DUDE! — WHERE'S MY PRELOAD?

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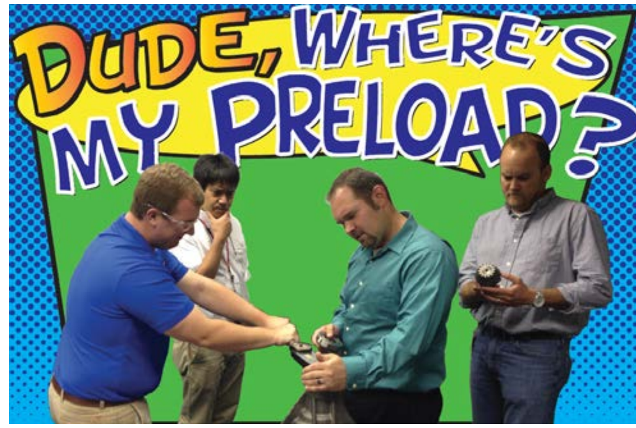

Introduction

Many of us have been there; the bearings *had* the correct preload. You know it, you were there, and you personally saw the measurements. Now, the testing is done and the preload is gone. Not a little gone, not sort of gone—*gone*, gone. Finger pointing ensues. Suppliers are dragged in by their wrinkly Polo collars. You know the drill. Losing preload in a tapered roller bearing (TRB) system over the life of your application can be a troublesome problem, particularly for gear sets that are prone to noise or severe applications that rely on a very rigid and stable system.

Two items that are often confused is the drop-in initial running torque vs. true loss of preload. Both will occur in your application, but it is important to understand the differences. Before we get too far into the details, we will review exactly where the drop-in torque and loss of preload originate.

Unlike ball bearings, there is a true sliding surface in a TRB between the bottom of the roller and the large rib. This interface (Photo 1) is also responsible for maintaining preload by physically supporting the roller position. We will learn exactly how to analyze and quantify this in a later section.

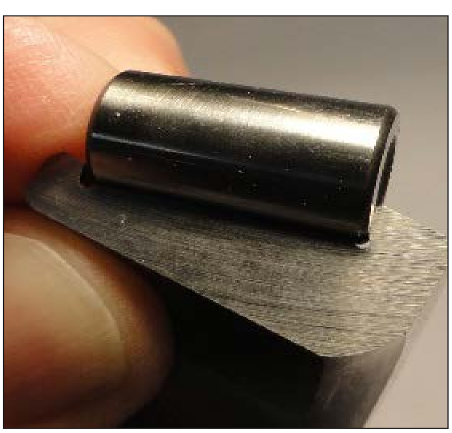

Photo 1 TRB cross-section highlighting rib/roller interface.

Loss of Rolling Torque

First, let's discuss the loss of rolling torque. It is very typical to lose, perhaps, half of the bearing rolling torque in a very short period of time after startup. When the bearings are new, the rib and roller surfaces are as rough as they are ever going to be. As soon as the bearing starts rolling, these surfaces immediately begin to polish each other. This period, often thought of as the "break-in" period, is nothing more than the surfaces of the roller ends and large rib wearing in together. This process produces almost no measurable wear, 1-2 µm at most. The bearings run hotter during this period due to the higher friction between the surfaces, which is why there is usually a 500 mile break-in recommendation for new vehicles and rebuilt axles prior to towing or heavy usage. A break-in period allows for the bearings to wear in without getting too hot (Fig. 1). A hot running bearing can damage the oil, which produces loss in viscosity and additives, which is even harder on the bearing. This is a downward cycle you want to avoid. If you feel like you may have abused a new axle, the bearings were *probably* not damaged (yet); a quick axle lube change can usually get you back on track.

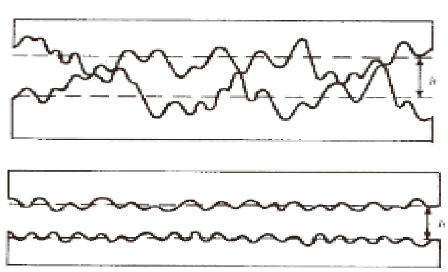

Figure 1 New bearing vs. broken-in bearing.

Of course there are numerous housing deflection issues that can be a source of losing preload. Before you begin to dig into your bearing system, make sure the housing is not playing a role. We are dealing with microns of deflection, so this is not something you are going to visually see happening.

Loss of Preload

Every single tapered roller bearing that is not made from a frictionless, massless material will wear between the roller ends and large rib beyond the initial break-in period. This is unavoidable because there is a true sliding surface between the rollers and large rib, and this wear is responsible for preload

Photo 2 TRB run with no lubrication.

loss. Photo 2 shows a TRB that was intentionally run at a low speed with no lubrication that really captures the major source of friction. The area outside of the rib interface remained undamaged while the rib and roller began to friction-weld together.

Our goal is to try and understand, mitigate and compensate for the anticipated wear. There does appear to be a point at which the system stabilizes and wear slows down considerably. It is thought that this is a combination of components fully wearing in together plus some potential work hardening occurring at the surfaces. While this effect is not yet completely understood, it has been observed that most of the wear occurs within the first 5,000 miles — or roughly 100 hours of usage (Fig. 2).

You can bury yourself trying to understand and predict lubrication flow, EHL and thermal equations — which are an integral part of the bearing function — but aren't incredibly useful in talking about wear. If we are just trying to understand the mechanics of bearing wear, the starting torque equations are a great place to begin. Where there is torque, there is sliding friction, and where there is sliding friction, there is wear.

$$M = \frac{e\mu_e l \sin(2\beta)}{D_{wl\,\sin(\alpha)}} F_a \tag{1}$$

Since $D_{wl} = 2\overline{OB}\sin\beta$ and $l = \overline{OB}\sin\alpha$

$$M = e\mu_e \cos\beta F_a \tag{2}$$

Let's walk through this fairly intuitive formula; as the contact point e between the roller and rib increases along with the outer diameter of the roller l, torque increases. The larger diameters increase sliding velocity, thus increasing torque/wear. As the roller taper angle β increases, torque also increases. One way to think of this is that there is a wedge created between the cup and cone with the roller in the middle; the steeper the wedge, the more force the roller wants to push out and into the rib. On the other side, as roller diameter D_{wl} increases, torque decreases. The larger the roller creates a smoother transition in and out of the rib contact area, reducing sliding velocity. Finally, as the bearing contact angle α increases, torque decreases. This is just a function of a force acting upon an inclined plane. As the net axial force on the bearing increases, the roller presses harder against the rib, increasing torque. If the raceway is steeper, proportionately more load will be pushed directly on the rolling part of the raceway rather than the rib.

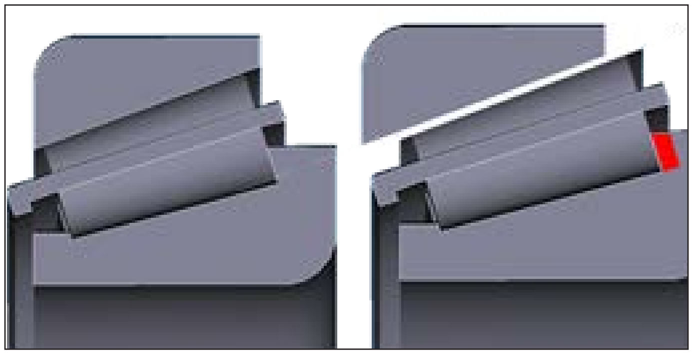

Figure 2 Schematic showing the physical effect of rib wear.

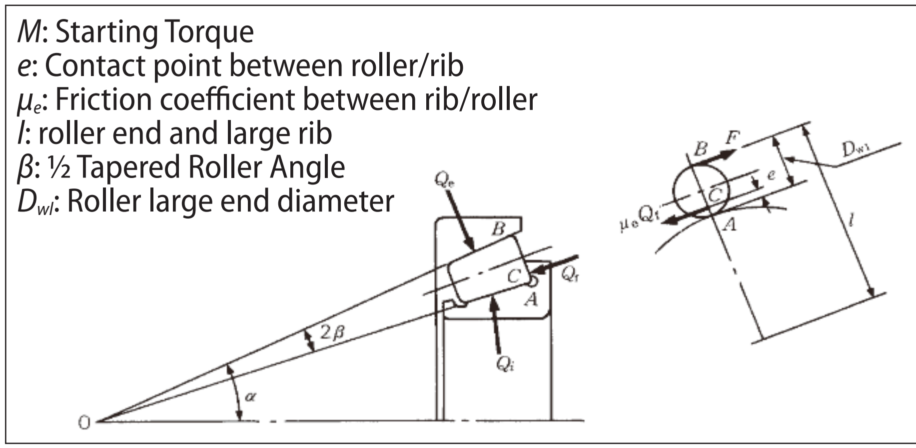

Figure 3 Starting torque equations.

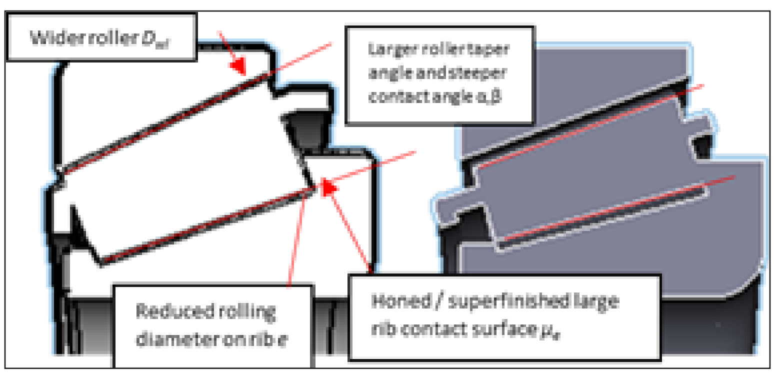

Figure 4 Modern high-efficiency bearing (L) vs. standard bearing (R).

Taking away notes from this formula, if we were to design a more efficient / low-wear bearing, we would:

- Make the contact point e of the roller as close to the inner diameter as possible
- Increase the roller diameter D_{wl}
- Increase the taper angle of the roller β
- Increase the contact angle of the bearing α
- Improve surface finish and/or hardness at the rib/roller interface to reduce μ_e

If you made these guesses based on the formula, you just designed a modern, high-efficiency bearing that is currently in tens of millions of vehicles.

For various reasons, we may not want all of these features in the bearing. A simple, effective improvement for TRB efficiency is just super-finishing or honing the large rib. Rib honing was quite difficult a decade ago, but improvements in manufacturing technology along with increased popularity of honing have made this modification accessible to most bearing manufacturers. Many companies now provide this as a standard feature. Of course this only provides us with the main driving mechanisms for wear at the bearing geometry level. Other significant factors are surface hardness, lubrication type, temperature, contamination, misalignment, etc. We can consider most of those noise factors that can be controlled to some extent.

Measuring and Quantifying

Now that we have established a basic understanding of the mechanics of losing preload, we will move on to measuring and quantifying. Being able to measure the rib directly is a great end-of-test feature to record, but we don't have an A/B to compare with because measuring the rib directly on a new part is not possible for an assembled bearing. A very simple method of directly measuring bearing wear is to measure the stand height as new and then again post-test (Fig. 5). The bearing companies will sometimes do this for you, and there are some inspection houses that will take these measurements if you don't have in-house capability. In a primary drive axle, you can expect head bearing losses in the 10-20 µm range (heavy loads), tail bearings around 5-10 µm (moderate loads), and differential bearings generally 5 µm or less (light/moderate loads). These are very general guides, but if you are seeing double or triple these values, you should have your lubrication analyzed to see if you are generating heavy wear particles from the gearset, or if your oil is breaking down for some other reason.

If you realize you have a preload loss

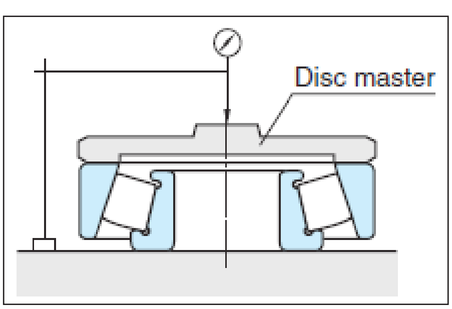

Figure 5 Measuring stand height.

issue after the fact and do not have the initial height measurements, simply measuring the bearing after test will serve little benefit. The stand height tolerance can be up to 200 µm for small TRB's, and we will only be looking for

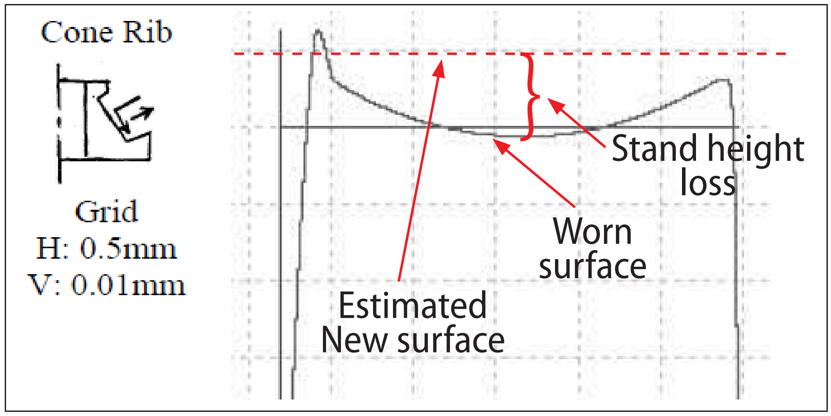

Figure 6 Rib height wear profile.

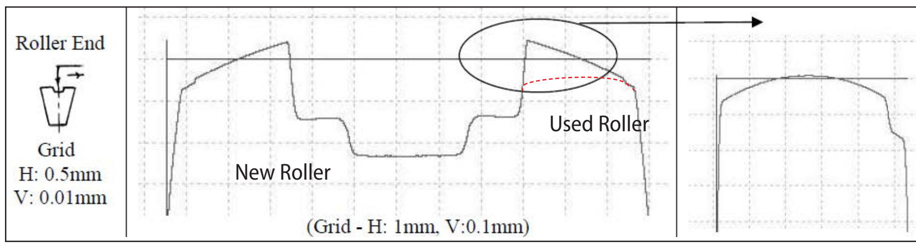

Figure 7 Roller end wear.

a loss of ~15 µm. All is not lost, though. We can still do a little forensic work with which to back up an educated guess. Due to the chamfers on the roller ends and the undercut of the cone where the large rib meets the raceway, there is a small area of the rib that doesn't wear very much. Using this area to project an estimated initial surface can be a decent estimate of the rib wear. In Figure 6 we see that comparing the inner area of the rib (the part that does not wear much) to the center of the rib, where the roller will settle, we see about a 13 µm drop. This is right on target for the values that had been seen for this particular application. For not knowing the initial stand height, this is a good estimate.

Similar measurements can be taken at the roller ends (Fig. 7), though these will usually only have 1-2 µm of wear at most, so invest your time and resources accordingly.

Now you have the pre- and post-test stand height measurements, or you had run traces for your tested bearings and you feel like you have some reasonable stand height loss numbers. Now what? Now we have to find the correlation between stand height loss and preload loss. The change in stand height will be our axial displacement and the loss in preload will be our axial load, . Now we find the fairly manageable relationship by:

$$\delta_a = K_a F_a^{0.9} \tag{3}$$

$$\text{Where } K_a = \frac{0.000077}{\sin \alpha^{1.9} z^{0.9} L_{we}^{0.8}} \tag{4}$$

And, α: Contact angle (½ cup angle) (deg)
L_{we} Effective contact roller length (mm)
F_a Axial Load (N)
Z Number of rollers

Most of these values can be found in catalogs or scaled from the models that are available on most bearing manufacturers' websites.

Let's try this calculation on a real application. A popular pinion bearing arrangement for light utility application is head bearing M802048/11 and tail bearing M88048/10. Let's look at a 5kN axial preload M802048/11 head bearing:

Plugging in values:

A 20°
Z 18
L_{we} 15.5 (mm)
F_a 5,000 (N)

$$\text{Where } K_a = \frac{0.000077}{\sin \alpha^{1.9} z^{0.9} L_{we}^{0.8}} = 4 \times 10^{-6}$$

Now calculating *Equation 3*,

$$\delta_a = 4 \times 10^{-6} \times 5000^{0.9} = 0.0091 \text{ mm}$$

The calculated deflection for M802048/11 with 5kN preload is 9.1 µm

To compare this result with a table calculated by a bearing manufacturer, see Table 1:

Table 1 Axial load vs. deflection

M802048/11	
Axial Load (kN)	Axial Deflection (μm)
1	2.24
2	4.19
3	6.03
4	7.81
5	9.55
6	11.3
7	12.9
8	14.6
9	16.2
10	17.8

Photo 3 Housing spreader tool to aid in installing differential bearings in a rear axle. When the housing is released, the bearings will be preloaded and the remaining housing tension will aid in maintaining preload as the bearings wear.

Not bad at all; our calculated value was within 5% of the manufacturer's calculated value. Also using manufacturer's data for the M88048/10, a 5kN preload will produce 10.6 μm of axial deflection for a total system deflection of ~ 20 μm.

Housing Elasticity

If we review our wear estimates of 10-20 μm for the head, and 5-10 μm for the tail, the situation begins to look grim; we could easily lose all of our 5kN preload just through expected wear. Fortunately, we have one other factor benefiting our system — the elasticity of the housing. If your company designs housings internally, very likely you can obtain an FEA analysis of the housing at the bearing journals to help you estimate the housing deflection in the preloaded state. If that is not available, a reasonable estimate can be had by assuming the bearings are roughly twice as stiff as the housing (true for most thin-wall housing applications). This means if you are axially displacing the bearings by 20 μm, you have to displace the housing by 40 μm in the process. After the preload is set, the total system deflection is a 60 μm.

With this added feature, we now see as we lose preload, the housing continues to spring back, helping to maintain preload. If we lose a total of 30 μm bearing height, using the housing / bearing ratio, we proportionately lose 10 μm of bearing load and make up the remainder with the housing springing back by 20 μm.

Just for simplicity: if we split the 10 μm between the 2 bearings, we have only *effectively* lost 5 μm worth of preload per bearing. If we review Table 1, we see that 5 μm of loss is roughly equal to 2.5 kN of preload, which is encouraging.

Even though we lost 30 μm of bearing stand height, we only lost approximately one-half of our preload with the aid of the housing providing some spring-back. After the system is dialed in through testing to see how much preload we expect to lose, it is a simple task to add 2 kN of preload in the initial setup to compensate for the expected wear. Testing has shown that marginally increasing the preload does not increase wear, so this is usually an effective compensation technique for battling preload loss.

Conclusion

All TRBs will have a break-in period, followed by a wear period.

Wear can be quantified by measuring new and tested/used bearings and increasing the initial preload to compensate for some or all of the anticipated wear.

Housing elasticity can play a role in maintaining preload over the life of the application, and should be understood through physical or analytical measurements.

Purchasing bearings with honed ribs along with good, clean lubrication and good shaft alignment will assist with the break-in period and reduce overall loss of preload. **PTE**

Norm Parker is the bearing technical specialist for the driveline division at General Motors LLC, and thus a certified PTE "Expert."