

Fuel Efficiency Improvements in Heavy Truck Wheel Systems through Advanced Bearing Design and Technology

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ABSTRACT

The base design of commercial vehicle wheel end systems has changed very little over the past 50 years. Current bearings for R-drive and trailer wheel end systems were designed between the 1920's and the 1960's and designs have essentially remained the same. Over the same period of time, considerable gains have been made in bearing design, manufacturing capabilities and materials science. These gains allow for the opportunity to significantly increase bearing load capacity and improve efficiency.

Government emissions regulations and the need for fuel efficiency improvements in truck fleets are driving the opportunity for redesigned wheel end systems. The EPA and NHTSA standard requires up to 23% reduction in emissions and fuel consumption by 2017 relative to the 2010 baseline for heavy-duty tractor combinations.

This paper summarizes the history of current wheel end bearing designs and the opportunity for change to lighter-weight, cooler-running and more fuel-efficient wheel bearing designs to help meet the new industry standards. Advanced design and modeling principles were applied to analyze improvements, leading to optimized designs. Physical testing validated that these new bearing designs can increase efficiency and reduce weight.

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INTRODUCTION

The majority of Class 6 through Class 8 commercial vehicles on US highways use standard wheel bearing configurations designed many decades ago. The most popular are the "R-drive" and "FF steer" spindle arrangements on the tractor and either "P-spindle" or "N-spindle" arrangements on the trailer wheel end systems.

The parameters of a commercially available, heavy truck trailer with a P-spindle wheel end arrangement were used as the basis for this study. Fuel efficient bearings were designed to replace the current bearings used at both the inboard and outboard wheel bearing positions.

The purpose of this study is to:

1. Review the history of current bearing designs
2. Highlight the opportunity for power density
3. Document efficiency test methodology
4. Review the test results
5. Highlight the opportunity for improved efficiency

HISTORY OF BEARING RATINGS

A typical commercial vehicle wheel system in North America uses two single-row tapered roller bearings mounted in an indirect configuration. See a sectioned view of a tapered bearing in [Figure 1](#).



Figure 1.

The bearing cones are loosely mounted on the spindle and the cups are tightly mounted in the wheel hub. The popular R-drive spindle and wheel bearing arrangement is used at the four drive wheel positions of a tandem axle. The inboard bearing

includes the 594A cone and 592A cup, both of which are in the 595 bearing design series. A sectioned view of a typical wheel end arrangement is shown in Figure 2.

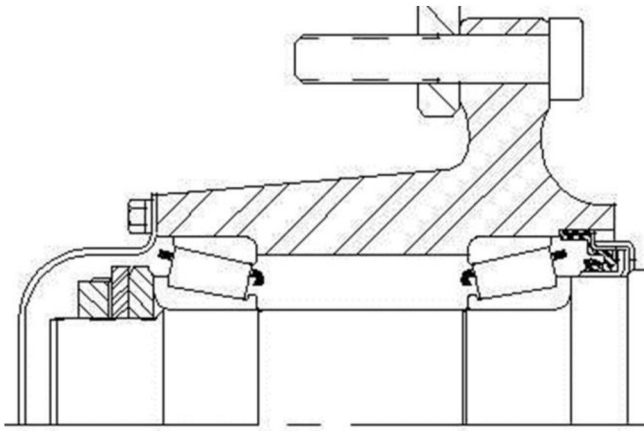


Figure 2.

The 595 series is first identified in Timken bearing design records dating back to 1913. The first records of the 592A cup are from 1926 and the first use of the 594A cone was identified in 1935. In 1937 the first mathematical rating formula based on fatigue performance of tapered roller bearings was published. (1) The C_{90} radial rating of the 594A bearing with 24 rollers was 9,350 pounds (lb) at this earliest release. Since that time, bearings have undergone numerous ratings increases due to improvements in material and manufacturing techniques discussed further below. With its largely unchanged internal geometry, this bearing today carries a C_{90} radial rating of 12,600 lb.

North American tanker and refrigerated trailers commonly use the P-spindle wheel bearing arrangement at the four wheel positions. The same mounting configuration previously described is used here as well, but with the HM518445 cone and the HM518410 cup. The HM518400 series, to which these components belong, is first identified in Timken bearing design records in 1965. The C_{90} radial rating of the bearing with 21 rollers was 11,000 lb at that time. The current C_{90} radial rating is 17,300 lb. These are just a few examples showing the typical ratings increases for tapered bearings over this time period.

MANUFACTURING AND STEEL IMPROVEMENTS

In the decades since the original production release of these bearings, significant improvements have been made in both bearing manufacturing and steel making.

Technological ingenuity in the areas of tube piercing and centerless grinding helped improve production in the 1930s. Statistical Process Control (SPC) was implemented to improve quality in the 1940s. Continuous improvement and consistency in meeting component dimensions because of these practices

allowed for enhanced bearing performance. And accelerated bearing life test equipment and methods were developed to validate these manufacturing improvements.

In the 1970s, efforts were made to improve the cleanliness (oxide inclusion content) of low carbon bearing steels. The development of ultrasonic inspection to compare cleanliness of bearing races showed a direct correlation between reducing the number of inclusions and improved bearing fatigue life. Initial work focused on reducing aluminum oxide inclusions from the alloy adjustment phase. Precipitation deoxidation - the earlier addition of aluminum alloy - replaced vacuum carbon deoxidation (1). Other improvements including argon stirring and a hot topping system were implemented to float the aluminum oxides to the top of the ingot, helping improve cleanliness and increase bearing performance. The next improvement employed argon shrouding of the teeming stream to protect the steel from reoxidation. This cleanliness improvement equated to another increase in bearing performance. Additional major improvements were the investment in bottom pour ingot technology and the electric arc furnace to serve as the melting/oxidation vessel. Because of this, the ladle became the steelmaking vessel where inclusion removal degassing, alloying to final chemistry and temperature corrections are made.(2) Along with additional continuous improvements, bottom pour ingots can now produce steel with cleanliness comparable to vacuum-arc remelted (VAR) steel.

The improvements in manufacturing practices starting in the 30's and 40's allowed for multiple dynamic load rating increases from the 1940's through the 1960's. Steelmaking practices to improve the cleanliness of steel again boosted dynamic load ratings in the 1980's. The effect was the ability to increase the loads on current bearing sizes or to use smaller bearings to carry the same load as before.

POWER DENSITY

The concept of "power density" was described as early as 1985 in a textbook by Winter and Niemann of the University of Munich. (3) For commercial truck wheel bearings designed as early as 1935, this meant that advances in manufacturing technology and steel making increased load-carrying capacity such that the same size bearings in these applications were now oversized.

A closer look at the formula to calculate bearing life will further explain this concept. The equation below for a tapered roller bearing is the life that 90% of a group of apparently identical bearings will meet or exceed before a fatigue spall develops.

$$L_{10} = \left(\frac{C_{90}}{P} \right)^{\frac{10}{3}} * \frac{1.5 \times 10^6}{n}$$

Where:

C_{90} = Dynamic Radial Load Rating (Capacity)

P = Dynamic Equivalent Radial Load

n = Rotational Speed

This indicates that an increase in load rating, C_{90} - with load remaining constant - will increase bearing life. Thus, validated through bearing life testing, load rating increases will equate to excess bearing life for the same application load conditions. The above equation is considered a "Catalog Life" equation, where a fixed set of environmental conditions is assumed. In reality, bearing environmental conditions vary for every bearing application.

To predict bearing life more accurately, environmental life adjustment factors are added to the life equation. The following equation includes environmental life adjustment factors that impact the potential for power density.

$$L_{10} = \left(\frac{C_{90}}{P} \right)^{\frac{10}{3}} * \frac{1.5 \times 10^6}{n} * a_{3k} * a_{3m} * a_{3l} * a_{3p}$$

Where:

a_{3k} = load zone factor

a_{3m} = misalignment factor

a_{3l} = lubrication factor

a_{3p} = low load factor

A commercial truck wheel bearing will operate under variable conditions over its life, from full to empty loads, straight driving, turns and scrubbing, high and low speeds, and cold and hot temperatures. Improved analysis methods now account for these various load conditions, including those low-load conditions. Along with the previously mentioned advances in manufacturing and material, these factors provide the capability to more accurately design bearings with the appropriate load rating to match specific application requirements.

Additional power density can be achieved within a current bearing size by improving the performance of the bearing under misaligned conditions. Under turning and scrubbing conditions, bearings are subject to misalignment due to lateral forces at the tire patch. This angular misalignment of the cup and cone centerline creates high stress levels at the ends of the line contact between the roller and cone and cup race surfaces. Standard bearing geometry uses a simple crowned profile machined onto the cone, cup and roller surfaces. These simple crowns compensate for small amounts of misalignment. However, they do not prevent stress increases from moderate to severe misalignment. High stress levels can lead to premature bearing damage. By designing an enhanced profile on the race surfaces, the edge stress levels can be reduced, thus improving bearing life.

Further power density can be achieved by addressing the lubrication conditions within a bearing. The lubrication film thickness at points of rolling contact is a critical variable impacting bearing life. The film thickness between rolling elements changes per condition and is dependent upon speed, oil viscosity, load, and surface finish. Under heavy loads or low speeds, marginal lubricant film may allow for asperity contact causing micro-spalling or peeling damage. Reducing the surface roughness will help prevent potential damage from asperity

contact. Advanced finishing processes can be applied to roller and race surfaces and the thrust rib face, minimizing the impact of thin lubricant conditions, thus improving bearing life.

POWER-DENSE VS STANDARD BEARING DESIGNS

A focus on the R-drive and P-spindle wheel end bearings highlights the benefits of power density. A comparison of the current R-drive inboard standard bearing and the power-dense design can be seen in [Table 1](#).

Table 1.

	Standard 594A – 592A	Power-Dense Bearing
Cone Bore (in)	3.750	3.750
Cup OD (in)	6.000	5.625
Bearing Width (in)	1.5625	1.0658
Radial Capacity C_{90} (lb)	12600	11300
Axial Capacity C_{a90} (lb)	9530	9300
Weight (lb)	5.40	2.40

It can be seen from [Table 1](#) that application of power-dense concepts can reduce the outside diameter and width of the bearing for a maintained bore. This size reduction equates to a 3.0-pound reduction in bearing weight.

The potential for redesign is possible because of the ratings increase above 9350 pounds, which was established in the 1930s. The new powerdense design with enhanced features and at a reduced size and weight also achieves more than sufficient predicted life per [Figure 3](#).

[Figure 3](#) demonstrates that the outboard bearing is limiting the bearing system life, since the load line is closer to this bearing for the wide single wheel arrangement. Enhanced features were applied to the outboard bearing to boost its performance. However, the standard inboard bearing was oversized which allowed for capacity, size and weight reduction without adversely impacting the bearing system life.

A comparison of the current P-spindle inboard/outboard bearing and the power dense-design can be seen in [Table 2](#).

It can be seen from [Table 2](#) that application of power-dense concepts can reduce the outside diameter and width of the P-Trailer bearing for a fixed bore size. This size reduction equates to a 2.44-pound reduction in bearing weight. Since the same bearing is used at the inboard and outboard positions on the P-spindle, the wheel end weight savings is 4.88 pounds.