$$\tau_{cr}(\bar{p}) = \frac{-8.9256+\sqrt{(-2.4456)^2+1.2\times 0.2274\times 10^{-6}\times 0.783\times 10^9}}{2\times 0.2274\times 10^{-6}}$$

$$= 12.962\times 10^6$$

Eqn (17)

$$\mu_{hs} = \frac{12.962\times 10^6}{0.783\times 10^9} = 0.0165$$

μ_{ls} is *NOT* $< \mu_{hs}$

Thus

$$\mu = \frac{\mu_{ls}+\mu_{hs}}{2} = \frac{0.0607+0.0165}{2} = 0.0386$$

Using $\mu_1 = 0.0386$ $F_{R1} = 2.46$, equation (6) can be used again to calculate an improved value of N_1

$$N_1 = \frac{1000+0.0572\times 2.46}{0.09397-0.0572\times 0.0386} = 10{,}899\text{N}$$

and the whole calculation can be made again. For the purposes of illustration the values already calculated are used to determine efficiency

Eqn (5)

$$M_I = 10{,}899 \times 0.09397-0.0112(0.0386 \times 10{,}899 -2.46)$$

$$= 1019.5 \text{ Nm}$$

$$E_{inst} = \frac{1000}{1019.5} = 98.09\%$$

Discussion

W. Coleman[1]. This is a very interesting paper relating to an area of gear design which is most pertinent at the present time. The subject of gear efficiency is very complicated and therefore difficult of analysis. The author here has attempted to set down a simplified method of approach to this analysis. But as stated in his discussion at the end of the paper he points out that several of the assumptions are artificial and therefore actual gears in service may not obey the analysis given here.

In a paper [A] by the discusser the results of tests on a Geared Roller Test Machine are plotted showing the relationship between load and average coefficient of friction at several different sliding velocities. These test confirm the fact that at the lower loads the coefficient of friction increases with an increase in load. But at some load level this relationship reverses. The reason for this was not clear. These tests were terminated at the load level when scuffing occurred.

Since efficiency is defined as the ratio of the ouptut load to the input load, it follows that the efficiency will be zero at no load. This paper infers that the efficiency will be at a maximum at no load. Does the author care to comment on this.

A Coleman, H., Bevel and Hypoid Gear Surface Durability-Pitting and Scuffing Lubrication and Wear Conference, Institution of Mechanical Engineers, London (Sept. 1967), 10/18/79.

Author's Closure

I am grateful to Mr. Coleman for drawing my attention to his paper [A]. The difficulty in answering his points is whether we are comparing the same things. My paper is concerned solely with the EHD lubrication regime and the equations used are limited to the range of maximum Hertz pressures of 0.5 - 2.0 GN/m^2, the computed results and conclusions thus only apply to heavily loaded gears. Also no account was taken of any other losses than those due to frictional effects between the teeth. Because the width of the discs is not given in [A] the maximum Hertz pressure cannot be calculated but it is reasonable to assume that the tests in [A] did pass through the range 0.5 - 2.0 GN/m^2.

Since Mr. Coleman states that his tests terminated when scuffing occurred this implies that at that stage the lubrication regime was not EHD but was very much into the mixed regime; his results suggest that his tests started (at low loads) in a regime of hydrodynamic lubrication and (as the load was increased) passed through the EHD regime and completed the tests in the mixed regime. Thus only parts of his test results can be compared with the predictions of the present paper.

The effect of load in the transition from EHD to mixed lubrication is very complicated and the present paper does not attempt to make any prediction in this area. However there is no doubt that the results of (1) do show that at high loads the coefficient of friction decreases as load increases and there is no theoretical support for this if the regime is considered as EHD; if the lubrication regime is mixed then there are few aids at the moment to allow prediction. I would be interested to know how Mr. Coleman measured his coefficient of friction - did this only account for friction between the gear teeth? Other losses are also of course, load sensitive and these could affect the situation.

I am not clear as to the meaning of Mr. Coleman's statement, "Since the efficiency is defined as the ratio of the output load to the input load, it follows that the efficiency will be zero at no load." I presume it is implied that with a real gear train there will be losses, and therefore zero efficiency, at no load and with this I agree. The equations used in this paper do not apply to zero load conditions but from EHD theory the coefficient of friction at zero load is zero so that *if all other losses were ignored* the efficiency will be predicted as 100%. A better estimate of efficiency in this case would be to use [B].

B Anderson N.E., and Loewenthal S.H., "Effect of Geometry and Operating Conditions on Spur Gear System Power Loss," Century 2, *International Power Transmissions and Gearing Conf.*, San Francisco, Aug. 18/21 1980, Paper No. 80-C2/DET-39.

[1] Gear Consultant.

Copyright © 1981 by ASME
Check for updates
Downloaded from http://electronicpackaging.asmedigitalcollection.asme.org/mechanicaldesign/article-pdf/103/1/169/5808914/169_1.pdf by guest on 10 November 2024