Emerald Insight

Industrial Lubrication and Tribology

Pitting load carrying capacity under increased thermal conditions

Bernd-Robert Höhn Klaus Michaelis Hans-Philipp Otto

Article information:

To cite this document:
Bernd-Robert Höhn Klaus Michaelis Hans-Philipp Otto, (2011),"Pitting load carrying capacity under increased thermal conditions", Industrial Lubrication and Tribology, Vol. 63 Iss 1 pp. 11 - 16
Permanent link to this document:
http://dx.doi.org/10.1108/00368791111101795

Downloaded on: 21 September 2016, At: 05:05 (PT)
References: this document contains references to 9 other documents.
To copy this document: permissions@emeraldinsight.com
The fulltext of this document has been downloaded 446 times since 2011*

Users who downloaded this article also downloaded:

(2011),"Influence factors on gearbox power loss", Industrial Lubrication and Tribology, Vol. 63 Iss 1 pp. 46-55 http://dx.doi.org/10.1108/00368791111101830

(2008),"Influence of lubricant type in gear scuffing", Industrial Lubrication and Tribology, Vol. 60 Iss 6 pp. 299-308 http://dx.doi.org/10.1108/00368790810918695

Technical University of Munich

Access to this document was granted through an Emerald subscription provided by emerald-srm:194764 []

For Authors

If you would like to write for this, or any other Emerald publication, then please use our Emerald for Authors service information about how to choose which publication to write for and submission guidelines are available for all. Please visit www.emeraldinsight.com/authors for more information.

About Emerald www.emeraldinsight.com

Emerald is a global publisher linking research and practice to the benefit of society. The company manages a portfolio of more than 290 journals and over 2,350 books and book series volumes, as well as providing an extensive range of online products and additional customer resources and services.

Emerald is both COUNTER 4 and TRANSFER compliant. The organization is a partner of the Committee on Publication Ethics (COPE) and also works with Portico and the LOCKSS initiative for digital archive preservation.

*Related content and download information correct at time of download.

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

Pitting load carrying capacity under increased thermal conditions

Bernd-Robert Höhn, Klaus Michaelis and Hans-Philipp Otto
Gear Research Centre, FZG, TU München, Munich, Germany

Abstract

Purpose – The purpose of this paper is to make an attempt to evaluate the pitting load carrying capacity under increased thermal conditions. This is the basis for an estimated lifetime which is one of the most important parameters defining transmission reliability.

Design/methodology/approach – Recommendations related to pitting load carrying capacity calculation of case hardened gears running at high gear bulk temperatures are formulated. These factors are based on extensive experimental data, obtained in pitting tests with high oil injection temperatures, high oil sump temperatures or high operational gear bulk temperatures due to a lack of heat dissipation caused by minimised lubrication.

Findings – Testing of gear type C-PT on FZG back-to-back test rig at high gear bulk temperatures by either heating up the lubricant or caused by a lack of heat dissipation as it appears with poor lubrication conditions resulted in a decrease of up to 30 per cent of the endurance strength in various investigations. This results in a reduction of the material strength due to tempering effects and high surface shear stress due to low oil film thicknesses caused by low operating oil viscosities.

Originality/value – The present calculation method in the standard DIN/ISO is not valid for high gear bulk temperatures. Nevertheless, the present calculation algorithms of the standards DIN/ISO are valid for low and moderate thermal operating conditions when using oil temperatures of up to 80 (90)°C in the case of a sufficient cooling oil supply to the gear mesh. With the presented modifications higher gear bulk temperatures (>120°C) can be taken into account.

Keywords Load capacity, Thermal testing

Paper type Research paper

Nomenclature

a = centre distance (mm)
b = face width (mm)
d = pitch diameter (mm)
D = parameter for lubrication conditions (-)
E = relative immersion depth of gear (-)
F_t = circumferential force (N)
h_{min} = minimum oil film thickness (μm)
K_A = application factor (-)
K_v = dynamic factor (-)
$K_{H\alpha}$ = transversal load distribution factor (-)
$K_{H\beta}$ = face load factor (-)
P_{VZP} = load dependent gear power loss (W)
Ra = mean arithmetic surface roughness (μm)
u = ratio (-)
X_S = lubrication factor (-)
X_{Ca} = factor for tip relief (-)
Z_B = zone factor for inner point of single pair contact of pinion (-)
Z_E = elasticity factor (-)
Z_H = zone factor for pitch point (-)
Z_L = lubricant factor (-)
Z_{NT} = life factor for contact stress (-)
Z_R = surface roughness factor (-)
Z_V = speed factor (-)
Z_W = material factor (-)
Z_X = size factor (-)
Z_β = helix angle factor (-)
Z_ε = contact ratio factor (-)
Z_ϑ = material factor (-)
Z_λ = film thickness factor (-)
Z_μ = friction factor (-)
μ_m = mean coefficient of friction (-)
σ_H = contact stress (N/mm^2)
σ_{Hlim} = endurance limit of contact stress (N/mm^2)
σ_{HP} = allowable contact stress (N/mm^2)
σ_{H0} = nominal contact stress at pitch point (N/mm^2)

Introduction

Gears are machine components which determine the capability and reliability of many technical products. Continuous demand for higher efficiency and reliability, increased load carrying capacity and endurance life, smaller size, lower noise and vibrations, prolonged service intervals, low environmental impact and low costs will remain the main driving forces in the development of gear drives in the future. The fatigue of contacting surfaces in gears is often the life-limiting factor in transmissions which should, ideally, operate reliably for 20 years or more.

Not only nowadays a general trend is to optimise the power consumption of the whole powertrain. A higher efficiency can

The current issue and full text archive of this journal is available at **www.emeraldinsight.com/0036-8792.htm**

Industrial Lubrication and Tribology
63/1 (2011) 11–16
© Emerald Group Publishing Limited [ISSN 0036-8792]
[DOI 10.1108/00368791111101795]

Paper presented during the 3rd International Conference on Integrity, Reliability and Failure – IRF'2009, within the Symposium on Integrity, Reliability and Failure of GEARS and TRANSMISSIONS, held at Faculdade de Engenharia da Universidade do Porto, Portugal, July 20-24, 2009.

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

be obtained by reducing no-load power losses such as squeezing, splashing and ventilation losses. These losses can be reduced by lowering the oil volume, namely the oil level in dip lubricated transmissions and the oil flow rate with oil injection lubrication. In these cases, the oil amount required for lubrication may be sufficient, but oil cooling may be adversely affected. Possibly a higher efficiency, smaller size and weight of a transmission can be obtained by allowing higher operating oil temperatures and therefore abandoning oil coolers and oil pumps.

Both leads to high gear bulk temperatures resulting in thin separating films with higher friction and wear on the mating surfaces and also an increased risk of pitting.

Pitting load carrying capacity according to DIN/ISO

Pits are large-scaled break-outs of the active flank and are usually shell-shaped (Figure 1). Pitting damage is a fatigue failure mode which occurs if the material strength is exceeded locally or across the whole width of the tooth by the imposed stress. Usually, pits are the result of surface or subsurface fatigue cracks caused by local metal-to-metal contact at the roughness asperities. In addition to Hertzian stress, oil viscosity and temperature, specific sliding, surface roughness and circumferential speed have a significant effect on pitting resistance.

The calculation of the surface durability concerning pitting damage for spur gears is based on the contact stress according to DIN 3990 Teil 1 bis 5 (1987) or ISO 6336 1-6 (2006) at either the inner point of single pair contact B or the contact at the pitch point C, whichever is greater.

The nominal contact stress σ_{H0} can be calculated with equation (1):

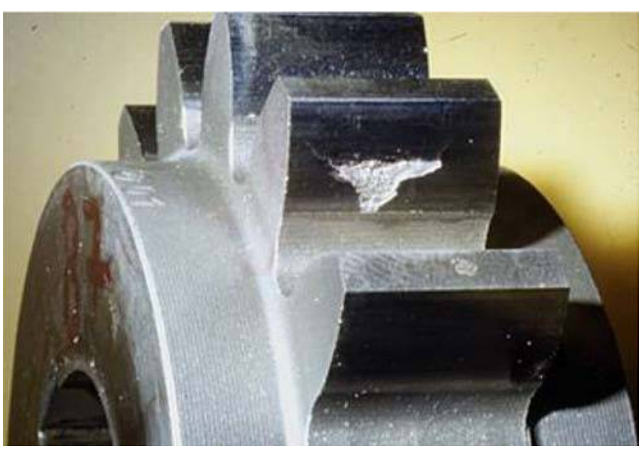
$$\sigma_{H0} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \tag{1}$$

The actual contact stress σ_H can be calculated with the following equation (2) and should be less than the allowable contact stress σ_{HP} in order to avoid a pitting damage:

$$\sigma_H = Z_B \cdot \sigma_{H0} \cdot \sqrt{K_A \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha}} \leq \sigma_{HP} \tag{2}$$

The allowable contact stress σ_{HP} can be calculated knowing the endurance limit σ_{Hlim} with the following formula (3):

Figure 1 Typical pitting damage



$$\sigma_{HP} = \frac{\sigma_{H\,lim} \cdot Z_{NT}}{S_{H\,min}} \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_X \tag{3}$$

According to DIN/ISO the lubrication conditions in the contact are taken into account by the Z_L, Z_v and Z_R.

The lubricant factor Z_L equation (4) accounts for the influence of lubricant viscosity:

$$Z_L = C_{ZL} + \frac{4 \cdot (1.0 - C_{ZL})}{(1.2 + (134/\nu_{40}))^2} \tag{4}$$

the speed factor Z_v (equation (5)) accounts for the influence of pitch line velocity:

$$Z_v = C_{Zv} + \frac{2 \cdot (1.0 - C_{Zv})}{\sqrt{0.8 + (32/v)}} \tag{5}$$

and the roughness factor Z_R (equation (6)) accounts for the influence of surface roughness on the surface endurance capacity:

$$Z_R = \left(\frac{3}{R_{z100}}\right)^{C_{ZR}} \tag{6}$$

The material factor Z_W accounts for the increase of the surface durability of a soft steel gear when meshing with a surface hardened or significantly harder gear with a smooth surface. The size factor Z_X accounts for statistics indicating that the stress levels at which the fatigue damage occurs decrease with an increase of the component size.

Pitting load carrying capacity calculation according to the standards DIN/ISO does only take into account the nominal lubricant viscosity at 40°C and not the operating viscosity, which is significantly lower especially at high gear bulk temperatures. Thus, the actual lubrication conditions in the tooth contact, especially the actual relative oil film thickness, are not sufficiently represented in the actuaöl/standardised calculation algorithms.

FZG pitting test procedure

Standard pitting tests according to Schedl (1997) are load stage tests in the area of limited life of the sn-curve which are run until a fatigue damage occurs.

Gear type C, which is used for pitting investigations, has a close-to-practical geometry with balanced sliding speed conditions.

For the pitting investigations, a standardised FZG back-to-back test rig (Figure 2) with a variable speed engine and a speed increaser for high pitch line velocities, was used (DIN 51354, Teil 1 und 2, 1990).

Pitting load carrying capacity at high gear bulk temeperatures

Knauer (1988) investigated the influence of high oil temperatures with oil injection lubrication using mineral oils of ISO VG 100 on the pitting load carrying capacity. His investigations showed a significant decrease of the endurance limit for pitting with increased oil temperatures due to the tempering effect on the case carburised material of the test gears and to low relative oil film thicknesses, possibly causing more frequent metal-to-metal contacts which result in a higher surface shear stress (Figure 3).

Figure 2 FZG back-to-back test rig according to DIN 51354

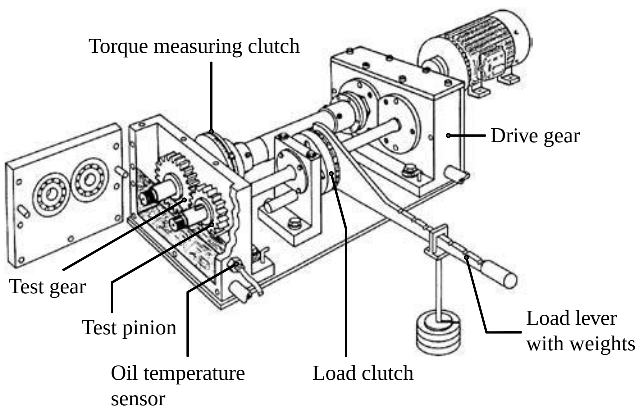
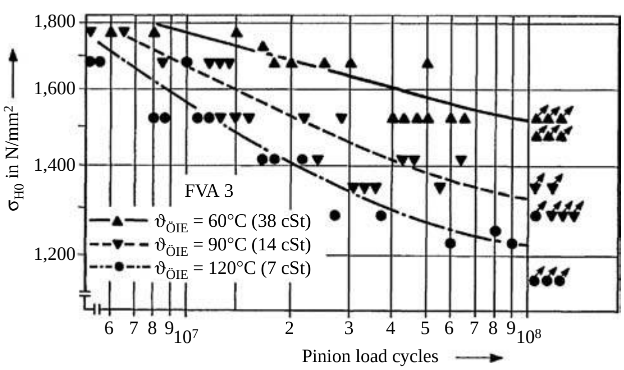


Figure 3 sn-Curves for pitting damage with mineral oil ISO VG 100 at different oil injection temperatures according to Knauer (1988)



High oil injection temperatures resulting in calculated gear bulk temperatures above 120°C cause a severe decrease of pitting life time in the area of limited life as well as strong decrease of the endurance stress in the area of endurance limit.

Knauer (1988), slightly modified later by Otto (2009), derived from these pitting investigations at high oil injection temperatures the material factor Z_ϑ, the film thickness factor Z_λ and the friction factor Z_μ which take high operational temperatures and the influence of the coefficients of friction into account.

The material factor Z_ϑ, which accounts for the decrease of the endurance strength due to a decrease of the material strength caused by tempering of the case carburised gear material at bulk temperatures above 120°C can be calculated by equation (7):

$$Z_\vartheta = 1 - 0.15 \cdot \log(\vartheta_M - 120) \tag{7}$$

The oil film thickness factor Z_λ, which accounts for the decrease of the actual relative oil film thickness in the gear contact due to a decreasing oil viscosity with high temperatures can be calculated with equation (8):

$$Z_\lambda = \lambda^{0.15} = \left(\frac{h_{min}}{Ra}\right)^{0.15} \tag{8}$$

The friction factor Z_μ, which takes into account the influence of the coefficient of friction of (e.g. with synthetic oils), can be calculated with equation (9):

$$Z_\mu = 1 + 1.2 \cdot \log\left(\frac{\mu_{ref}}{\mu_m}\right) \tag{9}$$

The parameter μ_{ref}, which represents a mean coefficient of friction for mineral oils for the testing conditions of Knauer (1988), can either be derived from efficiency experiments ($\mu_{ref} = 0.04$) or with common calculation algorithms for the mean coefficient of friction in the tooth contact.

According to Knauer (1988), the film thickness factor Z_λ and the friction factor Z_μ should replace the lubricant factor Z_L, the roughness factor Z_R and the speed factor Z_v of DIN 3990 and ISO 6336 which do not fully take into account the actual tribological conditions in the tooth contact (especially the operating viscosity) according to equation (10):

$$Z_L \cdot Z_R \cdot Z_v = 1.1 \cdot Z_\lambda \cdot Z_\mu \tag{10}$$

According to Knauer (1988), the material factor Z_ϑ is a necessary additional Z-factor for taking into account high bulk temperatures and their effect on the endurance strength of case carburised gears due to possible tempering.

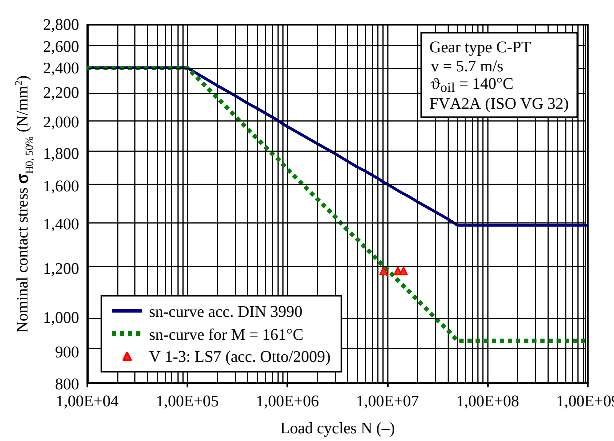
Pitting tests with mineral and synthetic oils (ISO VG 32 and ISO VG 220) performed by Otto (2009) and Otto and Maisch (2008) under rich lubrication conditions using dip lubrication with high oil sump temperatures showed equivalent test results as with high oil injection temperatures. The gears failed due to a pitting damage at loads far below the endurance limit of the standard (Figure 4). The sn-curve according to the standard has been calculated by taking into account the actual Z- and K-factors as required by the standard.

The modified sn-curve based on the calculated factors according to Knauer (1988) is additionally shown in Figure 4. The factors were calculated with the actual operating conditions. The gear bulk temperature was estimated according to formulas (11) and (12).

A very good correlation of the test results with high oil sump temperatures to the sn-curve calculated with the factors of Knauer (1988) can be found.

Additional pitting tests at a high oil sump temperature of 140°C with synthetic oils (Polyalphaolefin (PAO), Polyglykol and Ester) showed similar test results of decreasing pitting life

Figure 4 sn-Curves for pitting damage with mineral oil ISO VG 32 according DIN/ISO (continuous line) and at high oil sump temperatures (dotted line)



Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

Industrial Lubrication and Tribology
Volume 63 · Number 1 · 2011 · 11–16

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

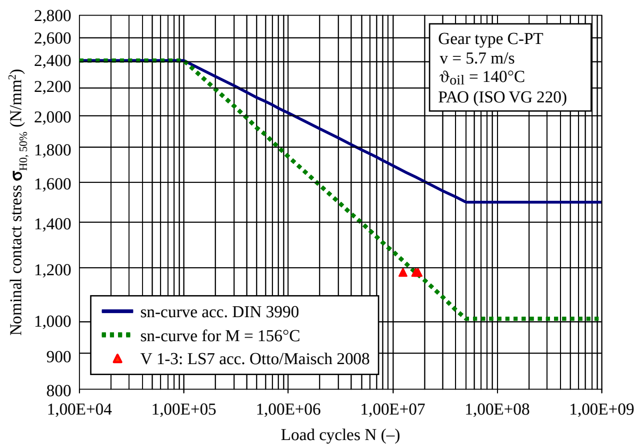
time or endurance strength compared to the present calculation algorithms of the standards (Figure 5).

The pitting life time is slightly higher with the synthetic oil compared to the test results with the mineral oil (Figure 4) done at the same loadstage. This is due to the higher viscosity on one hand resulting in a higher oil film thickness. On the other hand, the usage of this synthetic oil results in a lower mean coefficient of friction in the tooth contact and therefore in a lower gear bulk temperature. Thus, the reduction of the endurance strength according to Knauer (1988) is somewhat lower than with the mineral oil but nevertheless still severe. Again, a good correlation to the investigations of Knauer (1988) for high gear bulk temperatures can be found.

In the investigation at high oil injection temperatures as well as in the investigations with high oil sump temperatures, high gear bulk temperatures appeared due to heating of the oil. High gear bulk temperatures also occur due to a lack of heat dissipation caused by an insufficient amount of cooling oil provided in the gear contact as it happens with minimised dip (Figure 6) or injection lubrication.

Pitting tests with minimised dip and minimised oil/air lubrication at a medium pitch line velocity of 8.3 m/s at an oil temperature of 90°C performed by Otto (2009) showed a comparable trend of strongly decreasing endurance strength (Figure 7). The sn-curve according to the standard has been again calculated by taking into account the actual Z- and K-factors.

Figure 5 sn-Curves for pitting damage with synthetic oil (PAO) ISO VG 220 according DIN/ISO (continuous line) and at high oil sump temperatures (dotted line)



The pinion bulk temperatures, which were measured under similar operating conditions, are in the range from 123 to 138°C. The pinion bulk temperatures with oil/air lubrication with an oil quantity of less than 30 ml/h are in the same range as when using a very low immersion depth of only one time the module of the gear (half the height of one tooth at standstill). This indicates that no heat dissipation by cooling oil is available under those minimised lubrication conditions.

Pitting tests by Otto (2009) with minimised dip lubrication at a high pitch line velocity of 30 m/s, causing higher bulk temperatures on the one hand and an slightly improved surface separating oil film built-up on the other hand showed a comparable trend of strongly decreasing endurance strength, too (Figure 8). The sn-curve according to the standard has been again calculated by taking into account the actual Z- and K-factors.

At the higher pitch line velocity of 30 m/s, the pinion bulk temperatures are in the range from 154 to 196°C. Again, the sn-curve for the lowest immersion depth of only one time the module of the gear is valid for lubrication conditions without heat dissipation from the gear mesh by cooling oil.

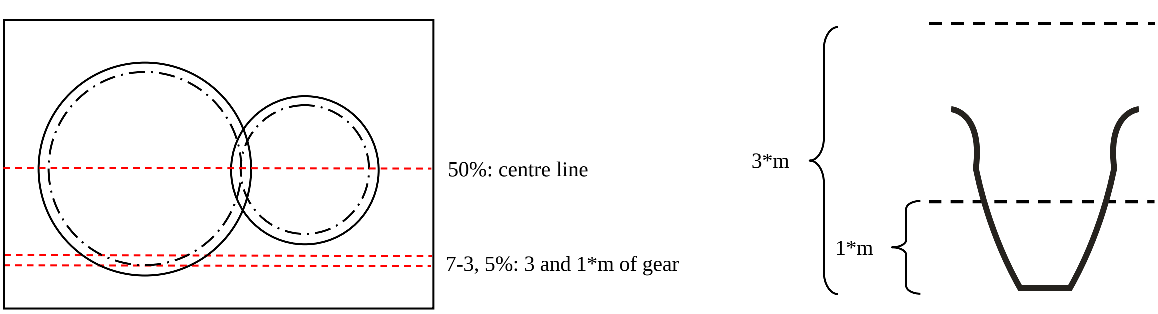
In Figure 7 as well as in Figure 8, the grey scattered areas indicate the calculated endurance strength according to Knauer (1988) for high gear bulk temperatures. The gear bulk temperature was estimated according to formulas (11) and (12). Again a very good accordance to the test results could be found.

The pitting test results at high oil temperatures as well as those under minimised lubrication conditions, which all resulted in high gear bulk temperatures, are not predictable with the present standards DIN/ISO, which are both obviously only valid for low thermal operating conditions.

Evaluating the test results at rich lubrication conditions using high oil temperatures (dip and oil injection lubrication) as well as those with minimised lubrication with the factors Z_λ, Z_μ and Z_ϑ presented first by Knauer (1988) showed a very good accordance and can therefore be used for operating conditions which result in higher gear bulk temperatures than 100°C.

The present calculation algorithms of the standards DIN/ISO are valid for low to moderate thermal conditions using oil temperatures of up to 60 to 80°C. All conducted investigations at operational conditions which lead to high gear bulk temperatures (above 120°C), by either heating of the oil or caused by a lack of cooling oil, showed a strong decrease of the endurance strength concerning pitting damage.

Figure 6 Immersion depth with dip lubrication



Downloaded by Technical University of Munich Library At 05:05 21 September 2016 (PT)

Figure 7 sn-Curves for pitting damage with mineral oil ISO VG 32 under minimised lubrication conditions at 8.3 m/s according to Otto (2009)

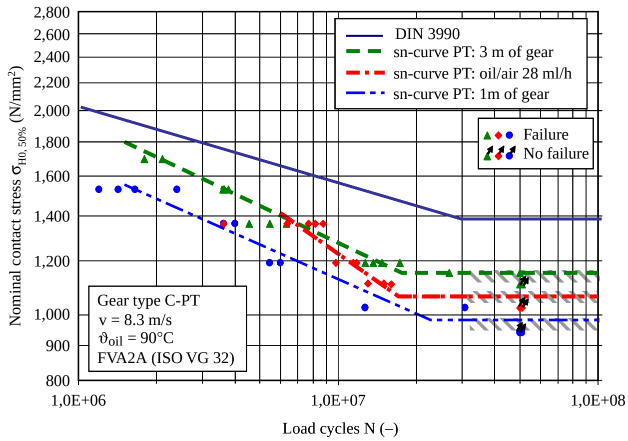
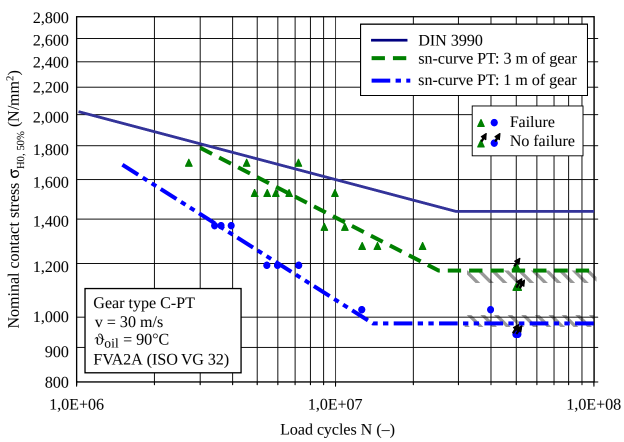


Figure 8 sn-Curves for pitting damage with mineral oil ISO VG 32 under minimised lubrication conditions at 30 m/s according to Otto (2009)



Gear bulk temperature calculation

The material factor Z_ϑ as well as the factor for the relative oil film thickness Z_λ are strongly determined by the actual gear bulk temperatures. They can be calculated by either inserting measured values or by estimating the actual bulk temperature ϑ_M with equation (11) according to Oster (1982):

$$\vartheta_M = \vartheta_{oil} + 7{,}400 \cdot \left(\frac{P_{VZP}}{a \cdot b}\right)^{0.72} \cdot \frac{X_S}{1.2 \cdot X_{Ca}} \qquad (11)$$

The lubrication factor X_S according to Otto (2009) can be estimated with equation (12), which depends on the relative immersion depth E of the gear, in case of minimised lubrication conditions:

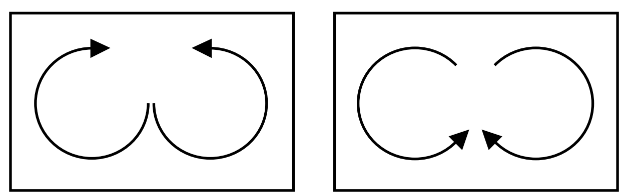
$$0.3 \leq X_S = 0.35 \cdot E^{-D} \leq 3.7 \qquad (12)$$

With $D = 0.75$ for poor lubricant supply to the gear mesh and $D = 0.5$ for direct transportation of cooling oil to the gear mesh (Figure 9).

A constant $X_S = 3.7$ can be used for starved lubrication conditions without any heat dissipation by cooling oil but only by convection and conduction to the surrounding metal components.

For rich oil injection lubrication a constant $X_S = 1.2$ can be used.

Figure 9 Lubricant supply conditions in the gear mesh depending on rotational direction of the gears with dip lubrication



"Poor" lubricant supply   "Rich" lubricant supply

Conclusions

Testing of gear type C-PT on FZG back-to-back test rig at high gear bulk temperatures by either heating up the lubricant or caused by a lack of heat dissipation as it appears with poor lubrication conditions resulted in significantly reduced endurance strength. This results in a reduction of the material strength due to tempering effects and high surface shear stress due to low oil film thicknesses caused by low operating oil viscosities. The present calculation method in the standard DIN/ISO is not valid for high gear bulk temperatures. Nevertheless, the present calculation algorithms of the standards DIN/ISO are valid for low and moderate thermal operating conditions when using oil temperatures of up to 80 (90)°C in the case of a sufficient cooling oil supply to the gear mesh. The presented modifications are only necessary to be taken into account at high gear bulk temperatures (>120°C). A decrease of up to 30 per cent of the endurance strength was found in various investigations at high thermal conditions.

References

DIN 51354, Teil 1 und 2 (1990), *FZG Zahnrad-Verspannungsprüfmaschine*, Beuth, Berlin.
DIN 3990 Teil 1 bis 5 (1987), *Tragfähigkeitsberechnung von Stirnrädern*, Beuth, Berlin.
ISO 6336 1-6 (2006), *Calculation of Load Capacity of Spur and Helical Gears*, ISO, Geneva.
Knauer, G. (1988), "Zur Grübchentragfähigkeit einsatzgehärteter Zahnräder – Einfluß von Werkstoff, Schmierstoff und Betriebstemperatur", dissertation, TU München, Munich.
Oster, P. (1982), "Beanspruchung der Zahnflanken unter Bedingungen der Elastohydrodynamik", dissertation, TU München, Munich.
Otto, H.-P. (2009), "Flank load carrying capacity and power loss reduction by minimised lubrication", dissertation, TU München, Munich.
Otto, H.-P. and Maisch, C. (2008), "Ermittlung einer Öl-Lebensdauerformel für Industriegetriebe", Forschungsvereinigung Antriebstechnik eV, Heft 861, FVA Frankfurt, Frankfurt.
Schedl, U. (1997), *Pitting Test*, FVA Information Sheet 2/IV, Forschungsvereinigung Antriebstechnik Frankfurt, Frankfurt.

Further reading

Niemann, G. and Winter, H. (1983), "Maschinenelemente, Band II: Getriebe allgemein", *Zahnradgetriebe – Grundlagen, Stirnradgetriebe*, 2 Auflage, Springer, Berlin.

Corresponding author

Hans-Philipp Otto can be contacted at: p.otto@fzg.mw.tum.de

To purchase reprints of this article please e-mail: **reprints@emeraldinsight.com**
Or visit our web site for further details: **www.emeraldinsight.com/reprints**

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

Downloaded by Technical University of Munich University Library At 05:05 21 September 2016 (PT)

This article has been cited by:

1. Jing Zhang, Qi Zhang, Changhong Wu, Zhezhu Xu, SungKi Lyu. 2014. Experimental application of pitting formation for 20MnCr5 carburized gear tooth. *International Journal of Precision Engineering and Manufacturing* **15**:5, 899-903. [CrossRef]