Calibration of Two-Flank Roll Testers

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Management Summary

The presence of significant errors in the two-flank roll test (a work gear rolled in tight mesh against a master gear) is well-known, but generally overlooked. Of the various sources of error, one is the attenuated response of a two-flank roll tester to gear defects. In this article, it will be shown how to design and inspect a device for measuring this particular error.

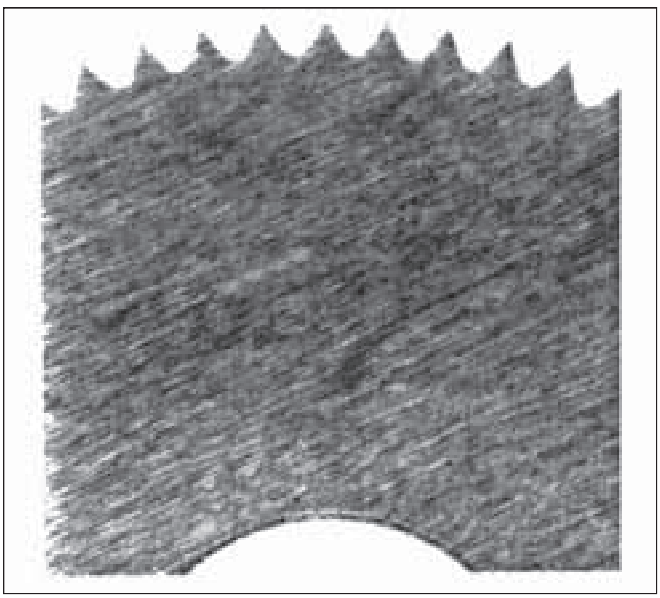

Figure 1—Pointed-tooth gear.

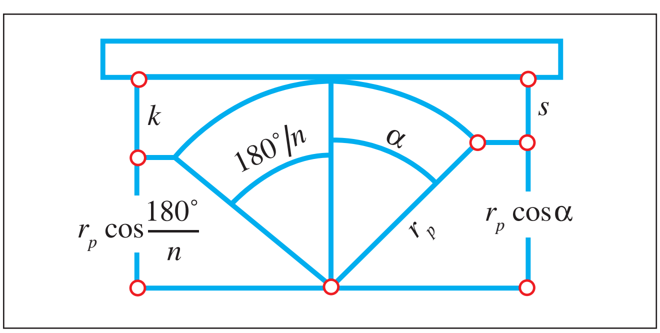

Figure 2—Calculation of pointed-radius and carriage displacement.

Introduction

Historical evidence shows that measured values for kickout (also known as tooth-to-tooth composite error; terminology rarely used by shop personnel) on the same gear can differ from one two-flank roll test to another. For example, Michalec and Karsch conducted a correlation study (Ref. 1) wherein an assortment of 100 fine-pitch precision gears was inspected at 20 different facilities for total composite error, tooth-to-tooth composite error (kickout) and testing radius. In their final report, in addition to finding "a wide variation of measurements among companies," they "decided to eliminate" the study of kickout, because "the readings contained considerable uncertainty."

If a similar correlation study were to be conducted today, the discrepancies probably would be similar, since there has been no marked improvements in inspection practice and test equipment. This article deals with a method of calibrating the response of two-flank roll testers to a known kickout.

Basic Geometry

One way to generate a known kickout is with a pointed-tooth gear (Fig. 1), where the points contact a flat (not an arbor) mounted on the floating carriage of the roll tester.

From Figure 2 it is seen that, for a given kickout (k) and tooth number (n), the pointed-radius (r_p) is:

$$k + r_p \cos \frac{180°}{n} = r_p \qquad (1)$$

so that…

$$r_p = \frac{k}{1 - \cos \frac{180°}{n}} \qquad (2)$$

Another way to generate a known kickout is with an eccentric disc. However, the carriage motion would be sinusoidal, unlike the abrupt reversals seen in practice.

Likewise, for a specified pointed radius (r_p) and rotation angle (α),

$$r_p \cos\alpha + s = r_p \quad (3)$$

so that the carriage displacement (s) is:

$$s = r_p\,(1-\cos\alpha) \quad (4)$$

From Figure 3 (a plot of Eq. 1), it is seen that the pointed radius (r_p) increases with both kickout (k) and tooth number (n). And from Figure 4 (a plot of Eq. 2 for n = 72 and k = 0.0008"), the carriage undergoes an abrupt reversal at $\alpha = 180°/n$.

From Figure 5 it is seen that the tooth thickness (t) that corresponds to a specified pointed radius (r_p) is

$$\text{inv}\varphi_p = \text{inv}\Phi + \frac{t/2}{r} \quad (5)$$

where…

$$t = \frac{\pi}{2P} + \Delta t \quad (6)$$

and…

$$r = \frac{n}{2P} \quad (7)$$

P is the diametral pitch and Φ is the profile angle, so that…

$$\Delta t = \frac{\left[(\text{inv}\varphi_p - \text{inv}\Phi)n - \frac{\pi}{2}\right]}{P} \quad (8)$$

where…

$$r_p \cos\varphi_p = r_b = \frac{n}{2P}\cos\Phi \quad (9)$$

so that…

$$\cos\varphi_p = \frac{n}{2P}\,\frac{\cos\Phi}{r_p} \quad (10)$$

Also, from Figure 6 it is seen that the clearance (c) between the pointed tip and the root of a topping hob is

$$r + \Delta C + \frac{1}{P} = r_p + c \quad (11)$$

so that …

$$c = \frac{n+2}{2P} + \Delta C - r_p \quad (12)$$

where, from the well-known equation for a rack (Ref. 2),

$$\Delta C = \frac{\Delta t}{2\tan\Phi} \quad (13)$$

Numerical Example

A pointed-tooth gear (Fig. 1) can be inspected with a micrometer by measuring over diametrically opposite tooth

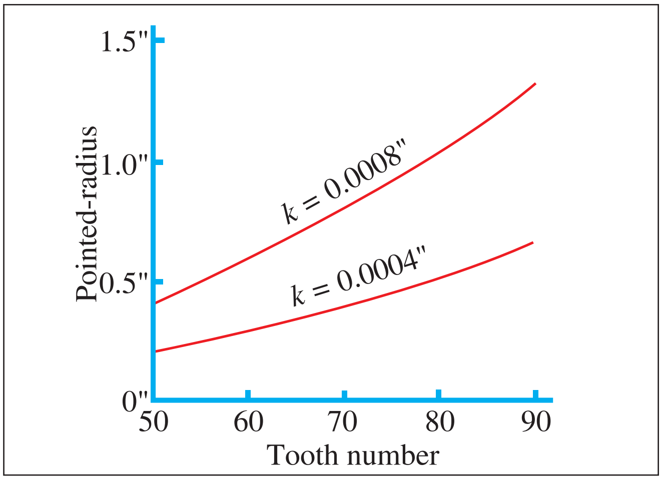

Figure 3—Pointed-radius versus kickout and tooth number.

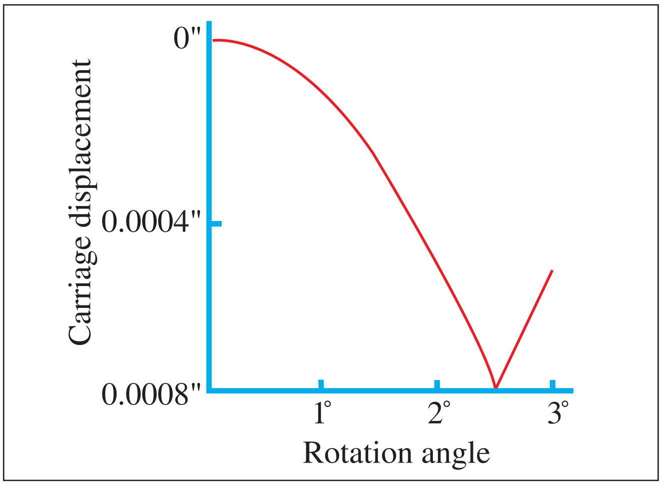

Figure 4—Carriage displacement versus rotation angle.

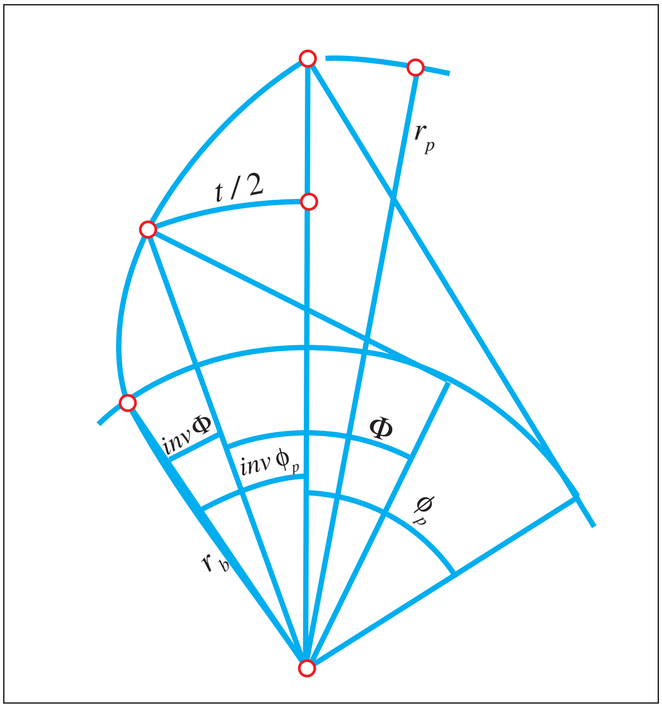

Figure 5—Calculation of tooth thickness for a specified pointed-radius.

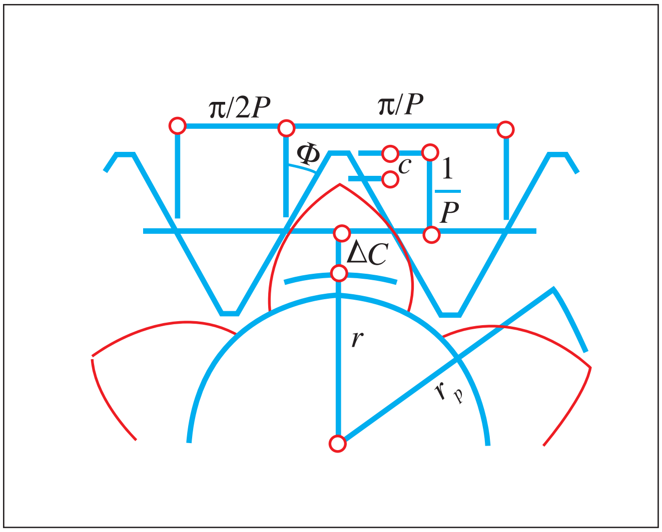

Figure 6—Calculation of clearance between pointed tip and root of topping hob.

Table 1—Calculated Data For Tip Clearance					
n	r_p	φ_p	Δt	ΔC	c
70	0.7945"	30.4106°	0.02747"	0.0377"	-0.007"
72	0.8405"	33.0167°	0.05528"	0.0760"	0.006"
74	0.8879"	35.3337°	0.08649"	0.1188"	0.023"

spaces (contact on four tooth tips, which requires an even tooth number), and over diametrically opposite tooth tips, the difference being double the kickout.

For a pointed-tooth gear of about 1.5 inches diameter (for ease of turning by hand) and a 48*P*, 20° topping hob, feasible tooth numbers are: 70, 72 and 74.

For k = 0.0008", the values for r_p (Eq. 1), φ_p (Eq. 4), Δt (Eq. 3), ΔC (Eq. 6) and c (Eq. 5) are tabulated in Table 1, wherein it is seen that the 70-tooth gear is topped, and that either the 72- or 74-tooth gears are applicable.

When hobbing a pointed-tooth gear, it is convenient to obtain size via a span measurement. For the 72-tooth gear, the span dimension over 14 teeth is 0.9032" (Ref. 3), which can be measured with an ordinary micrometer.

Conclusion

It should be noted that widespread calibration of two-flank roll testers is not likely to occur in the near future. For example, Darle Dudley was asked if "gear development has been evolutionary or revolutionary." He replied (Ref. 4):

"I think gearing evolves. Historically, 10 to 40 years may pass from when a new idea is first heard of to when it becomes an established trade practice."

Likewise, the National Bureau of Standards set up a Gear Metrology Laboratory (Refs. 5 and 6) in 1961, only to terminate it in 1970 due to lack of demand. (Editor's note: The national gear metrology lab was reborn in 1994. See our article on page 62 for details.)

Also, for more than 50 years it has been known that measured values for kickout and tooth thickness can vary with the number of teeth on the master gear (Refs. 7 and 8), a phenomenon ignored by one and all.

And George Grant, inventor of the hobbing machine (Ref. 9), encountered apathy during the transition from cycloids to involutes. He wrote (Ref. 10):

"There is no more need of two different kinds of tooth curves for gears of the same pitch than there is need for two different threads for standard screws, or two different coins of the same value. The cycloidal tooth would never be missed if it were dropped altogether. But it was first in the field, is simple in theory, has the recommendation of many well-meaning teachers and holds its position by means of 'human inertia,' or the natural reluctance of the average human mind to adopt a change, particularly a change for the better."

References

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