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A rolling-sliding bench test for investigating rear axle lubrication

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ABSTRACT

An automotive rear axle is composed of a set of hypoid gears, whose contact surfaces experience a complex combination of rolling contact fatigue damage and sliding wear. Full-scale rear axle dynamometer tests are used in the industry for efficiency and durability assessment. This study developed a bench-scale rolling-sliding test protocol by simulating the contact pressure, oil temperature, and lubrication regime experienced in a dynamometer duty cycle test. Initial bench results have demonstrated the ability of generating both rolling contact-induced micropitting and sliding wear and the feasibility of investigating the impact of slide-to-roll ratio, surface roughness, test duration, and oil temperature on the friction behavior, vibration noise, and surface damage. This bench test will allow studying candidate rear axle lubricants and materials under relevant conditions.

1. Introduction

The brothers Jacobus and Hendrik-Jan Spijker introduced the first four-wheel drive car using an internal combustion engine in 1903 [1]. Due to their superior performance on rough terrain, four-wheel drive vehicles were primarily used by militaries until Willys introduced the model CJ-2A to the general public in 1945 [2]. Since then, all wheel drive cars have continued to gain popularity especially in the past few decades as they enable more control in severe weather [3].

The rear axle itself is comprised of a set of hypoid gears, first introduced by Nikola Trbojevich in 1923 [4], which have been the staple for the drive mechanism of automobiles since 1931. The hypoid gear system is a type of spiral bevel gear, which has an offset between the axes of rotation offering several advantages [5], which allows designers to use this offset to lower the vehicle center of gravity. For hypoid gears, even the pitch line has nonzero sliding. Unlike spur gears that are designed to have rather low slide to roll ratios (SRR, as defined by $SRR = \frac{u_1 - u_2}{(u_1 + u_2)/2}$, where u_1 and u_2 represent the two gear teeth's surface speeds at the contact, respectively), hypoid gears may experience SRRs above 100% (out of the full 200% scale). Due to its complexity, there are limited literature studies on hypoid gears and even fewer on rear axles. One paper reported the SRRs for an actual rear axle system in a range from roughly 1.4% to above 36% at a given contact position between the pinion and ring [6].

The power losses of the automotive axle include both load-dependent (mechanical) and load-independent (spin) loss components, which depend on the operating conditions, contact surface roughness, and bearing preload [7]. The load-dependent losses are primarily caused by friction between gear contacts and the load-independent losses are largely related to the viscous effects. It has been measured that the hypoid gear axle losses are proportional to the oil viscosity [8]. The lubricants for hypoid gearboxes have historically been rather viscous as a way to protect the gear surface under the high contact pressure. Additionally, most passenger cars' rear axle fluids are filled for life so engineers have erred on the side of caution [9]. Recently, there has been an increased interest in improving the fuel economy via new system design, lubricant, and/or surface treatment. One such study in 2002 showed that changing the rear axle fluid viscosity grade from SAE 75W-140 to SAE 75W-90 offered up to 1.4% increase in fuel economy [10]. Tribological testing is critical in lubricant and gear material development and full-scale rear axle dynamometer testing commonly is used in the industry, which is time consuming and costly. To the best of our knowledge, there is no standard bench scale tribological test available for evaluating real axle fluids.

In this study, we attempted to establish a bench-scale rolling-sliding test protocol based on test conditions in a rear axle dynamometer duty cycle test. Initial results generated by this bench test have demonstrated the ability of investigating the impact of SRR, contact stress, oil temperature, and surface roughness on the friction behavior, vibration noise, and surface damage. Worn surface morphology generated in the bench test was correlated to that from the rear axle dynamometer test as well.

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2. Experimental and materials

2.1. An actual set of rear axle pinon and ring

A pinon and ring set from an actual rear axle system of an all-wheel drive SUV was ordered from Auto-Wares, Inc. (see Fig. 1a). This set of pinion and ring are similar to what are used in a dynamometer test, whose testing conditions were used to establish the bench test in this study. Both components are made of carburized case-hardened steel with a top surface layer of manganese-phosphate. After polishing off the phosphate film, the Vickers microindentation (at 100 g-f) hardness of the pinon and ring teeth was measured to be 811 and 829 HV, respectively. In order to determine the contact geometry and surface roughness, gear teeth were cut off both the pinion and ring using CNC programmed wire electrical discharge machining (EDM) (see Fig. 1b).

2.2. Tribological bench test

A micro-pitting rig (MPR, by PCS Instruments) was used to establish a tribological bench test to study the friction, surface damage, and vibration of the rolling-sliding contact interface under similar conditions between the pinion-ring contact in a rear axle dynamometer test. The MPR, as shown in Fig. 2, uses three rings rolling-sliding against a roller at the center at controlled load, SRR, and oil temperature. Both the rings and the roller are made of hardened AISI 52100 bearing steel. The rings are straight discs with a nominal diameter of 54 mm and thickness of 8 mm. The roller (see Fig. 2 right) has a central narrow cylindrical band with a nominal diameter of 12 mm and a nominal width of 1 mm. The cylindrical portion of the roller rolls and slides against the three rings simultaneously during the test with a contact width of roughly 1 mm. The Vickers microindentation (at 100 g-f) hardness of the ring and roller are 770 and 602 HV, respectively. The roughness is about 0.40 μm (R_a) for both the roller and ring contact surfaces in a regular set. Smoother rings (0.15 μm, R_a) were also used to rub against a roller with regular roughness (0.40 μm, R_a) in tests to study the impact of ring roughness on friction, vibration, and surface damage of the roller. The friction coefficient was converted from in situ measured torque and the vibration behavior was quantified by a unitless acceleration signal. In an MPR test, because the diameter of the ring is five times as large as that of the roller and the roller rubs against three rings in each revolution, the number of cycles of contact experienced by the roller is 15 times what experienced by the ring at pure rolling.

A commercial fully-formulated SAE 75W-90 rear axle fluid was used as the lubricant. The oil viscosities were measured at various temperatures using a MINIVIS II viscometer as shown in Table 1.

2.3. Lubrication regime calculation

The lubrication regimes of the pinion-ring interface in an actual rear axle and the ring-roller contact in the MPR test were determined by computing the ratio (λ) between the lubricant film thickness and the composite roughness at the contact area: $\lambda = h/\sigma$. There are three common lubrication regimes [9]: boundary (BL, $\lambda < 1$), mixed (ML, $1 \le \lambda < 3$), and elastohydrodynamic/hydrodynamic (EHL/HL, $\lambda \ge 3$).

The lubricant film thickness for the MPR bench test was calculated by using the Hamrock and Dowson line contact formula [9] as detailed in S1 in the Supporting Information. The following assumptions were made when deriving the formula in Ref. [9]: fluid is incompressible, fluid behaves linearly (Newtonian), can be treated as laminar flow, no-slip condition applies, fluid is isotropic, and the only variable that is a function of the z position (z axis points between the two surfaces in contact) is the speed. At high shear and contact pressures, several of these assumptions may break down, which poses a problem for applying this model to an actual rear axle where both speed and pressure are known to be high. This equation is already known to be up to 40% off the actual fluid film thickness when the non-dimensional load is an order of magnitude higher than the simulated conditions used for deriving these equations [9]. Fortunately, a fix has already been made to account for non-Newtonian behavior of fluids in the literature [12]. Equation (1) describes a non-Newtonian correction factor φ such that $h_{nn} = \frac{h_n}{\varphi}$, where h_{nn} is the fluid film thickness after non-Newtonian behavior is accounted for, h_n is the Newtonian fluid film thickness calculated from the Hamrock and Dowson formula.

$$\varphi = \left[1 + 4.44\left(\frac{u\mu_0}{h_n\tau_c}\right)^{1.69}\right]^{1.26\left(1-\frac{1}{2-m}\right)^{1.78}} \quad (1)$$

where

u – speed.
μ_0 – viscosity at atmospheric pressure.
h_n – Newtonian fluid minimum film thickness.
τ_c – critical shear stress.
m – non newtonian behavioral constant.

The new parameters introduced, τ_c and m, can be calculated by fitting the equation below to real data. Note that the non-Newtonian correction factor must be greater than or equal to 1, meaning this model will only take into account shear-thinning behavior. The equation $\mu =$

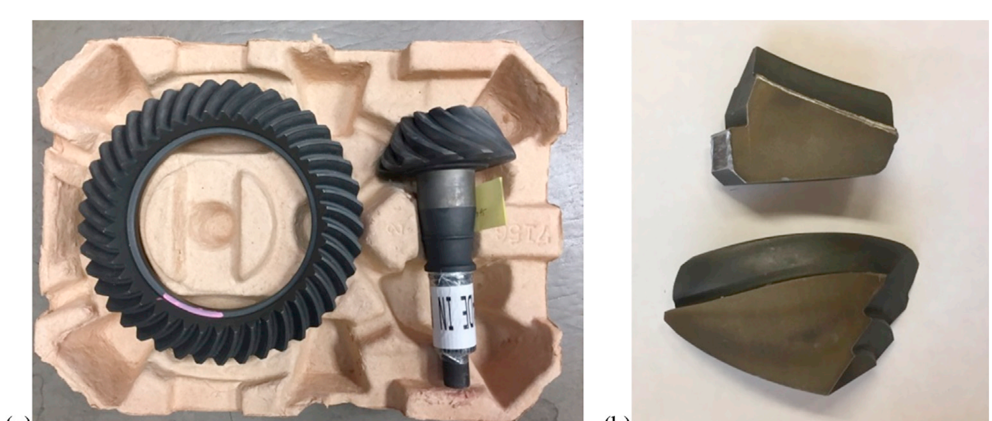

Fig. 1. A pinion and ring set from an actual rear axle system. (a) As received, (b) Individual pinion (bottom) and ring (top) teeth cut off by wire EDM.