Influence of Thermal Distortion on Spur Gear Tooth Contact

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Introduction

The increase of efficiency in automotive gear transmissions has been a key subject of research in the past decade (Ref. 1); to this aim, dip-lubricated gearboxes have been designed with minimized oil levels in search of reduced churning power losses (Ref. 2) and maximum power-to-weight ratios. However, when such gears' running conditions reach the highest pitch line velocities and torques, bulk temperatures increase rapidly, resulting in reduced oil film thickness, higher surface shear stresses and an increased number of asperity contacts — all of which may lead to a premature failure of the gear pair. Moreover, in such cases, not only high temperatures are reached but also significant temperature gradients are found between pinion and gear. Figure 1 shows a sample of the aforementioned operating case. As can be seen, the optimal conditions lie between a relative immersion depth of $H/D = 0.1$ to 0.25, where power losses are minimal. However, with these oil levels the temperature difference between pinion and gear reach 20°C to 30°C because the small size of the pinion results in a lower convection heat transfer due to the reduced oil-lubricated area and its increased rotational speed (with respect to that of the gear) increases the amount of heat flowing inwards, resulting in a higher bulk temperature of the pinion. This phenomenon leads to relative pitch deviations and thermally induced profile distortions (Ref. 3).

Several experimental results regarding the effects of thermal distortion on the gear mesh can be found in literature — primarily related to turbo-gears running at high pitch line velocities (close to 100 m/s) (Refs. 4–6) — where the differential thermal effects are known to cause large variations of contact and root stresses (Ref. 7). At lower tangential speeds, only thermal effects of plastic gears have been analyzed in scientific literature (Refs. 8–9) because the material properties are temperature-dependent and the increased thermal expansion coefficient causes significant profile distortion. Nevertheless, plastic gears do not reach the bulk temperature levels of their steel counterparts (Ref. 10) and, therefore, metallic gears may also show appreciable geometry distortion when they are subject to operating conditions such as those shown (Fig. 1).

In the present work, predicted steady-state temperature distributions from a thermal-lumped parameter model developed by the authors in (Ref. 11) are used as inputs for a 2-D finite element model. Geometry distortion and tooth contact parameters in the transverse plane are numerically computed where load distribution, transmission error and available backlash are studied. Narrow face width spur gears with b/d<0.5, running at high speeds and low oil levels are analyzed, and it will be shown that even with thermally favorable operating conditions with high immersion depths, profile deviations are appreciable. The results of the study will allow setting the limits of design backlash to avoid gear jamming and to size the initial profile modifications to reach the desired contact behavior at operating temperatures.

Thermo-Mechanical Model

The thermo-mechanical model developed in the current work is a combination of a simple thermal-lumped parameter model of the gear pair (Ref. 11) and a 2-D finite element model (FEM) (Ref. 12). The former allows calculating thermal gradients in the gear body as well as the temperature differences between pinion and gear; while the latter is used to compute geometry distortion and perform the loaded tooth contact analysis (LTCA).

Thermal-lumped parameter model. Unlike other thermal network models in gear literature that represent the complete

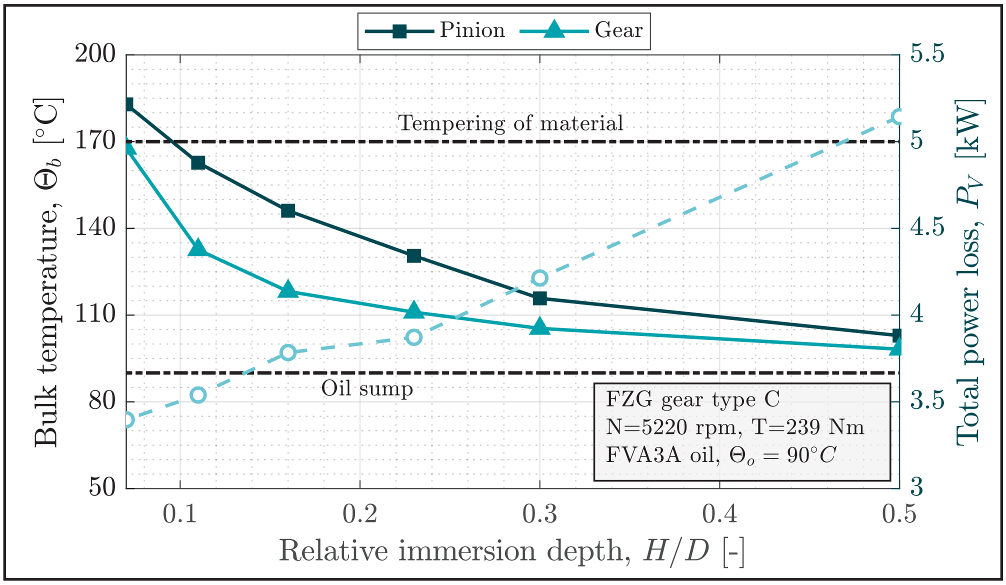

Figure 1 Influence of oil level in gear pair bulk temperature and thermal gradient (Data from Ref. 2).

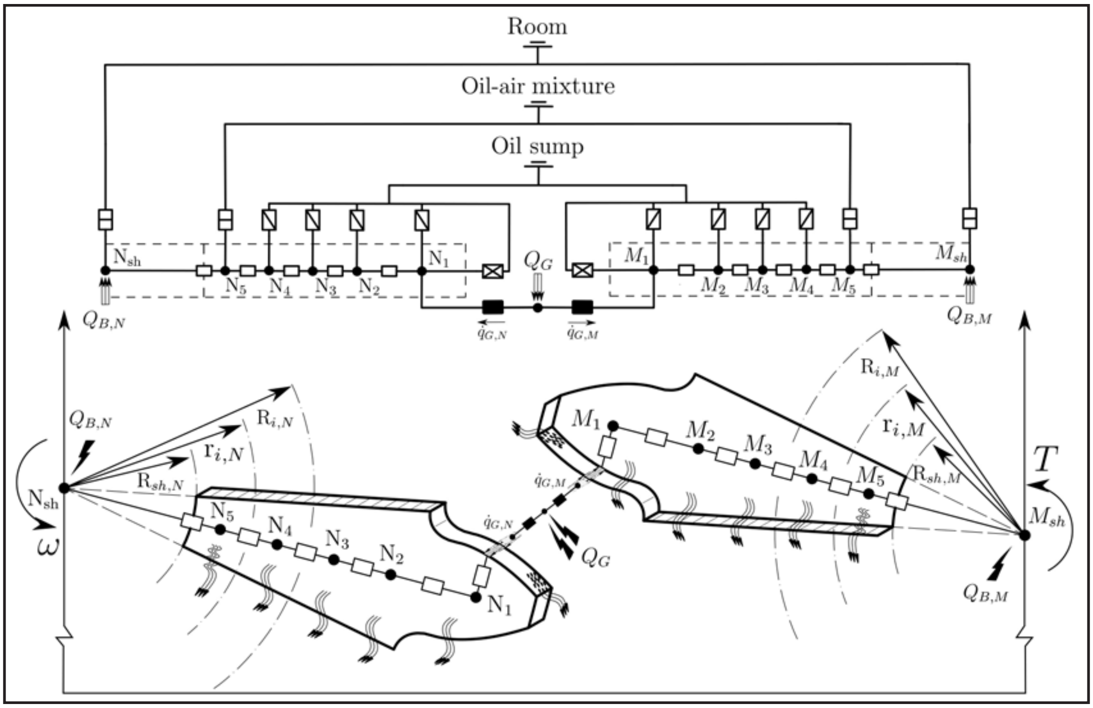

Figure 2 Thermal network model of a gear pair.

gearbox, the thermal network model developed herein focuses on the temperature field of the gear pair. The main underlying assumption is that it is possible to compute the steady-state temperature distribution within the solids, considering only the heat flux coming from the gear mesh and the shaft, as well as the steady-state temperatures of the surrounding fluids (oil and air inside the gearbox).

As it can be seen (Fig. 2), pinion and gear are discretized radially and each individual control volume is linked to its neighbors by means of thermal resistances. Circumferential thermal gradients within the gear body are neglected and therefore pure radial conduction is considered. This means that the control volumes are represented as annulus, where conduction resistances are computed from the classical one-dimensional cylindrical approach. The relative immersion depth being different for the pinion and gear, some of individual annulus are permanently in contact with the oil sump, while others only exchange heat with the oil-air mixture inside the gearbox. Therefore, convective heat transfer resistance between the external surfaces of the gear and the surrounding fluid depend on the local Nusselt number, *Nu*, which is evaluated considering the thermo-physical properties of the fluid at the boundary layer temperature. Heat exchange due to radiation is neglected because the temperature levels of the elements inside the gearbox are not high (usually below 200°C).

Two heat sources are considered: i) power losses from the gear mesh; and ii) heat from the bearings and sealings. The former is described in detail (Ref. 11), while the latter is computed from classical equations in gear literature (Ref. 13) where only a third of the total power loss is introduced in the gear axis nodes; the remaining two thirds are assumed to be dissipated through the housing and oil bath. If constant, steady-state oil/air temperatures are considered, it is not necessary to compute churning and windage power losses because the heat generated within the fluid no longer increases its bulk temperature.

Finally, the solution of the model is given by the system of equations assembled in matrix form in Equation 1, in which C is the capacity matrix; K is the conductivity matrix (where $K_{i,j} = R_{i,j}^{-1}$); Q is the heat input vector; Θ is the temperature vector; t denotes time; and the superimposed dot indicates time differentiation. In order to avoid instabilities due to non-linearities in fluid properties and convection coefficients, the Picard iteration method is used to solve the initial value problem.

$$C(\Theta,t) \times \dot{\Theta} + \mathrm{K}(\Theta,t) \times \Theta = Q(t) \quad (1)$$

The resulting radial temperature distribution of pinion and gear is loaded in a 2-D finite element model which computes the thermal growth of each point of the mesh following Equations (2) and (3).

$$r_y'(x) = r_y(x) + \Delta r_y(\Theta) \quad (2)$$

$$\Delta r_y(\nabla\Theta) = \alpha_L \cdot \sum_{i=1}^{i=n} (r_{i+1} - r_i) \cdot \left(\frac{T_i + T_{i+1}}{2}\right) - T_0 \quad (3)$$

Finite element model. A thermo-mechanical quasi-static and two-dimensional model of a gear pair is developed to analyze the effect of the thermal distortion on gear mesh behavior — namely, load distribution and transmission error. The numerical models are automatically generated using a custom gear mesh generator and the FEM models are calculated using *MSC.Marc* solver.

The gear and pinion mesh are generated following the methodology used by Litvin (Ref. 14). This methodology is characterized by defining a greater density of elements in the teeth than in the body of the gear (Fig. 3). Additionally, a progressive mesh density (i.e. — bias factor) has been determined from the symmetry axis of the tooth to the contact area (Ref. 15). A four-node, isoparametric, arbitrary quadrilateral element for plane stress applications is used (element type 3 of the *MSC.Marc* software (Ref. 16)).

The numerical model considers two load cases: in the first

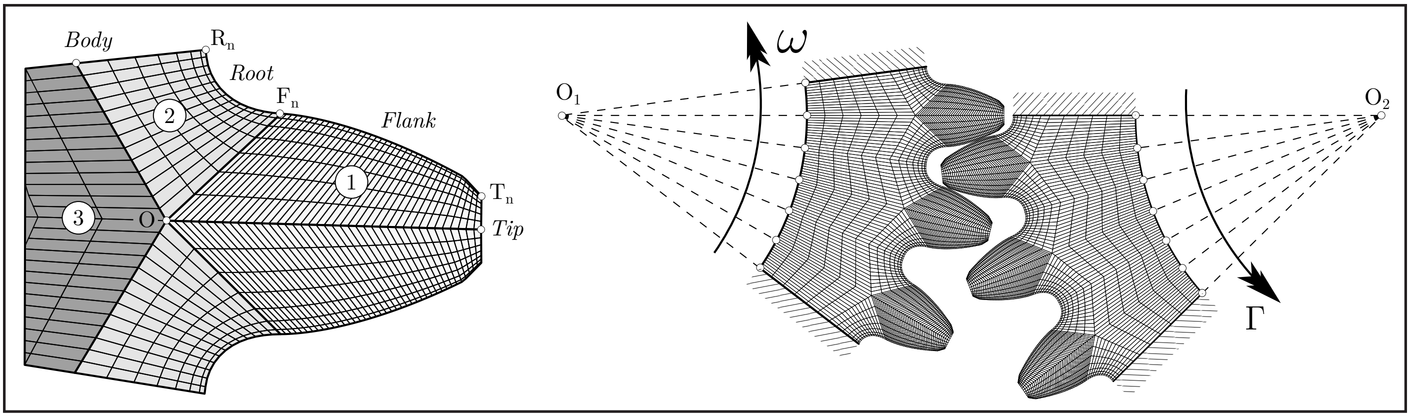

Figure 3 Mesh detail and boundary conditions of the numerical model.

step, the temperature of the gears as computed by the lumped thermal network model are introduced and then thermal growth is computed. In a second step, the expanded gears rotate and the contact between pinion and gear is computed. The rotation is applied in the pinion and the torque in the gear, both counter-clockwise. Rigid connectors are introduced between the point representing the gear axis centerline and the body, which are also used to apply the rotation and torque boundary conditions (Fig. 3). As it has been described by the authors in a previous work (Ref. 12), the complete gear was modelled in order to minimize the effect of boundary conditions in the results.

Case study. With the aim of studying the influence of thermally induced geometry distortions on contact behavior, a reference gear pair with identical teeth geometry has been selected. Such gears are characterized by having the same length of approach and recess, as well as the same thickness along the tooth height (i.e. — same manufacturing profile shift coefficients 'X_E', ensuring that they have the same bending and rotational stiffness at any point). With this approach, any difference in contact behavior can be easily identified from the horizontal and vertical displacements in the corresponding diagrams.

Table 1 summarizes the geometrical characteristics of the gear set under consideration. Five different torque levels are applied — from 100 Nm to 500 Nm at two different pitch line velocities — 10 m/s (2,000 rpm) and 20 m/s (4,000 rpm). The gears are lubricated with FVA3A mineral oil which is an ISO VG100 oil with 4% Anglamol EP additive. The numerical simulations consider oil levels from H/D = 0.5 to 0.1 at a constant oil sump temperature of 90°C. The material selected for the gears is a 16MnCr5 steel, 206 GPa Young modulus, 0.3 Poisson coefficient, 7,830 kg/m^3 and a thermal expansion coefficient of 12.5 μm/m *K*.

Results

First, resulting temperatures from the thermal network model are summarized and then thermally distorted geometry will be presented. Loaded tooth contact analysis results will be displayed at the end where the influence of different parameters

Table 1 Geometry parameters of the spur gearset

	Symbol	Value
Number of teeth	$z_{1,2}$[-]	25, 25
Normal module	m_n [mm]	4
Normal pressure angle	α_n [°]	20
Face width	b [mm]	15
Profile shift coefficient	$x_{1,2}$ [-]	0, 0
Operating center distance	a_w [mm]	100
Reference profile acc. ISO 53	[-]	C (1.25/0.25/1)
Tolerance field acc. DIN 3967	[-]	cd25

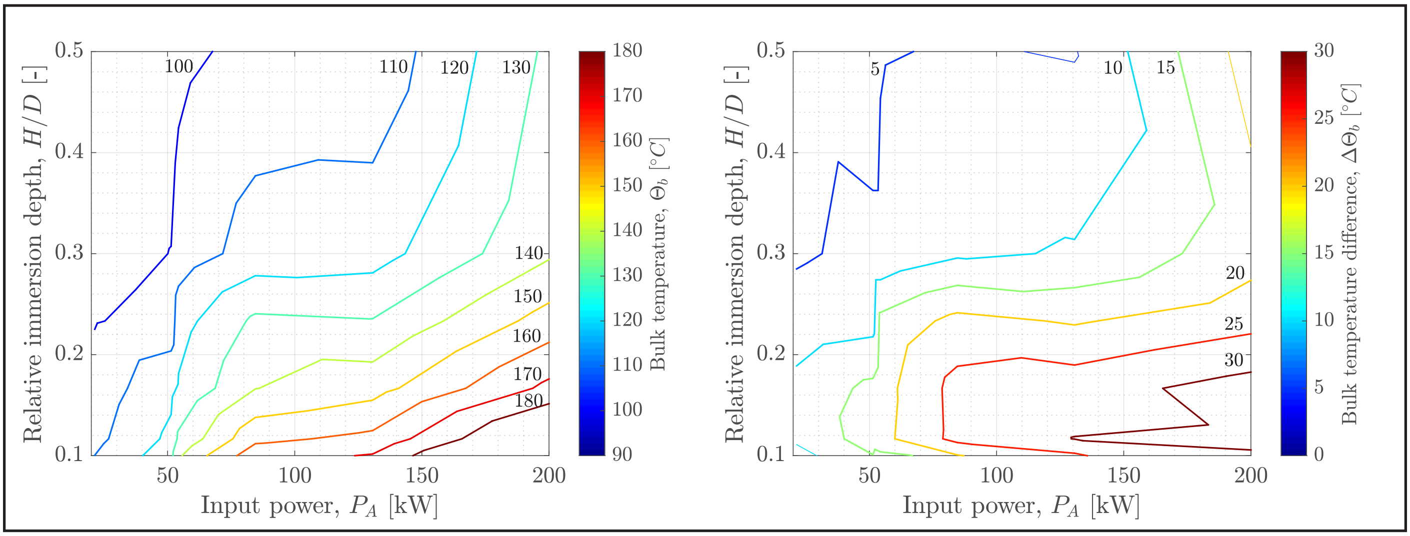

Figure 4 Predicted temperatures and gradients for different input powers and immersion depths.

will be analyzed in depth.

Temperature distribution. The different combinations of torque, speed and immersion depth result in bulk temperatures in the range between 90°C to 160°C. No appreciable thermal gradient has been found between pinion and gear because the speed ratio is 1 and the oil level covers the same area in both gears; therefore, the convection heat transfer is identical, which explains the absence of temperature differences.

In order to simulate unfavorable thermal conditions, two additional situations have been studied: i) idler gears with additional heat inputs from multiple meshes; and ii) reduction gear units. Figure 4 presents the results of the latter with $z_1=20$ and $z_2=30$, while the rest of the parameters are maintained. As it can be seen, significant temperature gradients exist at the highest input powers and lowest immersion depths.

Thermally induced geometry distortion. Figure 5 depicts the degree of distortion of the gears described in Table 1 with respect to the original geometry manufactured at 20°C. Distorted geometry at the oil sump temperature of 80°C is calculated along with the geometry change at a sufficiently high temperature, 140°C. In both cases, it is seen that a constant temperature increase of the gear (without temperature gradients in the direction of the radius), increases the tip diameter, reducing the manufacturing allowance that approaches zero. However, the difference at the root is not that important.

Such a difference between the tip and the root indicates that a considerable profile slope deviation exists. This is explained by the thermally induced base diameter increase, f_b, which is interrelated to the pressure angle deviation, f_α, by means of the following equation.

$$f_b = f_\alpha \cdot d_b \cdot \tan(\alpha) \tag{4}$$

Furthermore, profile form errors are also present because the diagram on the right (Fig. 5) shows a slight curvature that deviates from the mean shape of the profile. However, such differences seem to be strictly located in the root region and they will not be addressed in the current paper. Finally, tip diameter increase is also shown, which may act on the length of the line of action and the contact ratio. All of these parameters are further analyzed in the next sections.

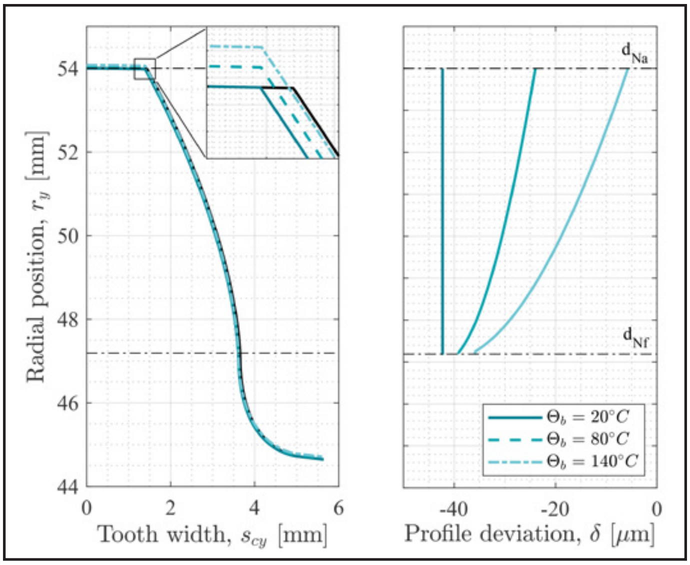

Figure 5 Profile geometry distortion at different operating temperatures.

Loaded tooth contact analysis without thermal effects. In order to understand the effects of thermally induced geometry distortions on contact parameters, the reference calculations with variable torque (and ambient temperature) are presented (Fig. 6). Loaded tooth contact analysis shows the normal behavior under load, with a non-dimensional load sharing factor, R, between 0.4 and 0.6. Significant contact outside the theoretically defined phase of mesh is also observed, where the simulated start/end of the active profile takes place 1.5 degrees before/after expected. On the other hand, loaded transmission error (Fig. 6b) shows an increasing peak-to-peak value with torque, but also decreasing mean values — indicating that the driven gear lags behind its theoretical value due to the increased tooth deflection.

Moreover, both Figures 6a and 6b show a symmetrical

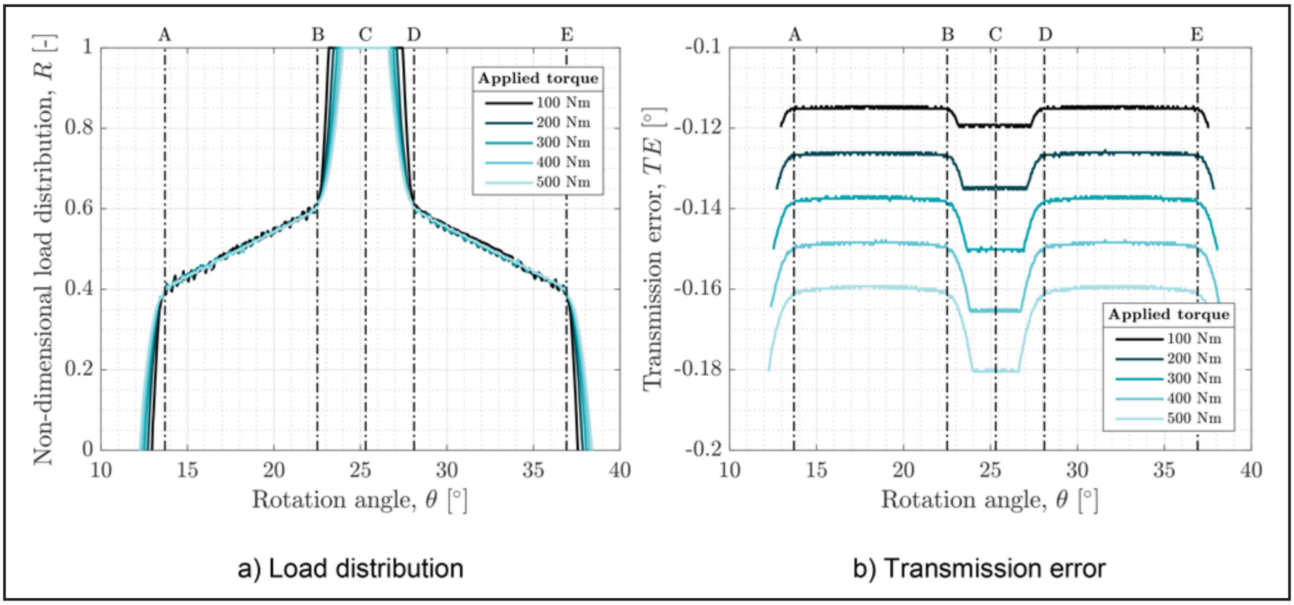

Figure 6 Influence of torque on load distribution and transmission error.

behavior with respect to the pitch point, C, indicating that both geometries are identical. That is to say, the length of approach equals the length of recess. Table 1 also shows that no design profile shift has been considered and the same tolerance field is applied to both gears, resulting in equal manufacturing profile shift coefficients $X_E = -0.0309$. Therefore, pinion and gear are geometrically identical and, as a result, have the same stiffness.

Influence of temperature increase on LTCA. In the same way as torque, temperature increase also extends the length of the path of contact. Tip diameter is increased proportionally to temperature and therefore the start of the active profile comes earlier and the end of it arrives later. In general terms, the trend of temperature increase is comparable to that of load. Figure 7a depicts this phenomena at a constant torque of 400 Nm and variable temperature (in the range of the numerically predicted values). However, as it can be observed (Fig. 7b) the influence of temperature on the length of the line of action is lower than that of torque. An increase of 100 Nm extends the path of contact 2%, while a 100°C increase implies approximately 1%.

If the shape of the thermally affected load distribution diagram (Fig. 7a) is compared to that of load (Fig. 6a), it is seen that temperature increase does not influence the symmetry. Thermal expansion affects pinion and gear shape in the same manner and therefore there is no geometrical difference; hence, no stiffness difference either.

On the other hand, the increase of temperature affects transmission error appreciably (Fig. 8a). The reduction of the mean level of transmission error is fully explained by the reduction of the available backlash angle (Fig. 8b). For instance, at 300 Nm the decrease of the backlash angle from 20°C to 140°C is 0.0527 degrees, which corresponds to the mean step between both curves at the same operating temperatures (Fig. 8a).

Influence of temperature gradient on LTCA. In order to analyze the effect of the temperature gradient, the pinion temperature is maintained constant at 140°C, while gear temperature decreases. This case, with different temperatures for identical gears, represents multi-mesh transmissions where a gear could be engaged with multiple others and consequently the

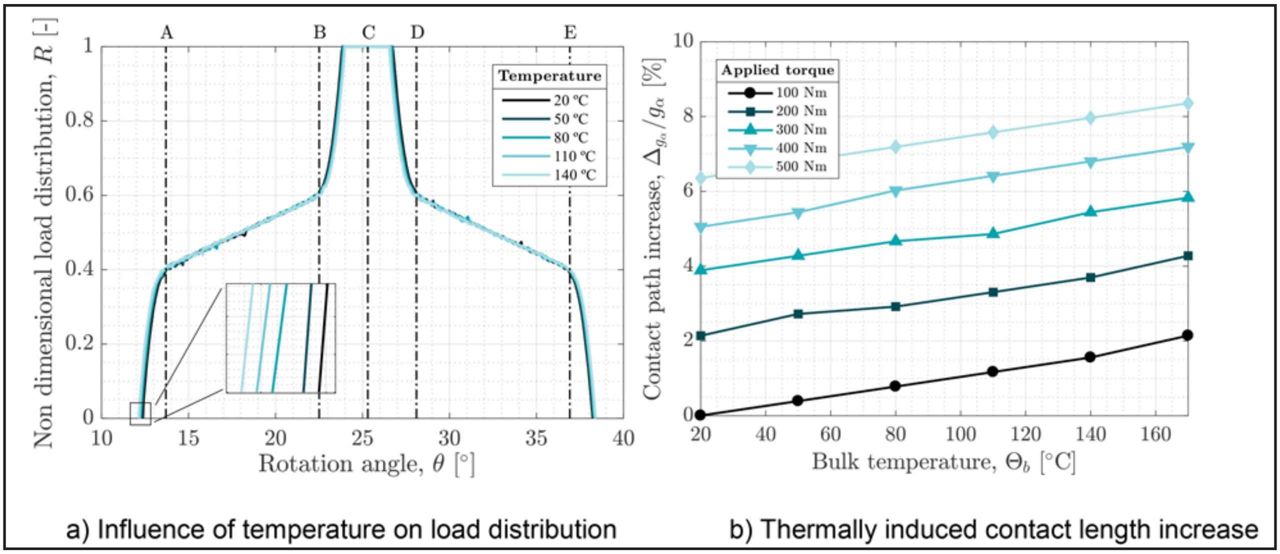

Figure 7 Influence of temperature increase on load distribution at T = 400 Nm.

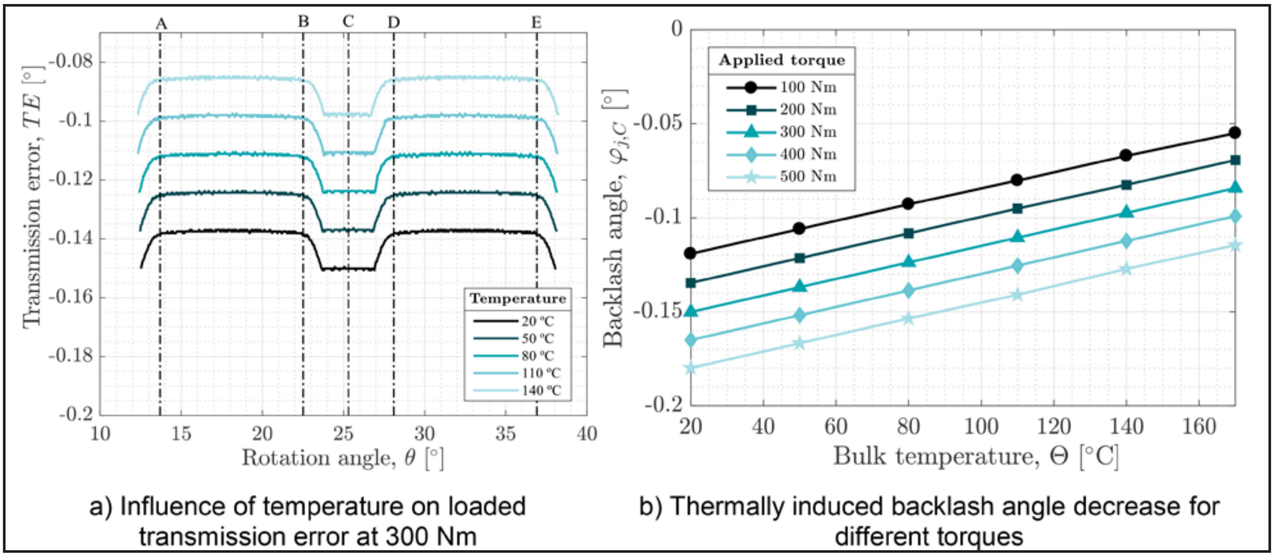

Figure 8 Influence of temperature increase on transmission error at T = 300 Nm.

temperature of the former would be higher than the rest.

When a temperature gradient exists, the thermal expansion affects in a different manner on the gear and the pinion; therefore the load distribution and the transmission error are no longer symmetrical. Figure 9 shows the influence of temperature gradient between pinion and gear on the load distribution and transmission error.

The asymmetrical behavior of the load distribution and transmission error slightly increases as the temperature gradient rises. Additionally, it is noticed an increase of the overall transmission error while in the double tooth contact region (between points A–B and D–E), transmission error shows an increasing slope. The increase in transmission error is explained by the differences in the profile deviations of the pinion and the gear caused by different temperatures.

Influence of gear ratio. Dissimilar gear pairs compose most gear transmissions; therefore, the temperature levels reached in each body are also different. Figure 10 shows the load distribution and the transmission error of a gear pair with a reduction factor of 1.5.

First, the case of no temperature gradient is analyzed (140°C) in order to account for the effects of the gear size. In this case the increase of the temperature produces a larger expansion on the gear than on the pinion, inducing an asymmetrical load distribution. Similar load distribution is observed when identical gears are subjected to thermal gradients, as has been seen in the previous section. If the thermal expansion of the pinion is larger than the gear's, the load distribution decreases in the approach double contact region and is increased in the recess double contact part.

However, when temperature gradients are introduced in a reduction gear unit, the load distribution asymmetry is compensated (Fig. 10a).This effect is fully explained with the compensation of pinion and gear thermal expansions. The pinion has a smaller size than the gear, but larger temperature, thus

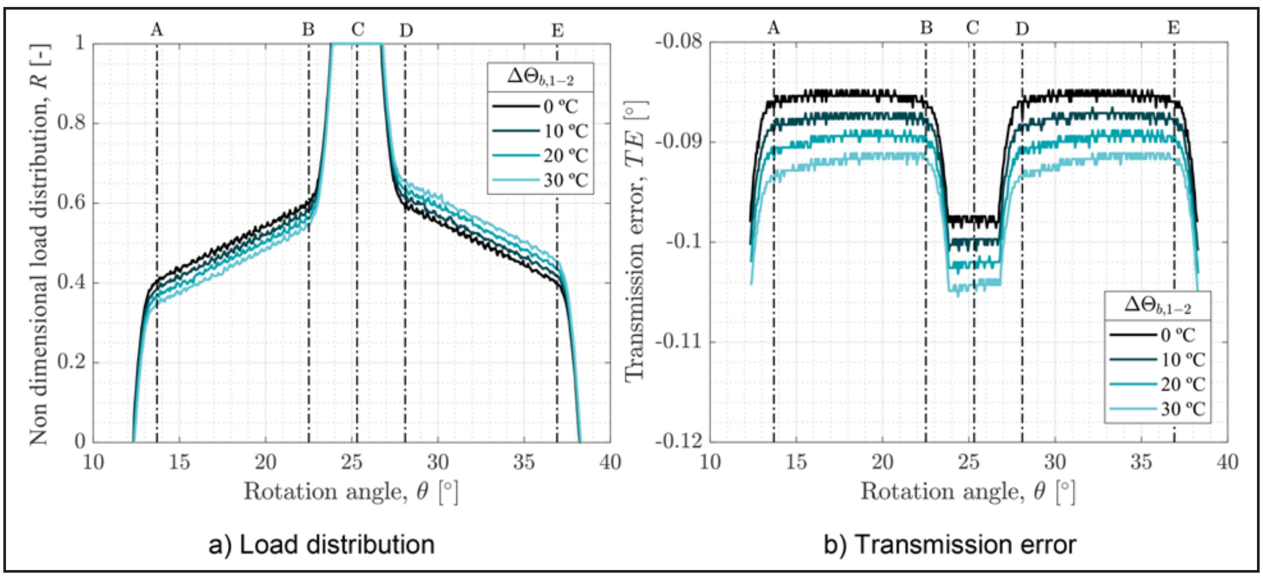

Figure 9 Influence of temperature gradient between pinion and gear on load distribution and transmission error at T = 300 Nm and Θ_1 = 140°C.

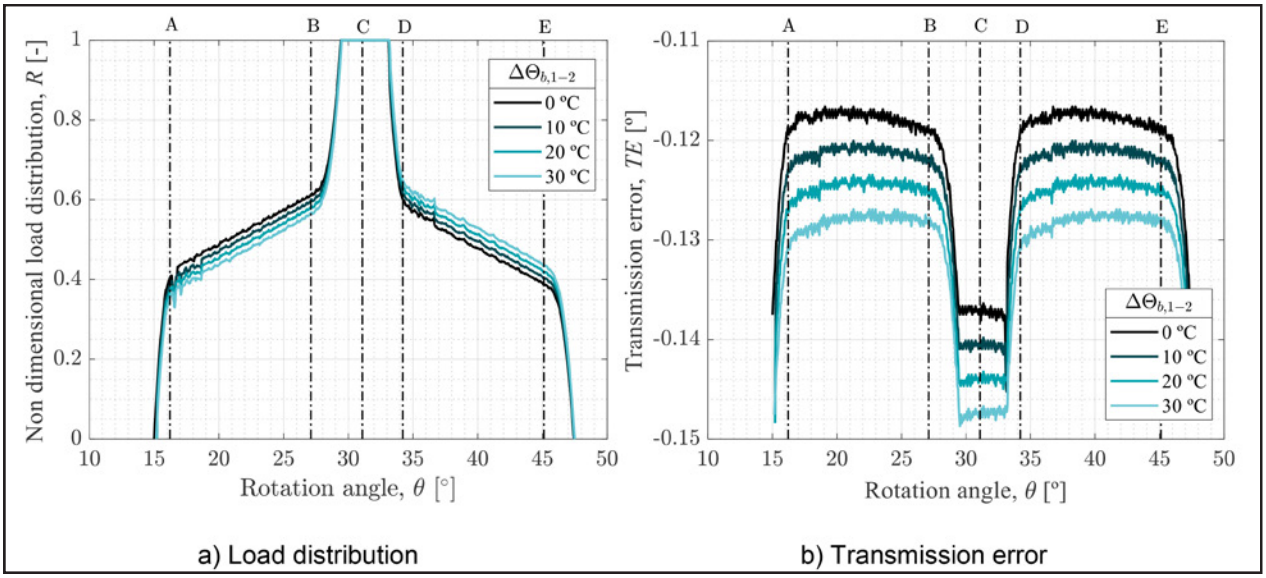

Figure 10 Influence of temperature gradient between pinion and gear on load distribution and transmission error at T = 300 Nm and Θ_1 = 140°C; case of reduction unit.

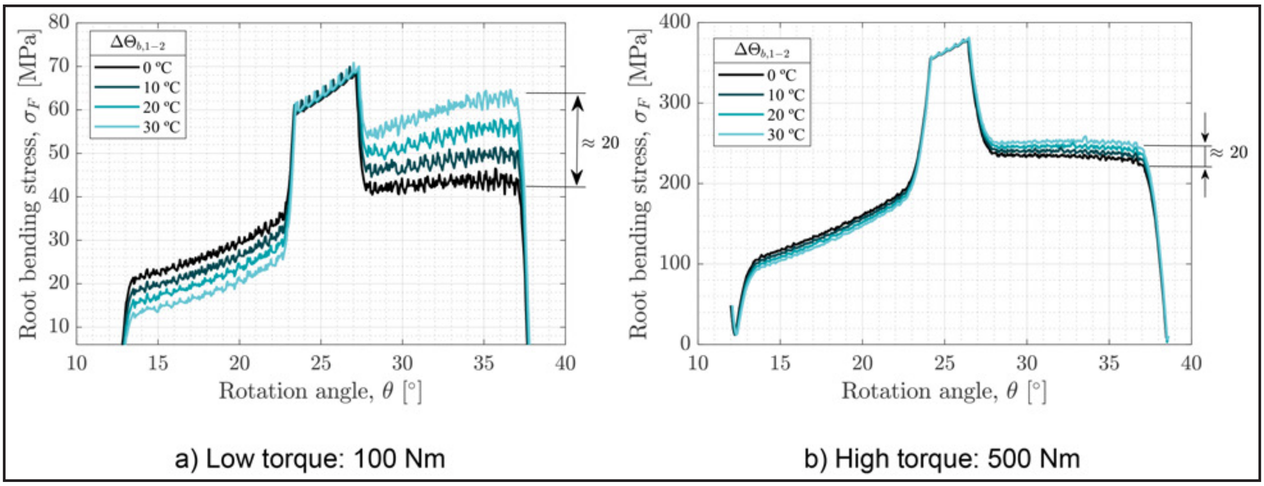

Figure 11 Influence of combined thermal and mechanical loads on root stresses at $\Theta_1 = 140°C$.

balancing the difference on thermal expansion and its effects.

Regarding the effects of thermal gradients of reduction transmissions on the transmission error, an increase of the transmission error is observed when the temperature gradients are higher (Fig. 10b). In this case, the decrease of the temperature on the gear increases the backlash while the transmission error increases.

Discussion

From the observation of Figures 6–10, it can be concluded that the most significant phenomena involved in the thermally and mechanically loaded gear behavior with respect to the simply loaded case is the reduction of transmission error; this is due to the available backlash angle decrease and the shift in the load distribution diagram when thermal gradients or relative pitch errors are present. However, in the preceding simulations load has been kept constant while varying temperature and their influence have not been interrelated.

Combined elastic and thermal deflections. In general, in cases where there is a differential base pitch increase, the geometry distortion is partially offset by elastic tooth deformations. However, when thermal loading is such that the expansion in the direction of the line of action is larger than the elastic deformation, load distribution can be significantly affected. This situation may affect mean tooth stress level and the position of the maximum value.

Figure 11 shows an example of such a situation. When the pinion drives the gear, the existence of torque reduces the base pitch of the pinion proportionally to the deflection of the loaded pinion tooth, which increases the base pitch of the gear by an amount equal to its deflection. The sum of both deflections determines the load distribution, and therefore tooth stresses. However, when thermal effects are considered, the existence of a temperature differential increases the base pitch of both the pinion and the gear proportionally to the tooth expansion. As a consequence, the load distribution diagram is altered beforehand. If thermally induced deformations are as large as elastic deflections, the load distribution diagram turns symmetrical because both phenomena compensate each other. Figure 11a shows the tooth root stress levels when thermal distortions are larger than elastic deformations, while Figure 11b shows compensated deflections. At 100 Nm the distortion of the bending stress diagram for a temperature difference of 30°C is as large as 30% in the recess, while at 500 Nm it barely represents 5%.

It is interesting to note that in both cases the stress difference between the isothermal case and that with maximum temperature gradient in the recess path of contact is approximately equal. This happens because in both cases the thermally induced pitch error is equal, as it is a function of the temperature gradient and not the torque. Therefore the additional deflection that the gears must overcome in both cases is the same, and the local load difference is almost equal — resulting in the same stress differential.

Design recommendations for enhanced contact behavior. In order to compensate the dissimilar root bending stresses (Fig. 11a), profile modifications need to be applied. To this aim, the amount of deviation at the operating temperature is computed and represented (Fig. 12). It is observed that pitch deviations increase proportionally to temperature, while profile slope decreases — both of which are the expected behaviors. For a pinion temperature of 140°C and a gear at 110°C, pitch differentials are 13 µm and 17 µm, respectively and the profile slope decrease corresponds to 27 µm and 20 µm. Therefore pinion and gear are applied to these modifications at ambient temperature to enhance contact behavior at steady-state operating conditions (assuming that the generated heat is not affected and the steady-state bulk temperature is the same).

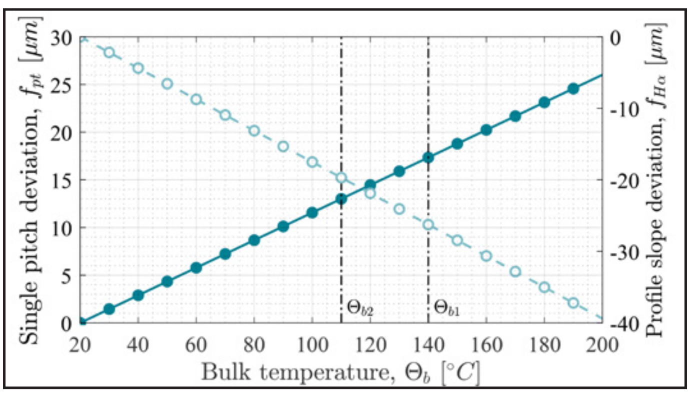

Figure 12 Profile deviations as a function of temperature and corresponding modifications.

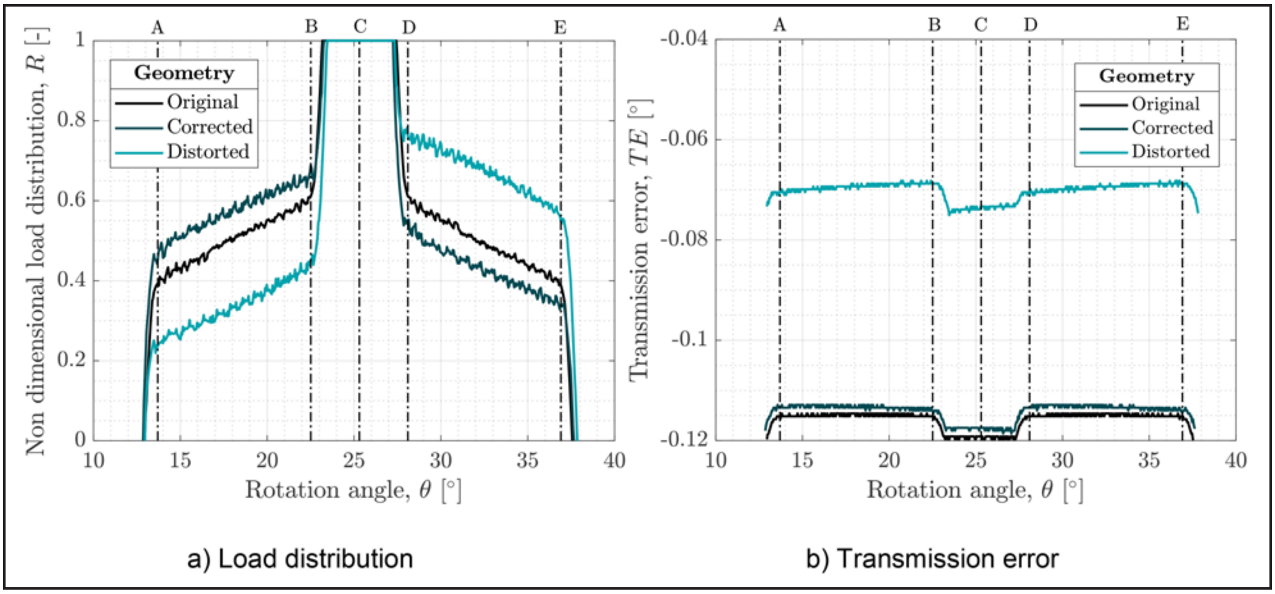

Figure 13 **Comparison of load distribution and transmission error between profiles at T = 100 Nm and Θ_1 = 140°C and Θ_2 = 110°C.**

Figure 13 shows the contact parameter comparison between the reference state at ambient temperature, the thermally distorted one, and the corrected geometry. As can be seen (Fig. 13a), the load distribution has been improved but a significant difference of up to 10% is still present, indicating that profile form errors may not be negligible. On the other hand, it is shown (Fig. 13b) that the initial transmission error levels are recovered with small differences due to the aforementioned form errors.

Conclusions

In this paper the effects of thermally induced geometry distortions on load distribution and transmission error have been analyzed. A thermal-lumped parameter model accounting for radial temperature distribution has been coupled to a two-dimensional finite element model. Analytically predicted temperatures have been loaded on the FEM model to compute geometry distortion, and tooth contact analysis has been carried out. The results show that at high input powers and low oil levels, temperature differences arise between pinion and gear, resulting in pitch errors and profile slope deviations. Such profile distortions affect load distribution — especially at low loads and high temperatures — leading to a non-symmetrical diagram that affects bending stress levels. Transmission error is fundamentally influenced by the backlash angle decrease shifting the mean level towards zero. Profile modifications have been proposed and it has been shown that mechanical and thermal deviations can be predicted and compensated at the manufacturing stage.

For more information.

Questions or comments regarding this paper? Contact Dr. Jon Larranaga at *jlarranaga@mondragon.edu*.

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