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INVESTIGATION OF THE EFFECTIVE PARAMETERS OF SCUFFING FAILURE IN GEARS

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Abstract. This study investigates the effective parameters of scuffing failure in gears using the integral temperature method. For this aim, the mass temperature, integral temperature and scuffing safety factor are calculated for a given parameters. Then, integral temperatures are simulated based on various geometrical, operational and lubrication parameters. Obtained results are presented graphically. The obtained results show that increasing the module m_n results in a decrease in the integral temperature ϑ_{int}. Similarly, increasing the pinion teeth number z_p results in a decrease in the integral temperature ϑ_{int}. Increasing the module and tooth number positively affects the scuffing failure in gears. In contrast, increasing the transmitted torque M_{T1T} results in an increase in the integral temperature ϑ_{int}. Similarly, increasing the pinion speed n_p increases the mass temperature ϑ_M, and increasing the lubricant (oil) $\vartheta_{Ö}$ temperature increases the integral temperature ϑ_{int}. Increasing the transmitted torque, lubricant temperature and the pinion speed negatively affects the scuffing failure in gears. Finally, increasing the nominal kinematic viscosity v_{40} decreases the integral temperature ϑ_{int}. Increasing the nominal kinematic viscosity positively affects the scuffing failure in gears. By considering the effective parameters of scuffing failure such as geometrical, operational and lubrication, one can design and manufacture the desired gears without scuffing failure.

Keywords: gear failure, scuffing, kinematic viscosity, mass temperature, integral temperature.

Introduction

Gears are widely used to mechanically transmit power in rotating machines and in the vehicle industry. Engineers always aim to design and manufacture high-strength gears to safely transmit the required power without failure.

Gear failures are classified into **tooth breakages** as a result of tooth bending stress, **surface pitting** as a result of tooth contact stress and **scuffing** as a result of high temperature and high contact stress. Gear failures can be prevented by considering these potential failure conditions.

Gears are designed by considering the potential failure mechanism, and different standards were developed by organizations such as ISO, DIN and AGMA. The following information on scuffing failure in gears can be found in the literature:

– the lubricant (oil) temperature, lubricant (oil) level and surface roughness are considered effective parameters for scuffing failure phenomena in gears. The effect of the oil temperature on gear failure in experiments is presented. The thickness of the protective elastohydrodynamic lubricant film depends on the lubricant viscosity at the operating temperature. Thus, high temperatures lead to the formation of a low-viscosity and thin oil film, high chemical activity and good tribological layer formation [1];

– lubricant (oil) level is considered an effective parameter for scuffing failure phenomena in gears. The influence of the immersion depth of dip-lubricated gears on the power loss, bulk temperature and scuffing capacity is experimentally and numerically studied. It is concluded that the occurrence of the gear scuffing failure mode is strongly determined by the lubricant (oil) levels [2].

Surface roughness is considered an effective parameter for scuffing failure phenomena in gears. A surface coating is applied to provide the required surface quality and low friction on gears. The effect of low-friction coatings on the scuffing load capacity and efficiency of gears is presented. Surface coatings can reduce the friction coefficient between gear teeth by 8–41 % depending on the operating speed and coating type. Coated gears improve the efficiency of a gearbox in low-speed, high-torque conditions [3].

The gear material, heat treatment and surface (contact) pressure are also effective parameters for scuffing failure phenomena. Scuffing risk to spur and helical gears in commercial vehicle transmissions is predicted by the flash temperature method. It is concluded that when the maximum contact temperature is close to or above the limiting scuffing temperature for the combination of lubricant and gear material, scuffing likely occurs [4].

Surface finishing is also an effective parameter to provide the required lubrication condition for scuffing failure phenomena. To reduce the risk of scuffing, isotropic superfinishing is presented. It is concluded that decreasing the surface roughness, particularly through isotropic superfinishing, significantly increases the scuffing resistance because of the increased lubricant film thickness and decreased friction [5].

The effective parameters of scuffing failure in gears are investigated using the integral temperature method, which is defined by ISO 6336 and DIN 3990 standards.

By varying geometrical parameters such as the module and number of teeth, the relationship between the module and integral temperature and the relationship between the number of teeth and integral temperature relation are graphically presented. Similarly, by varying the operational parameters such as the transmitted torque and pinion speed, the relationship between torque and integral temperature is graphically presented. By considering the lubrication (oil) parameters, the relationship between lubricant (oil) temperature and integral temperature and the relationship between the nominal viscosity of the oil and the integral temperature are graphically presented.

Problem Statement

A particularly severe form of gear tooth surface failure occurs when areas of tooth surfaces seize or are welded together due to the absence or breakdown of the lubricant film between the contacting tooth flanks of the mating gears, which is caused by high temperature and high pressure. This form of failure is termed **"scuffing"** and is most relevant when the surface velocities are high. Scuffing failure may also occur at relatively low sliding velocities when the tooth surface pressures are sufficiently high. A scuffed gear surface feature has a matte appearance, as shown in Fig. 1.

Fig. 1. Scuffed gear surface [5]

Methodology

The integral temperature criterion to evaluate the probability of scuffing is based on the assumption that scuffing likely occurs when the mean contact temperature along the path of contact is equal to or exceeds the corresponding **"permissible integral temperature"**.

In the integral temperature method, the sum of the bulk temperature and the weighted mean of the integrated values of flash temperatures along the path of contact is the **“integral temperature”**.

The probability of scuffing is assessed by comparing the integral temperature with a corresponding **“permissible integral temperature”**, which is derived from the gear testing of lubricants for the scuffing resistance or from gears that have scuffed in service.

The integral temperature ϑ_{int} is calculated as follows [6]–[9]:

$$\vartheta_{\mathrm{int}} = \vartheta_M + C_2 \vartheta_{fla\,\mathrm{int}} \le \vartheta_{\mathrm{int}P}, \tag{1}$$

where ϑ_{M} [°C] is the mass temperature of the tooth surface immediately before the intervention; C_2 is the weighting factor from experiments, and we assume that C_2=1.5; $\vartheta_{fla\mathrm{int}}$ [°C] is the average flash temperature; and $\vartheta_{\mathrm{intP}}$ [°C] is the permissible integral temperature.

Mass temperature ϑ_{M} [°C] is calculated as follows [6]–[9]:

$$\vartheta_M = X_S \cdot \left(\vartheta_{\ddot{O}} + C_1 \vartheta_{fla\,\mathrm{int}}\right), \tag{2}$$

where X_S is the lubrication factor and we assume that X_S = 1 for splash lubrication, X_S = 1.2 for injection lubrication, and X_S = 0.2 for gear wheels that completely run in oil; $\vartheta_{\ddot{O}}$ [°C] is the lubricating oil temperature before tooth engagement; C_1 is the weighting factor from experiments, and we assume that C_1 = 0.7; and $\vartheta_{fla\mathrm{int}}$ [°C] is the average flash temperature.

The average flash temperature $\vartheta_{fla\mathrm{int}}$ is calculated as follows [6]–[9]:

$$\vartheta_{fla\,\mathrm{int}} = \vartheta_{flaE} . X_{\varepsilon}, \tag{3}$$

where ϑ_{flaE} [°C] is the flash temperature at head engagement point E of the pinion; and X_{ε} is the contact ratio factor.

The flash temperature at the head engagement point E of the pinion ϑ_{flaE} [°C] is calculated as follows [6]–[9]:

$$\vartheta_{flaE} = K.\mu_{mC} \cdot \frac{w_n \left|v_{t1} - v_{t2}\right|}{\sqrt{2.b_h} \cdot \left(B_{M1}.\sqrt{v_{t1}} + B_{M2}.\sqrt{v_{t2}}\right)}, \tag{4}$$

where K is the empirical factor K = 1.11; μ_{mC} is the average coefficient of friction; w_n [N/mm] is the line load in the normal direction; $v_{t1,2}$ [m/s] are the tangential velocities of radii 1 and 2; b_h is the face width; and $B_{M1,2}$ are the thermal contact coefficients of radii 1 and 2.

The permissible integral temperature $\vartheta_{\mathrm{intP}}$ is calculated as follows [6]–[9]:

$$\vartheta_{\mathrm{int}P} = \frac{\vartheta_{\mathrm{int}S}}{S_{S\min}} \tag{5}$$

with

$$\vartheta_{\mathrm{int}S} \approx \vartheta_{MT} + X_{WrelT} . C_2 \vartheta_{fla\,\mathrm{int}T}, \tag{6}$$

where ϑ_{MT} [°C] is the mass temperature as a function of the pinion torque; X_{WrelT} is the relative structure factor; $\vartheta_{fla\mathrm{int}T}$ [°C] is the average flash temperature, index T denotes the test values during the scuffing test run in the test bench; and $S_{S\min}$ is the minimum scuffing safety factor.

The mass temperature ϑ_{MT} and the average flash temperature $\vartheta_{fla\mathrm{int}T}$ during the scuffing test run in the test bench are calculated as follows [6]–[9]:

$$\vartheta_{MT} = 80 + 0.23 \times M_{T1T} \tag{7}$$

and

$$\vartheta_{fla\,\mathrm{int}T} = 0,08.M_{T1T}^{\;1,2} \cdot \left(\frac{100}{v_{40}}\right)^{v_{40}^{-0,4}}, \tag{8}$$

where M_{T1T} [N·m] is the pinion torque; and v_{40} is the nominal kinematic viscosity.

The computational scuffing safety factor $S_{\mathrm{int}S}$ with the integral temperature process is calculated as follows [6]–[9]:

$$S_{\mathrm{int}S} = \frac{\vartheta_{\mathrm{int}S}}{\vartheta_{\mathrm{int}}} \ge S_{S\min}. \tag{9}$$

The following three conditions are true for scuffing failure phenomena:
– if $S_{\mathrm{int}S} > 1.0$: scuffing failure does not occur in the gears;
– if $S_{\mathrm{int}S} = 1.0$: scuffing failure limit;
– if $S_{\mathrm{int}S} < 1.0$: scuffing failure occurs in the gears.

Numerical Example

In the numerical study, the relationship between the effective parameters and the integral temperature are simulated based on the assumed *geometrical, operational and lubrication* parameters. The geometrical parameters such as *module* m_n, *number of teeth* z, and *face width* b_h are shown in Table 1.

Table 1

Geometrical parameters

Geometrical parameters	Unit	Value
Normal module m_n	[mm]	9
Number of pinion teeth z_1	[–]	17
Face width b_h	[mm]	60

Operational parameters such as *transmitted power* P, *pinion speed* n_p and *scuffing torque* (or transmitted torque) M_{T1T} are shown in Table 2.

Table 2

Operational parameters

Operational parameters	Unit	Value
Transmitted power P	[kW]	410
Pinion speed n_p	[min^{-1}]	1300
Scuffing torque M_{T1T}	[N·m]	372

Lubrication parameters such as *oil temperature* $\vartheta_{Ö}$, *lubricant kinematic viscosity* v_{40}, *lubrication factor* X_S, and *friction coefficient* μ are shown in Table 3.

Table 3

Lubrication parameters

Lubrication parameters	Unit	Value
Oil temperature	[°C]	100
Lubricant kinematic viscosity at 40 °C v_{40}	[mm^2/s]	220
Lubrication factor X_S	[–]	1.0
Friction coefficient μ	[–]	0.073

By considering the geometrical parameters, operational parameters and lubrication parameters in Tables 1–3, the scuffing parameters are calculated and presented in Table 4.

Table 4

Scuffing parameters

Scuffing parameters	Unit	Value
Mass temperature ϑ_{MT}	[°C]	165.56
Integral temperature ϑ_{int}	[°C]	263.67
Permissible integral temperature $\vartheta_{\mathrm{int}P}$	[°C]	298.68
Scuffing safety factor $S_{\mathrm{int}S}$	[–]	1.1328

The integral temperature ϑ_{int} reached the permissible integral temperature $\vartheta_{\mathrm{int}P}$, so the scuffing safety factor $S_{\mathrm{int}S}$ was equal to 1.1328 based on Equation (9), and the obtained values under safety for scuffing failure are presented in Table 4.

Simulation of the scuffing parameters

By varying the effective parameters of scuffing failure such as the geometrical, operational and lubrication parameters, the integral temperatures are simulated.

Effect of the geometrical parameters

By varying module m_n, the change in the integral temperature ϑ_{int} is simulated as shown in Fig. 2. Increasing module m_n results in a decrease in the integral temperature ϑ_{int}.

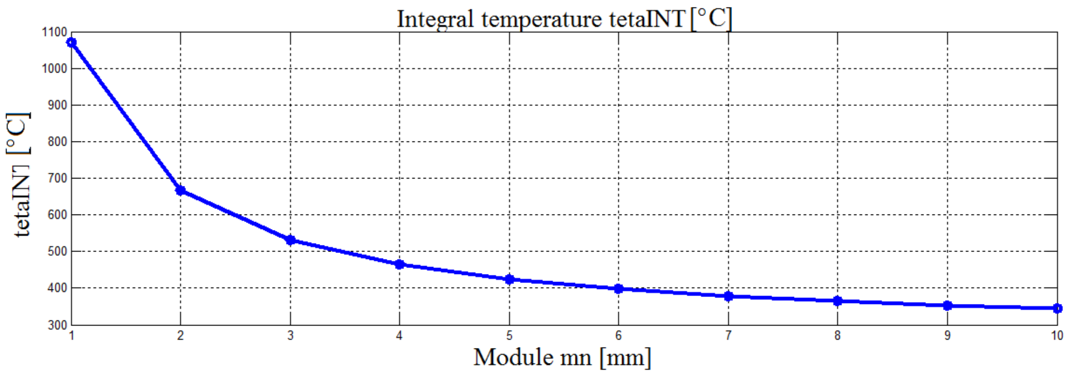

Fig. 2. Relationship between the module and integral temperature

By varying the tooth number z_p, the change in the integral temperature ϑ_{int} is simulated as shown in Fig. 3. Increasing the tooth number z_p results in a decrease in the integral temperature ϑ_{int}.

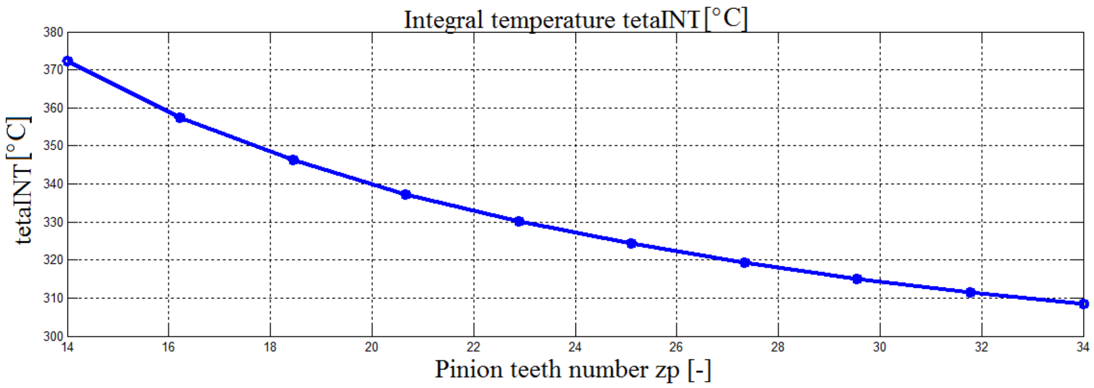

Fig. 3. Relationship between number of pinion teeth and integral temperature

By varying the transmitted torque M_{T1T}, the change in the integral temperature ϑ_{int} is simulated as shown in Fig. 4. Increasing torque M_{T1T} results in an increase in the integral temperature ϑ_{int}.

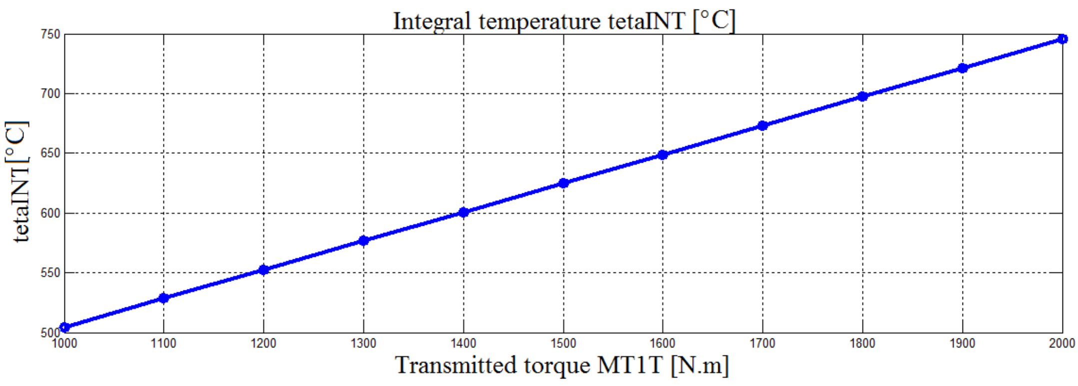

Fig. 4. Relationship between transmitted torque and scuffing integral temperature

By varying the pinion speed n_p, the change in the mass temperature ϑ_M is simulated as shown in Fig. 5. Increasing the pinion speed n_p results in an increase in the mass temperature ϑ_M.

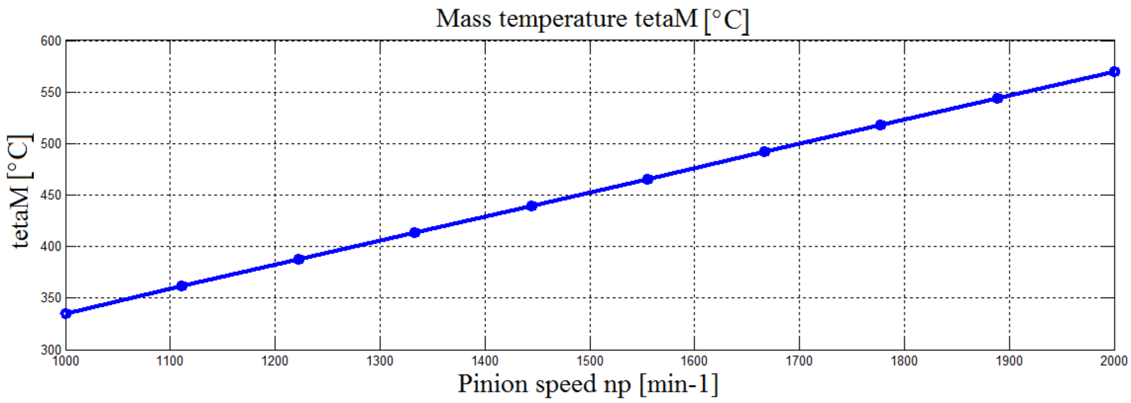

Fig. 5. Relationship between pinion speed and mass temperature

Effect of the lubrication parameters

By varying lubrication (oil) temperature $\vartheta_{Ö}$, the change in integral temperature ϑ_{int} is simulated as shown in Fig. 6. Increasing the lubricant (oil) temperature $\vartheta_{Ö}$ results in an increase in the integral temperature ϑ_{int}.

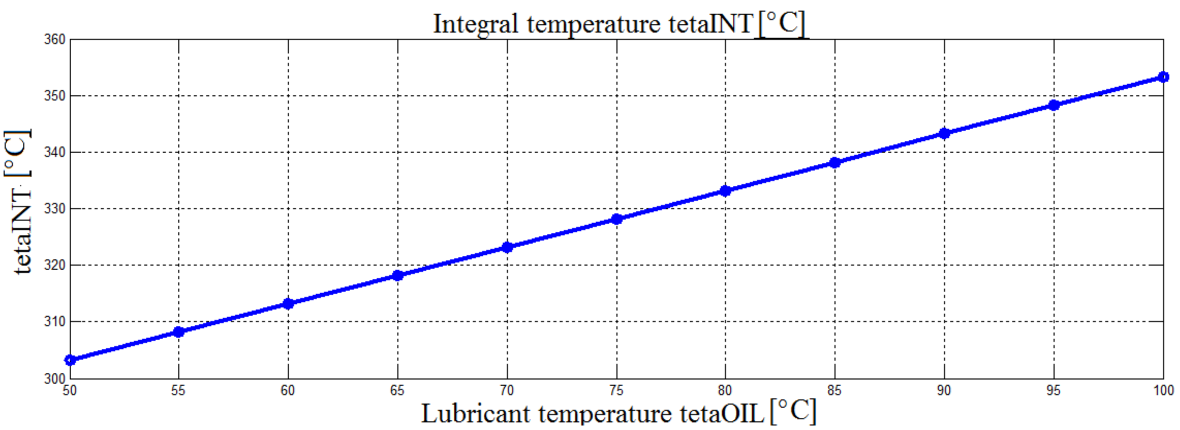

Fig. 6. Relationship between oil temperature and integral temperature

By varying the nominal viscosity ν_{40}, the change in the integral temperature ϑ_{int} is simulated as shown in Fig. 7. Increasing nominal kinematic viscosity ν_{40} results in a decrease in integral temperature ϑ_{int}.

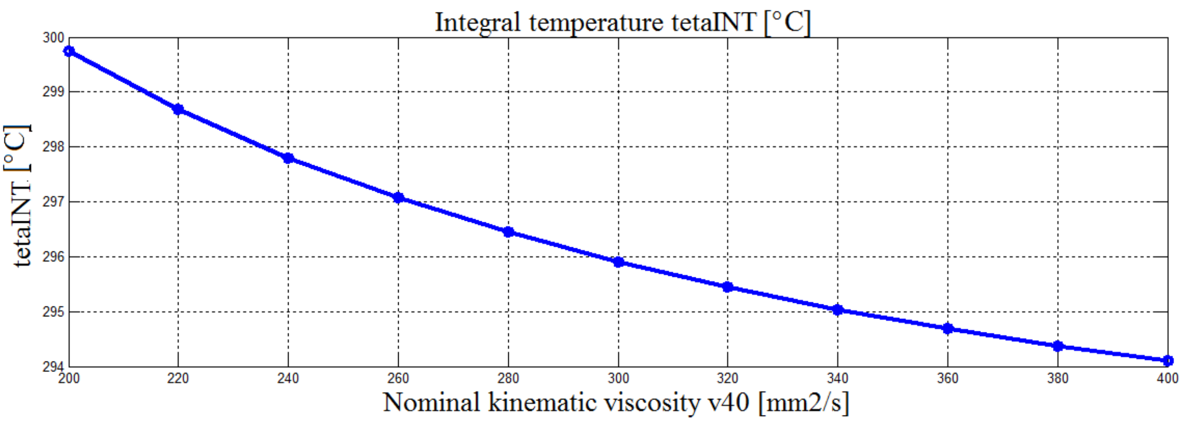

Fig. 7. Relationship between nominal kinematic viscosity and scuffing integral temperature

Results

The increase in the module m_n from 1 mm to 10 mm results in a decrease in the integral temperature ϑ_{int} from 1070 °C to 344 °C, as presented in Fig. 2. The increase in the number of pinion teeth z_p from 14 to 34 results in a decrease in the integral temperature ϑ_{int} from 372.40 °C to 308.39 °C, as presented in Fig. 3.

The increase in the torque M_{T1T} from 200 N·m to 2000 N·m increases the integral temperature ϑ_{int} from 504.49 °C to 745.39 °C, as shown in Fig. 4. The increase in the pinion speed n_p from 1000 min^{-1} to 2000 mm^{-1} increases the mass temperature ϑ_M from 335.23 °C to 570.23 °C, as presented in Fig. 5.

The increase in the lubricant (oil) temperature $\vartheta_{Ö}$ from 50 °C to 100 °C increases the integral temperature ϑ_{int} from 303.20 °C to 353.20 °C, as presented in Figure 6. Increasing the nominal kinematic viscosity v_{40} from 200 mm^2/s to 400 mm^2/s decreases the integral temperature $\vartheta_{\text{int}S}$ from 299.73 °C to 294.09 °C, as presented in Fig. 7.

Conclusions

Among the geometrical parameters, increasing the module m_n strongly decreases the integral temperature ϑ_{int}, and increasing the number of pinion teeth z_p slightly decreases the integral temperature ϑ_{int}. Increasing the module and tooth number positively affects the scuffing failure in gears. Furthermore, increasing the module and tooth number positively affects the tooth bending and tooth contact failure in gears

Among the operational parameters, increasing the torque M_{T1T} strongly increases the integral temperature ϑ_{int}. Similarly, increasing the pinion speed n_p strongly increases the mass temperature ϑ_M. Increasing the transmitted torque, lubricant temperature and the pinion speed negatively affects the scuffing failure in gears.

Among the lubrication parameters, increasing the lubricant (oil) temperature $\vartheta_{Ö}$ strongly increases the integral temperature ϑ_{int}. In contrast, increasing the nominal kinematic viscosity v_{40} slightly decreases the integral temperature ϑ_{int}. Increasing the nominal kinematic viscosity positively affects the scuffing failure in gears.

By considering the effective parameters of scuffing failure such as geometrical, operational and lubrication, one can design and manufacture the desired gears without scuffing failure.

The scuffing capacity of gears will be experimentally and statistically investigated by the author in a future study to determine the reliability level of gears.

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