

Effects of Temperature Fields on Gear Contact Fatigue Performance

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Abstract: Temperature fields can alter gear contact fatigue through changes in elastic properties and redistribution of tooth loading. This study examines these competing effects using a parameterized steel spur-gear model with prescribed uniform temperatures and linear gradients across the face width. A constrained thermal-displacement model is coupled to unilateral strip contact, local Hertzian pressure, and an assumed relative fatigue-life relationship. All numerical results are model estimates rather than experimental measurements. Uniform heating from 40 to 80 °C reduces peak contact pressure by approximately 0.8% and increases the pressure-only life indicator to 1.050 times its reference value. At the same mean temperature of 80 °C, end-to-end temperature differences of 10, 20, and 40 K increase peak pressure by approximately 19.3%, 36.5%, and 61.8%, respectively, relative to the reference condition. Corresponding pressure-only life indicators decrease to 0.346, 0.155, and 0.056. An assumed temperature-dependent fatigue-strength reduction further decreases these indicators. Constrained differential expansion therefore dominates the modest pressure reduction associated with elastic softening in the selected scenarios. The magnitude depends on support compliance, displacement transfer, and fatigue calibration; absolute service life cannot be inferred.

Keywords: Gear contact fatigue, temperature field, thermal deformation, Hertzian contact, load distribution, relative fatigue life.

1. Introduction

Gears with the same transmission torque will not have the same local contact pressure if their temperature distributions are different. Uniform heating will change the elastic modulus, but the face-width load will remain almost the same. A Spatial Temperature Difference will cause the mating surface to move slightly and concentrate the load at one place. Therefore, contact fatigue is determined by how temperature changes the contact geometry and material properties, not by the average temperature.

Free expansion, constrained differential expansion and temperature-dependent properties should be dealt with separately, as they will change in size, redistribute load, and thus affect pressure and fatigue. Only one temperature correction is unlikely to be appropriate. That is, the contact force will not increase due to thermal expansion if the transmitted torque is constant.

The above indicate the two. Li and others have combined mixed elastohydrodynamic lubrication, rough-surface contact and depth-dependent properties in the prediction of gear contact fatigue and found that the nominal smooth-surface pressure is not suitable for describing initiation [1]. Thermal lubrication analysis of rough spiral bevel gears and low-sliding-ratio face gears has shown that local interface conditions need to be taken into account [4, 8], and it has distinguished between prescribed bulk temperatures and temperatures generated at the lubricated contact.

Heat will also spread in the surrounding area. Mo and others have studied the thermo-mechanical coupling of a

gear-rotor-bearing system with multiple clearances; that is to say, thermal deformation changes stiffness and load distribution [5], so the same tooth temperature field can result in different contact patterns under different bearing arrangements. According to the evidence in gradient-structured gear steel, the reduction of Hertzian stress caused by a temperature change is compensated for by alterations in fatigue properties [7], so a lower pressure does not necessarily result in higher fatigue strength.

Now I want to investigate a particular problem: Given an explicit displacement constraint, how much do mean temperature and a face-width gradient affect the peak pressure and normalized fatigue life? The first analysis model ignores heat generation and distributes the load according to force equilibrium. Matched-mean conditions are not caused by even heating. It is a reproducible sensitivity study that has not been experimentally verified, nor is there material-specific fatigue calibration or full thermal lubrication prediction.

2. Model and Methods

2.1. Gear Configuration and Prescribed Temperature Fields

The identical steel spur gear pair is shown at the pitch circle. One equivalent tooth pair bears the whole of the normal load, and tooth pair load sharing, frictional traction and dynamic amplification are neglected. Table 1 is the input. All constitutive values are analyst-selected assumptions, and the representative steel properties are also such; none are measurements or coefficients taken from other studies.

Table 1. Assumed geometry, loading, material properties, and sensitivity parameters.

Quantity	Symbol	Value	Status and role
Teeth on each gear	z	40	Assumed geometry
Module	m_g	3 mm	Assumed geometry
Pressure angle	ϕ	20°	Assumed geometry
Pitch radius	r	60 mm	Derived from geometry
Face width	b	30 mm	Assumed geometry
Torque	M	563.8 N m	Selected nominal loading
Normal load	F_n	10,000 N	Rounded calculation load
Equivalent pitch-point radius	R'	10.26 mm	Derived curvature
Reference temperature	T_{ref}	40 °C	Assumed baseline
Reference Young's modulus	E_0	210 GPa	Representative assumption
Poisson's ratio	ν	0.30	Assumed constant
Thermal expansion coefficient	α	$12 \times 10^{-6} \text{ K}^{-1}$	Assumed constant
Elastic temperature coefficient	a_E	$4 \times 10^{-4} \text{ K}^{-1}$	Assumed sensitivity
Effective expansion length	L	50 mm	Assumed support parameter
Displacement transfer factor	η	0.50	Assumed constraint
Distributed normal stiffness	k	10^{11} N m^{-2}	Assumed structural compliance
Fatigue exponent	m	6; range 4–8	Uncalibrated sensitivity
Strength temperature coefficient	β	0–0.001 K^{-1}	Assumed sensitivity

The nominal load follows $F_n = M / (r \cos \phi)$. Individual pitch-point curvature is $r \sin \phi$, giving $R' = r \sin \phi / 2$. Curvature remains fixed to isolate compliance and modulus effects.

With face coordinate y measured from the center, the imposed field is

$$T(y) = \bar{T} + \Delta T \frac{y}{b}, -\frac{b}{2} \leq y \leq \frac{b}{2} \quad (1)$$

Where $\bar{T}$ is mean temperature and ΔT is the end-to-end difference. Figure 1 defines six cases using cool-edge, center, and hot-edge temperatures. Both flanks receive the same local temperature for the elastic calculation. The field also applies throughout one effective expansion path representing net relative support-to-flank displacement; its effect is not counted twice.

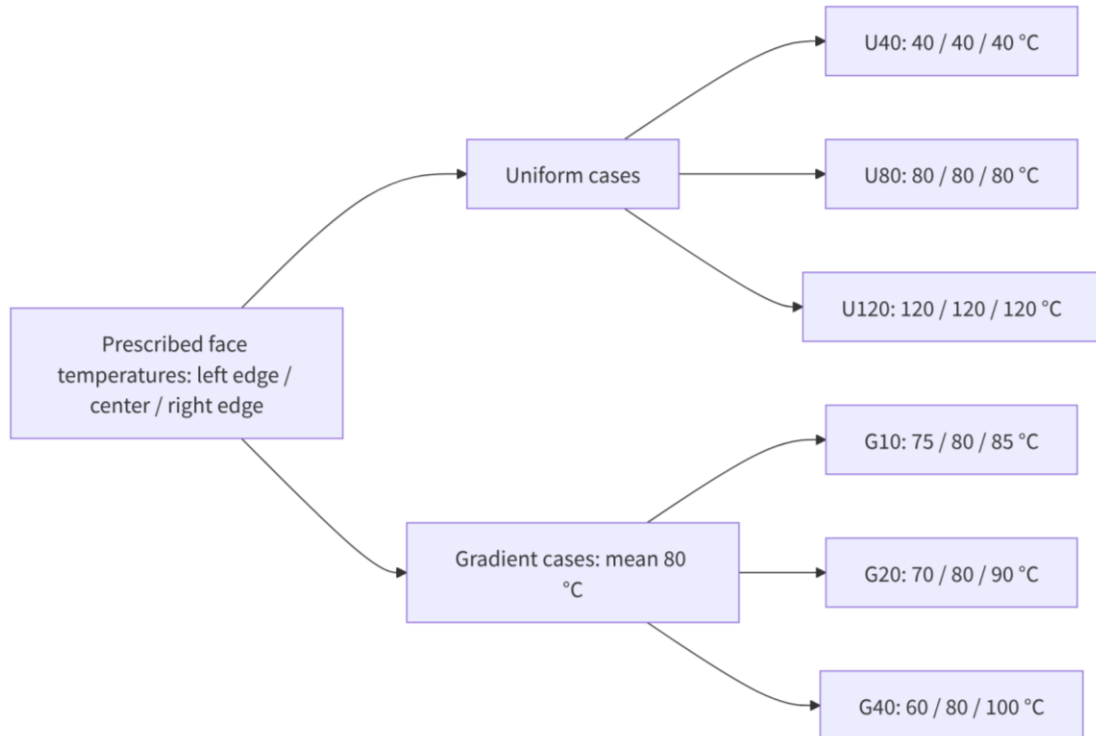


Figure 1. Prescribed thermal scenarios. Edge coordinates are $y = \pm 15$ mm. Temperatures at the listed places are linear. U80 is the matched-mean comparison for all gradient cases.

These are external heat conditions. As there is no heat flux, convection or the lubricant energy equation, the model cannot find the operating temperature.

2.2. Thermal Mismatch, Equilibrium, And Local Pressure

Thermal strain is $\epsilon_{th} = \alpha(T - T_{ref})$. Support can be added for normal translation, but not for relative tilting to separate the face-width mismatch. Subtracting the common part of

transferred displacement yields

$$g(y) = -\eta \alpha L [T(y) - \bar{T}] \quad (2)$$

Positive temperature deviation reduces the gap at the constraint; the sign depends on the assembly. The solution of the general problem is uniform expansion.

For strips of width Δy , unilateral compliance and equilibrium require

$$F_i = k \Delta y \max[\delta - g(y_i), 0], \sum_i F_i = F_n \quad (3)$$

Approach δ , gap g and strip coordinates are in metres, and strip force F_i is in Newtons. Nonnegative is a division condition. Thermal displacement moves the fixed load; it does not introduce a pre-determined thermal force.

Temperature-dependent Modulus and local Hertzian pressure are

$$E(T) = E_0 [1 - a_E (T - T_{ref})], E_i' = \frac{E(T_i)}{2(1 - \nu^2)} \quad (4)$$

$$w_i = \frac{F_i}{\Delta y}, p_{0,i} = \sqrt{\frac{w_i E_i'}{\pi R}} \quad (5)$$

Here, E_i' is the reduced modulus, w_i is the load intensity in newtons per metre, and $p_{0,i}$ is the peak strip pressure in pascals. The stiffness of the structure is constant and does not vary with the local Hertzian deformation; it is not a fully coupled thermoelastic compliance solution.

The continuous estimation result is the analytical solution. Define $w_{ref} = F_n / b$, $\delta_0 = w_{ref} / k = 3.333 \mu\text{m}$, and

$$A = \frac{\eta \alpha L \Delta T}{2 \delta_0} = (0.045 \text{ K}^{-1}) \Delta T \quad (6)$$

For $A \leq 1$, contact spans the width and

$$\frac{w(y)}{w_{ref}} = 1 + 2A \frac{y}{b} \quad (7)$$

For $A > 1$, loading is triangular over fraction $f = A^{-1/2}$, with $w_{max} / w_{ref} = 2\sqrt{A}$. Integration returns F_n , and the zero gradient normalises uniformly. Midpoint strip sums approximate the continuous expression by adding a small strip width. Analysis consistency checking is not physical verification or the conclusion of a numerical mesh study.

2.3. Relative Fatigue-Life Indicator

The assumed local relationship is

$$\frac{N_i}{N_{ref}} = \left[\frac{S_H(T_i)}{S_H(T_{ref})} \right]^m \left(\frac{p_{ref}}{p_{0,i}} \right)^m, \frac{S_H(T)}{S_H(T_{ref})} = 1 - \beta(T - T_{ref}) \quad (8)$$

The fatigue-strength scale S_H has pressure units; N_i and N_{ref} denote cycle counts, and p_{ref} is U40 pressure. Only dimensionless life ratios are computed. The smallest loaded-strip ratio is the starting value before adding the stressed-volume and cumulative-damage effects. Unloaded strips are not shown in this figure.

Research on polymer gears has focused on the temperature-dependent properties of materials, but the obtained coefficients cannot be applied to steel [2, 6]. Therefore, the strength scale, exponent and thermal coefficients need to be calibrated independently for actual steel, heat treatment and surface conditions. Calculations distinguish between unchanged strength and assumed thermal-strength sensitivity.

3. Results and Discussion

3.1. Uniform Heating Versus Spatial Gradients

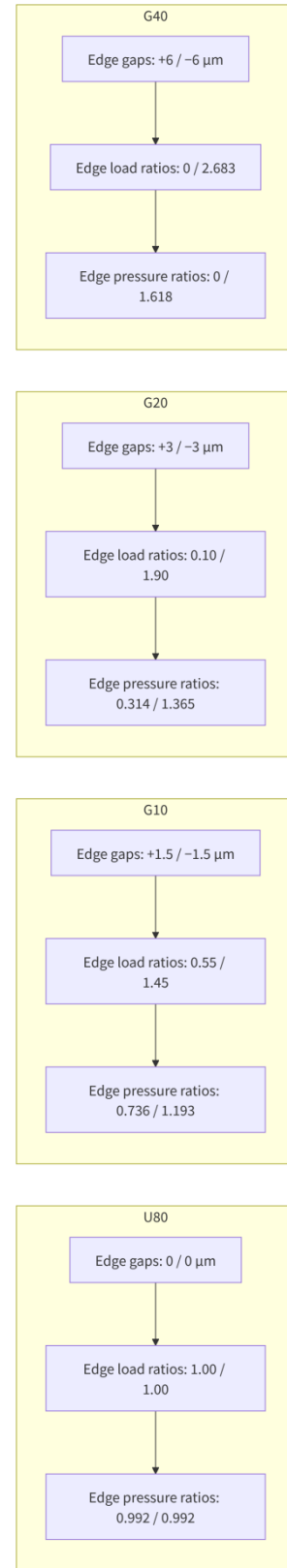


Figure 2. Calculated thermal-gap, load, and pressure comparisons.

Each pair lists cool then hot edges. Reference scales are $w_{ref} = 333.3 \text{ kN m}^{-1}$ and $p_{ref} = 1.092 \text{ GPa}$. Gap and engaged-region loading are linear; pressure follows the Hertzian equation. G40 is unloaded below $y = -7.36 \text{ mm}$.

The reference reduced modulus is approximately 115.4 GPa, producing 1.092 GPa contact pressure. Heat uniformly to 80°C, and then reduce the modulus and pressure by about

1.6% and 0.8% respectively, according to the square-root Hertzian law. At 120 °C, pressure decreases approximately 1.6%. Load is uniform; there will be no uneven expansion.

A Gradient Alters Load Distribution. G10 is a 3 μm gap different from the 3.333 μm reference method. Normalised Edge Loads are 0.55 and 1.45. G20 has the values 0.10 and 1.90, and the total force is unchanged. Figure 2 shows thermal mismatch, loading and pressure using aligned case sequences;

Table 2. Continuum model estimates of peak pressure, engaged width, and minimum normalized life. Both life columns use $m=6$; strength reduction uses $\beta=0.001 \text{ K}^{-1}$.

Case	Mean temperature (°C)	Temperature difference (K)	Peak pressure (GPa)	Pressure change from U40 (%)	Loaded fraction	Pressure-only life ratio	Life ratio with strength reduction
U40	40	0	1.092	0.0	1.000	1.000	1.000
U80	80	0	1.083	-0.8	1.000	1.050	0.822
U120	120	0	1.074	-1.6	1.000	1.103	0.669
G10	80	10	1.303	+19.3	1.000	0.346	0.263
G20	80	20	1.490	+36.5	1.000	0.155	0.114
G40	80	40	1.767	+61.8	0.745	0.056	0.038

The G40 peak pressure is about 63.1 per cent higher than U80. The matched-mean comparison is the gradient term. Hot-edge elastic softening will reduce the rise slightly, but it cannot solve the concentration at the given constraint.

3.2. Competing Effects on Fatigue Response

Reduce the modulus to decrease the pressure; only the pressure indicator will rise uniformly with temperature. Given the same fatigue strength, it cannot be concluded that hotter steel gears have a longer service life. The assumed strength slope is negative; thus, U80 and U120 are reduced to 0.822 and 0.669, respectively.

The hot edge is subjected to a high stress in the gradient

Table 2 lists the total fatigue indices.

Partial contact starts at above 22.2 K for the chosen parameters when $A>1$. G40 is above this limit; its loaded fraction is 0.745 and the cool-side region is unloaded. The top-load intensity of the equilibrium condition is 2.683 times the reference value. Therefore, it is a concentration effect of pressure and not an increase in torque.

case, and because of a decrease in strength, it has a low fatigue-strength index. G20: A 5% reduction in the local strength index is assumed, so it will be 0.114. Raising the strength ratio to the sixth power increases the small constitutive change. Pressure is therefore not the only cause.

Chang and others have presented a qualitative comparison; they have found that stress reduction can offset the effect of thermal change on gear-steel fatigue performance and supported the separation of these two paths [7], but the current calculation has not reproduced their gradient microstructure or experiments. Figure 3 is the change in the assumed exponent of predicted magnitude.

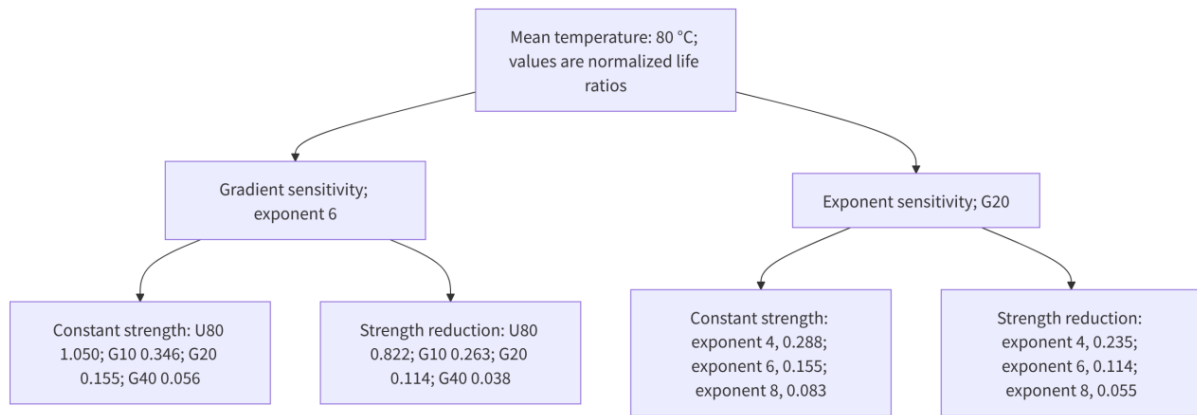


Figure 3. Calculated gradient and exponent sensitivities. Constant strength uses $\beta=0$; strength reduction uses $\beta=0.001 \text{ K}^{-1}$. The exponent comparison holds $\Delta T=20 \text{ K}$.

Gradient ranking is performed on all positive exponents, the hot-edge pressure increases from U80 to G40, and the strength factor remains the same or decreases. Therefore, the ratio to the power will decline gradually. It is a small amount, and raising the G20 exponent from four to eight reduces the pressure-only index from 0.288 to 0.083. Therefore, it is relatively easy to provide directional comparisons than numerical service-life multipliers.

The other is Support Sensitivity. Since A is proportional to ηk , halving the displacement transfer or stiffness will reduce G40 from 1.8 to 0.9 and restore full-width contact without increasing the temperature. Relative Rotation can further reduce the linear mismatch, but rotation is not included here. A Gradient Alone is Not Enough for Fatigue Performance.

3.3. Engineering Implications and Model Boundaries

Temperature uniformity is to distribute the load in this model. Mean-temperature control refers to elasticity and fatigue. Therefore, both of the above indices will provide a more complete picture than just one. To reduce the transfer of differential expansion at the flank mismatch, one may reduce the gradient to the same extent; however, the chosen constraint should represent the assembly.

Small at the edges. The strip model ignores lateral elastic coupling, crowning, end relief and three-dimensional free-edge stresses. It is the upper bound for local Hertzian theory and not an actual edge stress. There is a problem of partial

contact, and a three-dimensional solution can increase the maximum value and alter the size.

Lubricant viscosity, film thickness, asperity contact, flash temperature, wear and residual stress are also excluded. Li and others have carried out coupled thermal lubrication-wear-fatigue studies to add more physics to the general prediction [3], but this cannot be verified by experiments. Some new knowledge has also been gained about the surface and sub-surface of mixed-lubrication fatigue research.

Lubricant loss will change the lubrication conditions and damage modes. Braumann and others have carried out a serious operation in this regard, but raising the upper limit of the set temperature is not considered starvation or lubricant loss [9]; the latter requires the addition of an explicit lubricant and transient thermal analysis.

Validation will determine the face-width temperature, thermally induced flank displacement, contact pattern at a fixed torque, and fatigue results for the same material and lubricant. Tests of these links will find that there are offsetting errors not covered in the final agreement. The above measurements will also give us the corresponding support-transfer and fatigue values.

4. Conclusion

According to the parameter model, the mean temperature and the spatial distribution of temperature both affect gear contact fatigue. Uniform heating at 40-80 °C lowers the calculated pressure by about 0.8%, and therefore the pressure-only life index is 1.050. Considering that the fatigue strength at different temperatures is lower, the index drops to 0.822.

At a matched mean temperature of 80 °C, the peak pressure at end-to-end differences of 10 K, 20 K and 40 K is about 19.3%, 36.5 and 61.8% higher than that of U40, respectively. The pressure-only life indicators are 0.346, 0.155 and 0.056. The first drop in engaged width is about 0.745. The above figures are the result of redistributing a fixed normal load under the given support conditions and do not represent an increase in the transmitted load.

The most general conclusion is that thermal fatigue evaluation should consider both changes in temperature and the resulting contact mismatch under mechanical load. Gradient ranking is the same for all selected fatigue sensitivities, but the numerical life ratio is different after

calibration. The first two are the assumed displacement transfer and uncalibrated fatigue parameters. Lubrication, three-dimensional edge contact and material-specific fatigue behaviour should be taken into account before applying the relative indicator to predict the absolute service life.

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