# Maximum Surface Temperature of the Thermoplastic Gear in a Non-Lubricated Plastic/Steel Gear Pair

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**Abstract**

One of the major problems of plastic gear design is the knowledge of their running temperature. Of special interest is the bulk temperature of the tooth to predict the fatigue life, and the peak temperature on the surface of the tooth to avert surface failure. This paper presents the results of an experimental method that uses an infrared radiometer to measure the temperature variation along the profile of a plastic gear tooth in operation. Measurements are made on 5.08, 3.17, 2.54, 2.12 mm module hob cut gears made from nylon 6-6, acetal and UHMWPE (Ultra High Molecular Weight Polyethylene). All the tests are made on a four square testing rig with thermoplastic/steel gear pairs where the plastic gear is the driver. Maximum temperature prediction curves obtained through statistical analysis of the results are presented and compared to data available from literature.

## Introduction

It is common practice to rate thermoplastic gears by using the metal gear equation developed by Lewis[1] with correction factors to allow for differences in properties between metals and plastics. Although mechanical properties of plastics as a function of temperature are relatively well known,[2, 3, 4] they are of little help, if the temperature at which the gear operates, is not known. There is some information on how to calculate the bulk[5, 6] and the mean tooth surface temperatures.[6, 7, 8, 9] However, the maximum tooth surface temperature on the flank, which can be related to the flash temperature concept developed by Blok[10] for metal gears, is especially difficult to measure, and virtually no information is available in the literature concerning plastic gears.

This article presents the measurements made of the temperature variation along the tooth profile of a plastic gear in operation. The technique is similar to that used in a previous program to measure the mean tooth surface temperature.[8] For the present investigation, the plastic materials are unfilled and unreinforced polyamide 6-6 (nylon), polyoxymethylene homopolymer (acetal) and Ultra-High-Molecular-Weight-Polyethylene (U H.M.W.P.E.). Table 1 summarizes the geometry of the gears used. In all cases, the plastic gear acts as the pinion (driver) and is paired with a metal gear. The gears are run without lubrication on a four square testing rig.[7] A parametric equation, based on a large number of measurements, is proposed for each material investigated to predict the maximum temperature in the operating conditions described.

### Analysis of Temperature Evolution Around Gear Contour

Neglecting heat flow through shafts and radiation from surroundings, heat generated in a gear pair containing at least one plastic gear has two sources: friction and viscoelastic hysteresis.

Fig. 1 represents, on a arbitrary scale, the ideal distribution of friction losses on a tooth segment, A-B-C-P-D-E, of a

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typical 24 tooth nylon gear in mesh with a steel gear. The calculations are done using the load sharing technique. [11] The conditions are pitch line velocity $V = 7.2$ m/sec, tangential load per unit face width $W_t = 70$ N/mm and no lubrication. A representative coefficient of friction of $\mu = 0.1$ is used for dry nylon on steel.[1]

Hysteresis losses are due to cyclic stressing of the viscoelastic material. Assuming a linear viscoelastic material, hysteresis losses per unit volume of material are, for unidirectional stressing:[12]

$$\frac{\tan(\delta)}{1 + \tan^2(\delta)} \quad \frac{\omega}{2} \quad \frac{\sigma_0^2}{E'} \tag{1}$$

where $\tan(\delta)$ is the loss factor; $\sigma_0$ is the maximum stress amplitude; $E'$ is the storage modulus and $\omega$ the loading frequency.

Hysteresis losses in a volume of material to a depth of $p_b/8$ below the tooth surface ($p_b$ is the base pitch) are also shown on the same scale in Fig. 1. Both types of stressing are represented in that figure, i.e. contact compression and bending. A value of $\tan(\delta) = 0.17$ is used for nylon at 50 percent relative humidity and 37°C.

Fig. 1 indicates that, theoretically, most of the heat generated in a plastic/steel gear pair comes from friction and contact compression hysteresis on the loaded flank. Bending hysteresis losses, shown on the non loaded profile, appear to be almost negligible. Since little heat is generated on the non loaded profile (region A to B in Fig. 1) and knowing that plastics are poor heat conductors, the temperature in this region of the tooth is expected to be relatively low. However, beyond point B, temperature may reach a peak in the vicinity of point C, decrease towards the operating pitch point P, increase again to another peak in the region of the lowest point of contact D to finally reach, at E, the same value as A.

In other words, assuming friction and hysteresis as being the only heat sources, the temperature distribution along a tooth profile may be expected to follow the pattern pictured

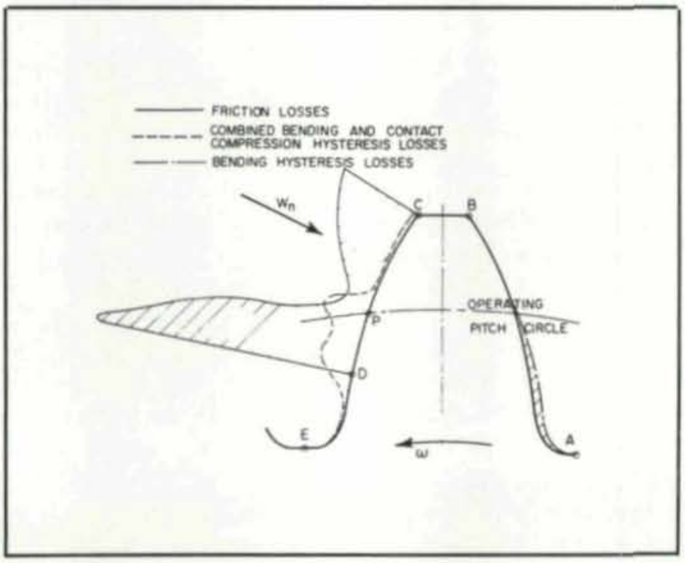


Fig. 1—Repartition of friction losses along gear profile.

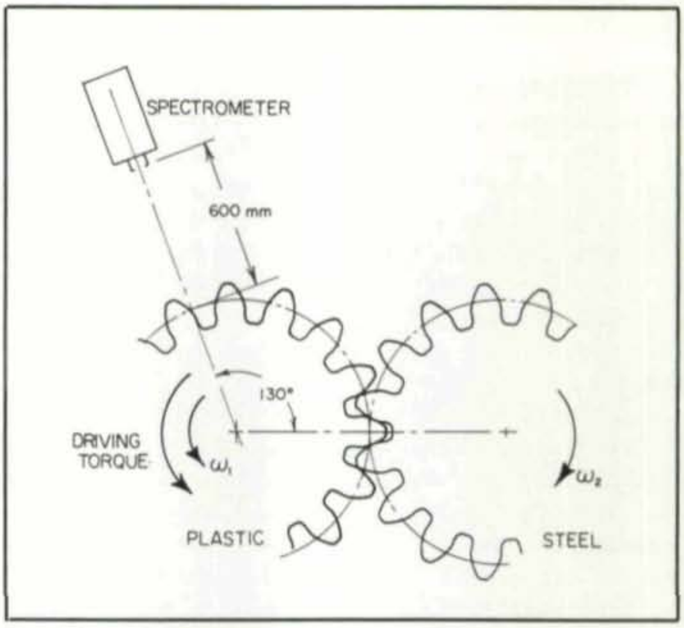


Fig. 2—Set-up for radiance measurements.

in Fig. 1. However, the peaks will be attenuated by conduction in the plastic gear tooth and the mating gear tooth, which is, in the present case, made of steel. Nevertheless, according to the theoretical evaluation of reference,[13] depending on load, module, speed, and mating gear material, temperature differences as much as 40°C may be recorded between region A-B and point C, 20°C between points C and P, and 50°C between P and D.

As Fig. 1 shows, the temperature gradients mentioned above occur over a relatively small surface on the tooth profile. From gear geometry and cinematic, it can be shown[13] that this surface decreases with the increasing number of teeth and speed. In the present investigation, the highest number of teeth in the plastic gear is 36 (2.116 mm module) and the highest speed is 9.2 m/sec. Noting that one complete cycle occurs between points B and D (Fig. 1), the frequency appears to be in the order of 3 KHz. This is the lowest frequency response characteristic the measuring instrument must have in order to detect the hottest points on the tooth profile.

Besides frequency response requirements, the measuring apparatus must be able to measure temperatures with adequate precision, have sufficient resolution to follow accurately temperature variations along the tooth profile, and sufficient depth of view to compensate for working distance variations due to the height of the teeth.

### System Used for Measurements

Fig. 2 is a schematic representation of the measuring setup. The measuring instrument is an infrared radiometer, which uses a nitrogen cooled indium antimonide detector, to provide a theoretical accuracy of .02°C on the calculated temperature, and can measure signals with a frequency up to 10 KHz.[14] This is well over the 3 KHz requirements stated in the preceding section. However, in the present application, such performances cannot be obtained mainly because of variations in the gear emissivity, which translate into an error on temperature, and the diameter of the measuring spot, which rounds off temperature peaks.

The experimental error on temperature is evaluated from the standard deviation of emissivity measurements made on new and worn gears of each material. From these measurements, it appears that the maximum error on temperature lies within limits set as $\pm 1.5$°C at 20°C and $\pm 3$C at 180°C. This error includes a variation of emissivity from gear to gear, which accounts for about 70 percent of the total, and a smaller error from tooth to tooth, which accounts for the rest. Therefore, for a given gear, it can be said that the maximum error at 180°C is in the order of $\pm 1$°C.

The size of the measuring spot is directly related to the field of view, which represents the solid angle that the target subtends, as viewed from the apparatus, and is defined by an unvariable field aperture built into the apparatus. Theoretically, the particular apparatus used has a conical field of view of 2.5 milliradians. This means that the target should be a circle of diameter equal to the base of a cone having an apex angle of 2.5 milliradians, and a height equal to the operating distance. This corresponds to a measuring spot of 1.5 mm at a working distance of 600 mm. Also, theoretically the detector output should be directly proportional to all of the radiation emitted from the surface of the circular target.

In practice, this does not appear to be exactly true; the actual shape of the target, field of view, and response curve of the detector, as given by the manufacturer,[14] are presented in Fig. 3. This figure shows that the actual field of view is an elliptical cone with minor and major axes of 3.0 and 5.0 milliradians respectively. For measurements reported in this paper, minor and major axes of the target are respectively aligned perpendicular and parallel to the axis of rotation of the gear.

If the response curve of the detector is approximated by a sinusoid, and the output signal is analyzed by Fourier decomposition, the original signal can, theoretically, be reconstituted by a computer program[15] to simulate a punctual measuring spot. However, this method is limited by random noise present in the signal. As the measuring spot size is further reduced by the Fourier decomposition, the noise becomes more and more important in relation to the temperature signal. Practical considerations limit the lower bound of the simulated spot size to approximately .5 mm. This is a threefold increase in the original resolution of 1.5 mm.

The depth view is measured as about 10 mm, which is adequate for a 5.08 mm module gear. The system used answers the requirements outlined in section 2.

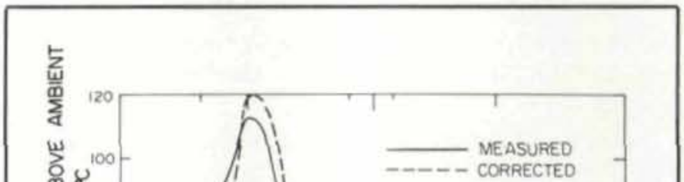


Fig. 4—Measured and corrected temperature signal for a used nylon. 5.08 mm module gear, Wt = 38 N/mm, V = 9.6 m/s.

For example, Fig. 4 presents the orginal and corrected temperature signals obtained with a used 5.08 mm module nylon gear. Although the variation of the maximum temperature is small, the shape of the signal is significantly altered, with peaks and throughs amplified, as can be expected with a smaller measuring spot. In this particular case, the correction is in the order of 8°C on the peak temperature (110°C). This is representative of the correction on temperatures of this order of magnitude.

Fig. 5 shows the corrected temperature signal for a 5.08 mm module UHMWPE gear. On this particular tooth, a reflective aluminum foil of low emissivity is installed on the top land of the tooth (C-D in Fig. 1) in order to precisely correlate the measured signal with specific points on the gear tooth surface. This explains the difference of the shape of the signals between Figs. 4 and 5.

Comparison of heat generation as calculated (Fig. 1) and temperature signals, as measured (Fig. 5) shows that the main temperature peak, which is predicted to be below the pitch circle by theoretical analysis, actually lies above the pitch circle for this particular signal. In fact, the shape of the temperature signal varies widely from one material to the other. In general, it is observed that the maximum temperature peak first appears below the pitch circle as predicted by the theory, when the gear is new; but, as the gear wears, two peaks of different amplitude, which could

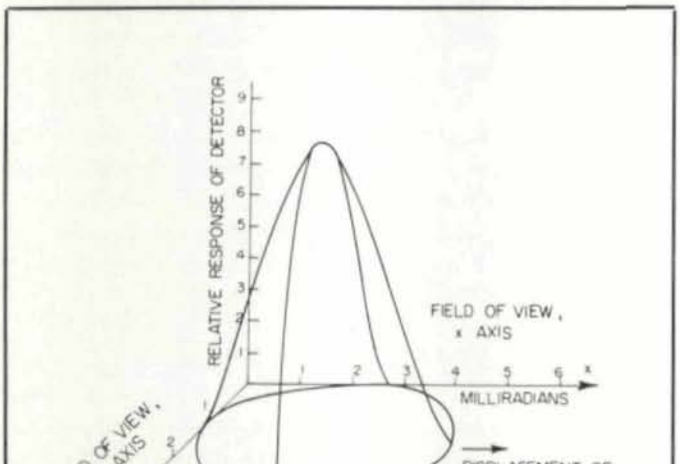


Fig. 3—Shape of the measuring spot and relative response of the detector.

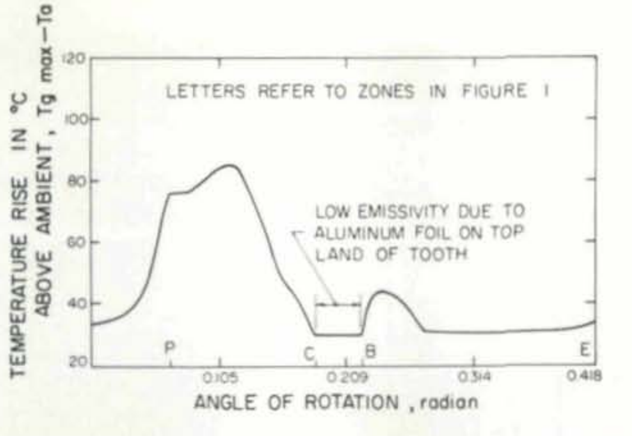


**Fig. 5**—Corrected temperature signal with aluminised land on top of tooth, 5.08 mm module UHMWPE, Wt = 28 N/mm, V = 9.6 m/s.

vary from tooth to tooth, start to develop gradually. This is attributed to the degradation of the profiles with wear and, the consequent departure from their exact shape with which the theoretical calculations are done.

### Results and Discussion

Some two hundred thirty measurements have been done with the radiometer. These were made with several conditions of load and speed for each material and modules. The plastic-metal gear pairs were first run until the operating temperature stabilizes. As mentioned previously, the measurements are taken on the plastic gear (driver) 130 degrees after the operating pitch point. The experimental results are then corrected for spot diameter, and they are statistically analyzed by computer to derive a parametric expression to match the maximum surface temperature as a function of the operating parameters. Of several models

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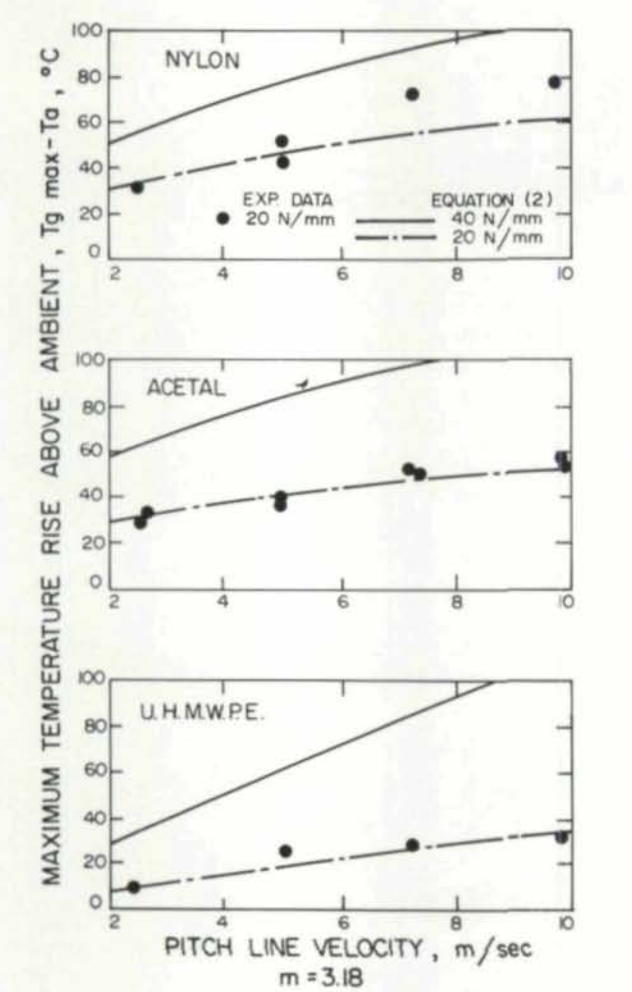


**Fig. 6**—Predicted maximum temperature rise above ambient ($tg_{max} - Ta$) as a function of pitch line velocity (V) with two different loads, three materials, all: 8 pitch (3.175 module)

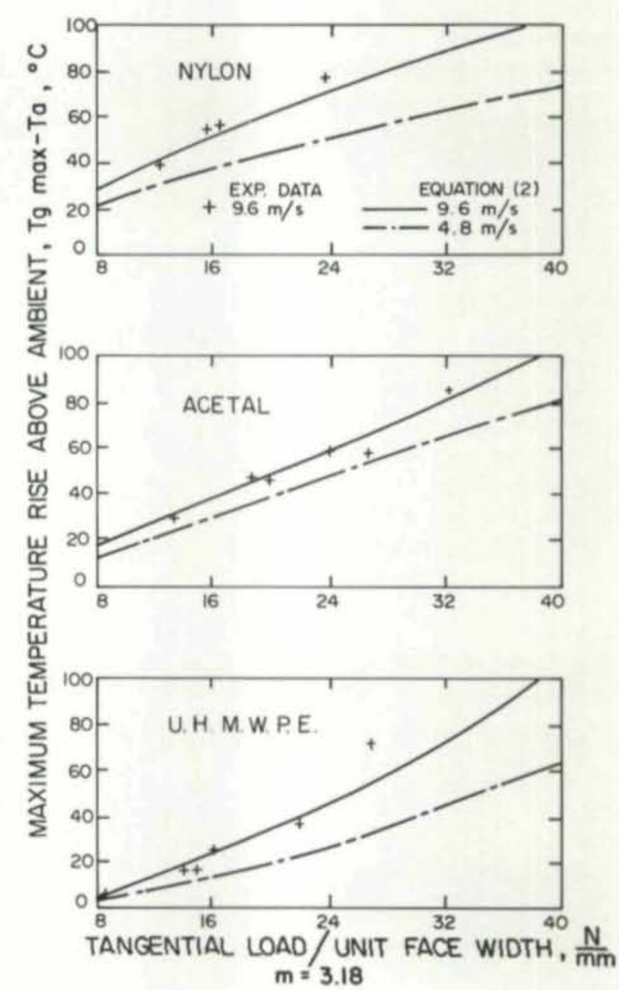


**Fig. 7**—Predicted Maximum temperature rise above ambient ($Tg_{max} - Ta$) as a function of the tangential load per unit face width (Wt) for two different speeds. Three materials, all: 8 pitch (3.175 module)

tried, the following gives the best fit:

$$(T_g \max - T_a) = b_0 W_t^{b_1} V^{b_2} m^{b_3} \qquad (2)$$

where $T_g$ max $T_a$ are expressed in °C and $b_0$, $b_1$, $b_2$, $b_b$ are regression coefficients calculated for each material and given in table 2.

Using the values of regression coefficients from table 2 and equation (2), curves are drawn on Figs. 6, 7 and 8 for the three materials studied and typical operating parameters. Some of the experimental results, for which the operating parameters were similar to those chosen to draw the curves, are given for reference purposes.

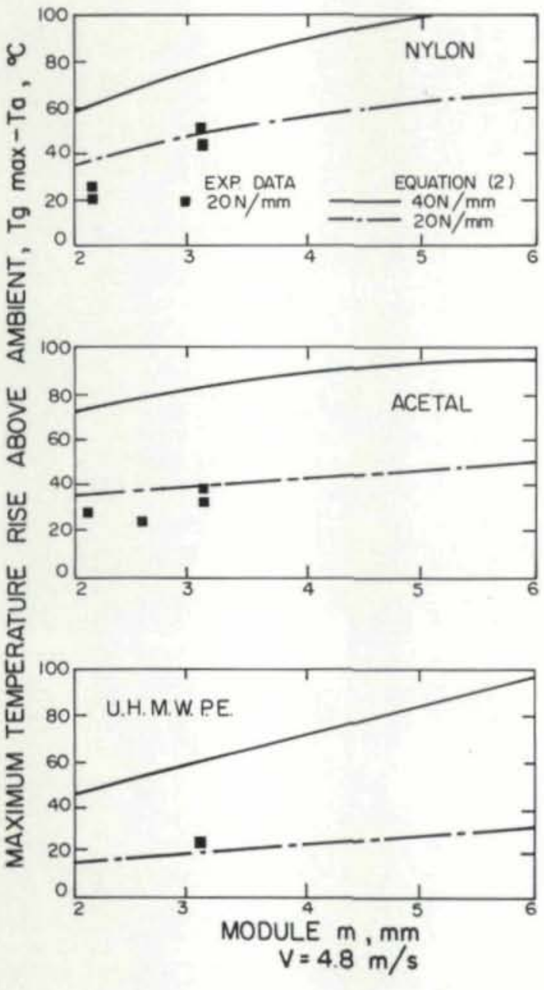


Fig. 8—Predicted Maximum temperature rise above ambient ($Tg_{max} - Ta$) as a function of the module m for two different loads. Three materials, all 4.8 m/s pitch line velocity.

As expected, the temperature increases with increasing velocity, load and module (coarser). However, their respective influence varies from one material to another and this reflects on values of the coefficients in table 2. While an exponent smaller than 1.0 produces a convex curve, Figs. 6 and 8, an exponent greater than one produces a concave curve. This is illustrated by comparing curves for UHMWPE to these for acetal or nylon in Fig. 7. The fact that UHMWPE behaves differently than the two other materials might be related to its peculiar properties. As a matter of fact, nylon and acetal are viscous liquids above their melting points, while UHMWPE is so high in molecular weight that it does not really melt and behaves more like a rubber at elevated temperatures. Since the maximum temperature generated by friction along the tooth profile can exceed the melting temperature of the material, a reduction in the friction coefficient for nylon and acetal can be expected, because of the viscous liquid generated which moderates the rate of increase in temperature. But no such liquid is generated for UHMWPE, and temperature rises more and more steeply with load with a corresponding increase in tooth flexibility and degradation of meshing qualities.

It must be understood that equation (2) is derived for a certain range of operating parameters which can be obtained from the various axis of Figs. 6 to 8. They are: pitch

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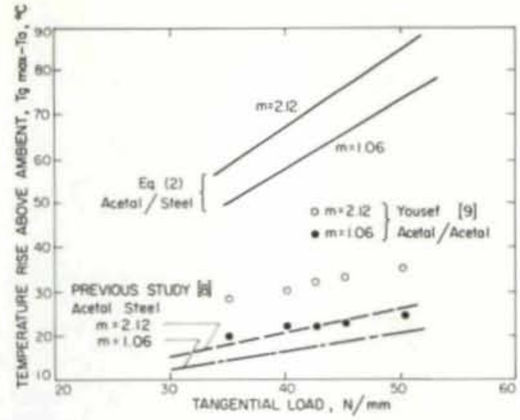


Fig. 9—Comparison of mean tooth surface temperature from ref. [8] and [9] with maximum tooth surface temperature obtained with eq (2).

line velocity from 2 to 10 m/sec, tangential load 8 to 40 N/mm, and modules 2.11, 2.54, 3.175 and 5.08 mm. Even though there is no reason to believe that the general behavior of the parameter would change outside these limits, caution must be takin in extending the use of equation (2).

*Youssef et al.*[9] *measured the mean tooth surface* temperature of running gears with an infrared microscope for an acetal/acetal contact. Their results, along with those of a previous study[8] for mean tooth surface temperature of an acetal/steel contact, are compared in Fig. 9, with the maximum tooth surface temperature, as calculated by equation (2) for an acetal/steel contact. Comparable values for mean tooth surface temperature are obtained from both references, even though one is for acetal/acetal, while the other one, which gives the lower values as expected, is for acetal/steel contact. As one can see, however, maximum tooth surface temperature can be considerably higher than mean tooth surface temperature.

## Conclusion

The infrared radiometer permits to measure the temperature peaks on the profile of running gears. A considerable amount of data was collected and analyzed by computer to come up with an equation to predict the maximum tooth surface temperature as a function of the operating parameters.

The results show the influence of the material, as well as that of tangential load, speed, and gear module. By comparison with published data for mean tooth surface temperature, it is seen that the maximum temperature on the tooth surface might be considerably higher than the mean surface temperature, which is usually considered in gear analysis.

**Nomenclature**

| | |
|---|---|
| $b_0, b_1, b_2, b_3$ | regression coefficients to use in equation (2) |
| b | gear face width, mm (inch) |
| $d_p$ | pitch diameter, mm (inch) |
| $E'$ | storage modulus of a viscoelastic material, MPa (lb/inch$^2$) |
| m | module, mm |
| P | pitch, (diametral) |
| $p_b$ | normal base pitch of a spur gear, mm (inch) |
| $\tan(\partial)$ | hysteresis loss factor for a viscoelastic material |
| $T_a$ | ambient temperature, °C (°F) |
| $T_{gm}$ | mean temperature on the surface of the plastic tooth, °C (°F) |
| $T_{g\,max}$ | maximum temperature on the surface of the plastic tooth, °C (°F) |
| V | pitch line velocity, m/sec (ft/min) |
| $W_t$ | tangential load per unit of face width, N/mm (lb/inch) |
| $_tY$ | Lewis tooth form factor determined with load applied near the middle of the tooth |
| $Z_1, Z_2$ | number of teeth in pinion (driver) and driven gear respectively |
| $\alpha$ | cutting pressure angle deg. |
| $\epsilon$ | plastic gear material emissivity |
| $\mu$ | coefficient of friction between contacting profiles |
| $\omega$ | angular frequency of loading |
| $\sigma_o$ | maximum stress amplitude |

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## *MAXIMUM SURFACE TEMPERATURE . . .*

*(Continued from page 27)*

TABLE 1

GEAR GEOMETRY

| | PINION (DRIVER) PLASTIC(1) | DRIVEN GEAR STEEL |
|---|---|---|
| module, m, mm (Pitch, P, diametral) | 5, 8, 10, 12 (5.08, 3.175, 2.54, 2.116) | |
| Pitch diameter, $d_p$ mm (inch) | 3.0 (76.2) | 3, 4.5(2) (76.2, 114.3) |
| Number of teeth $Z_{1,2}$ | 15, 24, 30, 36 | 15, 24, 30, 36, 45(2) |
| Pressure angle $\alpha$, deg | 20 | |
| Face width, b mm (inch) | 0.5 (12.7) | 1.0 (25.4) |
| AGMA quality number | 5 to 8 | 11 |

TABLE 2

VALUE OF COEFFICIENTS USED IN EQ (2)

| MATERIAL | $b_0$ | $b_1$ | $b_2$ | $b_3$ |
|---|---|---|---|---|
| nylon 6-6 | .2354 | .755 | .420 | .502 |
| acetal | $5.556 \times 10^{-2}$ | 1.08 | .354 | .225 |
| U.H.M.W.P.E. | $1.985 \times 10^{-4}$ | 1.76 | .831 | .687 |

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CIRCLE E-3 ON READER REPLY CARD

*This article was reprinted from the American Gear Manufacturers Association.*

**CORRECTION:**

**ROBERT E. SMITH's** first name was listed incorrectly on the content page of the last issue. Mr. Smith was the author of "Single Flank Testing of Gears."