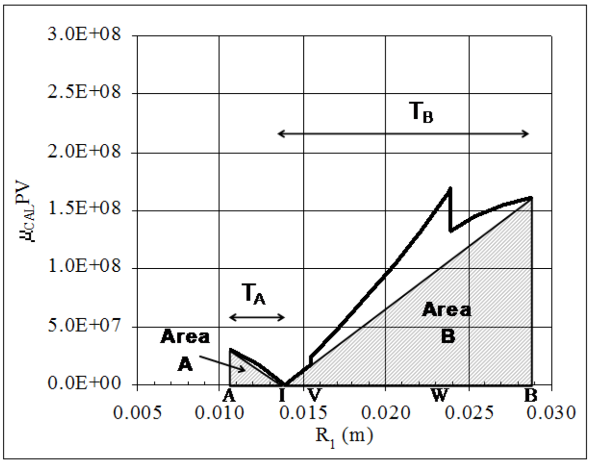


Figure 1 - Variation of the friction power intensity (μ_{FZG}-P_0V_S) along the meshing line of a type C FZG gear (left) and type A (right).

This integral FPI, or each area, could be calculated approximately from the product of maximum FPI, which occurs at the point A or the point B in the gear meshing line, for the distance between those points to the pitch point (I), respectively T_A or T_B.

4.- THE SCUFFING CRITERIA BASED ON POWER DISSIPATION AND TEMPERATURE

Figure 2 shows the integral FPI plotted against the corresponding oil bath temperatures in scuffing conditions, for the different mineral base oils considered. In all cases the integral FPI product (at scuffing) have a linear decrease with the oil bath temperature increase. Figure 2 also shows that behaviour is consistent with both types (A and C) of FZG gears.
This linear behaviour could be represented by an expression (2) that predicts a critical temperature for each lubricant, where there is no power dissipation possibility or no load carrying capacity, where the integral FPI is equal zero:

$$T_{CR} - T_0 = \frac{\mu_{FZG} P_0 V_S T}{Cp} \qquad (2)$$

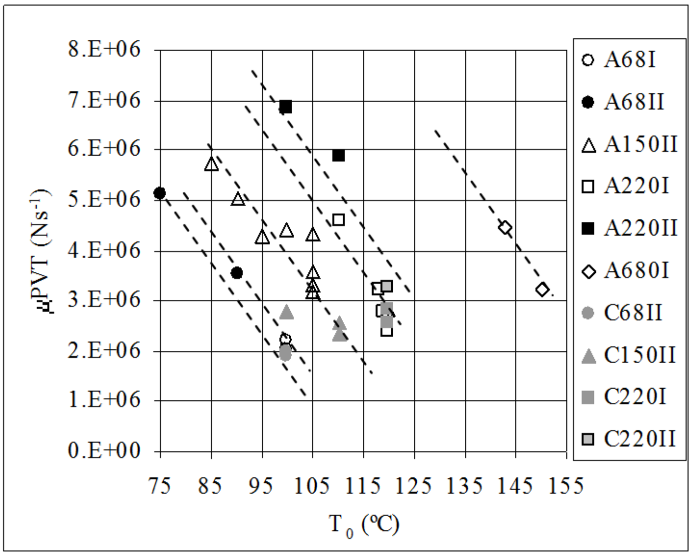


Figure 2 - Integral FPI (μ_{FZG}-P_0V_ST) vs oil bath temperature, T_0.

Figure 3 shows the correlation between the criteria proposed and the experimental results and in practical terms fully agree with experimental results. For another point of view the criteria predicts critical gear mass temperatures that leads to scuffing, that is reinforced by the constant *C* presented in the criteria expression (2) that have thermal conductivity units.

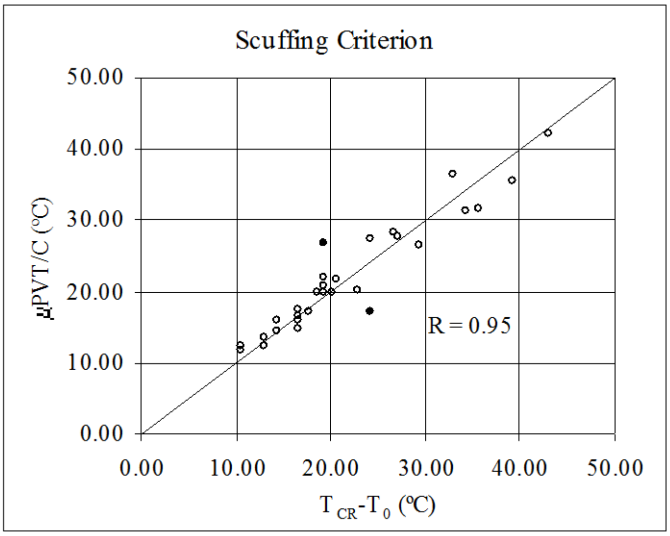


Figure 3 - Correlation between experimental and analytical results.

5.- THE SCUFFING CRITERIA AND GEAR MASS TEMPERATURE

In fact, the critical temperature can be considered as a critical mass temperature, or the temperature of the surface at the beginning of the mesh. Figure 4 shows the heat generation along the meshing line for a FZG type A pinion. The point A is the beginning of gear tooth meshing and point B is the last meshing point between two contacting teeth. Point I is the pitch point, where no friction power is generated. Assuming that the heat generated during gear teeth meshing is mainly evacuated by heat conduction (*Cp* is the thermal conductivity of the gears) through the gear teeth surfaces, the general expression for the heat conduction is,

$$\Delta T = \frac{FPI}{Cp} \Delta z \quad (3)$$

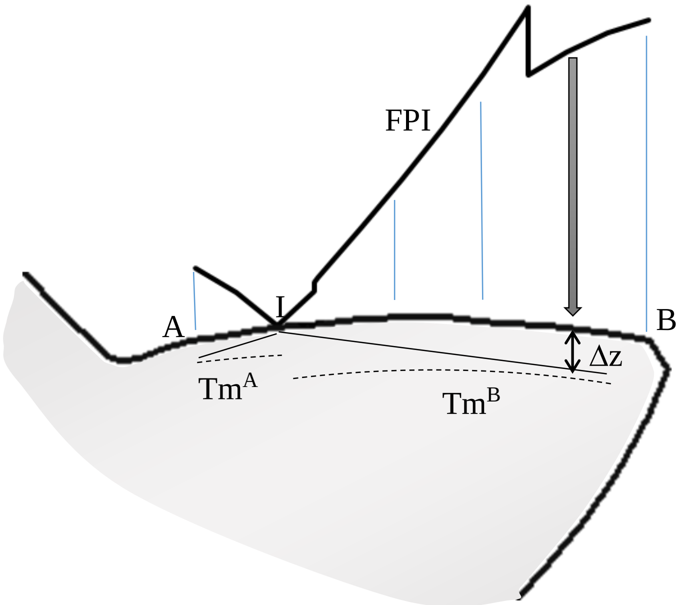


Figure 4 - Heat generation and mass temperature in a FZG type A pinion.

where Δz is the depth from the contact point on the tooth surface to the point in subsurface where the maximum subsurface temperature occurs, along the meshing line, as presented in figure 4. The mass temperature, Tm is the consequence of the heat power input through the surface, assuming for simplification that this temperature increases proportionally to the distance between the pitch point and the contact point on the meshing line.

An additional simplification was used, considering that the mass temperatures in the approach segment $\overline{AI}$, Tm^A, and in the recess segment $\overline{IB}$, Tm^B, are constant but different from the each other. In case of FZG type A gears, shown in Figure 4, it is clear that the heat generation is higher along the recess segment $\overline{IB}$, explaining why tooth scuffing always occurs in this zone. However, in the case of type C gears, the heat generation is higher in the approach segment, and tooth scuffing always start in this zone, although sometimes it may extend to the recess zone, where the heat generation is smaller but still very significant.

The power dissipation intensity, FPI, is given by

$$FPI = \mu_{FGZ}.\frac{2}{3}.P_0 V_S \quad (4)$$

The difference between the subsurface maximum temperature and the surface temperature, reflects the difference between mass temperature and bath oil temperature, that is

$$\Delta T = T_m - T_0 \quad (5)$$

The mass temperature at the beginning of the mesh, could be rewritten as,

$$T_m = T_0 + \frac{FPI}{Cp}\Delta z = T_0 + \frac{\mu_{FZG}\frac{2}{3}P_0.V_S.}{Cp}\Delta z \tag{6}$$

When the mass temperature, reaches a critical value that depends on oil viscosity, scuffing may occur for any conditions, meaning that film generation is not possible, and scuffing occurs even for light loads.

For a steel thermal conductivity of $C_p = 52 Wm^{-1}K^{-1}$, and considering the results from the original criteria, eq. (2), the values obtained for Δz are function of the distance from the pitch point to the contact point, T,

$$\Delta z = 0{,}000483\, T \tag{7}$$

According to equation (7), the maximum subsurface temperature is about 71μm below the surface in type A gears, and about 47μm in type C gears.

For the oils tested, the critical mass temperatures, Tm, are presented in Table 2, which also shows the corresponding dynamic viscosities at those critical mass temperatures. It can be noticed that the viscosity at those critical temperatures are very close, meaning that is possible to associate these temperatures to a constant "scuffing" viscosity (for mineral oils without no anti-wear additives).

Table 2 – Critical mass temperatures (ºC) and dynamic viscosity (cP) for each oil.

	68I	68II	150II	220I	220II	680I
Tm^{CR}	113,0	110,6	124,3	136,7	142,9	170,2
η	5,1	5,5	5,8	6,2	5,6	5,8

A dynamic viscosity of 5,7cP is the average value obtained from these results. For such dynamic viscosity, Table 3 presents the calculated critical mass temperature for each oil. Comparing Tables 2 and 3, the difference between these critical temperatures are very small, enhancing the potential of this "scuffing" viscosity, for the determination of other critical temperatures for other gear mineral oils without anti-wear and friction modifiers.

Table 2 – Critical mass temperatures for constant oil dynamic viscosity.

	68I	68II	150II	220I	220II	680I
Tm^{CR}	108,0	109	125,7	141,5	141,8	171,1

This approach could be extended other oil formulations, and in future works can be experimentally validated. Another important future work is to calculate accurately the effective mass temperature and eventually considering that mass temperature is not constant along the contact path, analysing deeply the conductive heat transfer from gear tooth surface to subsurface and convection heat transfer from surface to the oil sump.

6.- CONCLUSIONS

1. A scuffing criterion for gears lubricated with base mineral oils was developed, merging together the concepts of the Friction Power Intensity and of the Integral Temperature Criteria.
2. The new scuffing criterion proposes the existence of critical gear mass temperatures, which are characteristic of each lubricant (viscosity grade). The difference between the Critical Mass Temperature and the oil bath temperature is proportional to the energy intensity dissipated along the gear meshing line.
3. The new scuffing criterion shows very good correlation with experimental results obtained with FZG type A and type C gears lubricated with base mineral oils of 4 different viscosity grades. In global terms from 40 tests performed, 34 are fully justified by the present scuffing criterion, 4 are justified by insufficient running-in and only 2 are slightly away from the criterion prediction.
4. For the critical mass temperatures, the dynamic viscosity of the oils are very near, that means that is possible to associate this temperatures to a constant viscosity that enhances the potential of this critical viscosity, for the determination of other critical temperatures for other gear mineral oils without anti-wear and anti-friction modifiers.

REFERENCES

[1] B.-R. Hohn, K. Michaelis and A. Doleschel, “Frictional Behaviour of Synthetic Gear Lubricants”, Tribology Research: From Model Experiment to Industrial Problem, G. Dalmaz et al. (Editors), 2001 Elsevier Science B. V.

[2] J. Castro and J. Seabra, Coefficient of friction in mixed film lubrication: Gear versus twin-disc, Proc. Instn of Mech. Engrs, Part J: Journal of Engeneering Tribology, 2007, 221 (J3), 399-411, doi:10.1243/13506501.JET256.

[3] Winter, H. and Michaelis, K. ”FZG Gear Test Rig - Description and Test Possibilities”. Co-ordinate European Council Second International Symposium on “The Performance Evaluation of Automotive Fuels and Lubricants”. Wolfsburg, West Germany, June 5-7, 1985.

[4] Henriot, G. ”La Lubrification Industriele - La Lubrification de Engrenages”. Tome 1 - Transmissions, Compresseurs, Turbines. Publications de l’Institute Français du Pétrole. Éditions Technip, 1984, pg. 297 - 385.

[5] J. Castro e J. Seabra, Global and local analysis of gear scuffing tests using a mixed film lubrication model, Tribology International, Vol. 41/4 (2008), pp. 244-255, doi:10.1016/j.triboint.2007.07.005.

Appendix 1 – Gears and lubricants characteristics

A1. Gears

FZG type A and type C gears are made of case hardened 20 MnCr 5 steel. Table A1 shows some of the geometric characteristics of these gears. The contact ratio ε_α=1.36 in type A gears and 1.46 in type C gears.

Table A1 – Geometric characteristics of FZG type A and type C gears.

Parameter	Symbol	Pinion		Wheel	
		Type A	Type C	Type A	Type C
Working centre distance	a'	91.5			
Module	m	4.5			
Working pressure angle	α_W	20			
Number of teeth	Z	16		24	
Profile shift factor	x	73.20		109.80	
Pitch diameter	d_w	88.68	82.64	112.5	118.5
Addendum diameter	d_{aw}	88.68	82.64	112.5	118.5
Tooth flank roughness	Ra	0.35	0.30	0.30	0.30

A2. Lubricants

The lubricants tested are mineral parafinic base oils without any kind of additives, in particular, extreme pressure (EP) and anti-wear (AW) additives.

Table A2 shows some of the lubricants characteristics. They are divided in two groups (I and II). Group II pretends to be better base oils more resistant to scuffing.

Table A2 – Lubricant characteristics.

Reference	Kinematic Viscosity - cSt		Specific mass - Kg/m^3
	ν_{40} (40ºC)	ν_{100} (100ºC)	ρ_{15} (15ºC)
68I	72.8	8.3	888
68II	69.5	8.5	887
150II	148.0	14.0	893
220I	236.8	18.5	900
220II	211.9	18.3	897
680I	680	37.7	921