

Tapered Roller Bearing Application Guide

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This article summarizes some common hurdles, issues and questions encountered by the newer and casual users of tapered roller bearings.

There are few other bearings that can drive people into fits like tapered roller bearings. They are supplied as two separate pieces, the installation must be perfect, they usually need to be preloaded, they are temperamental and hard to service; but, when your application demands maximum precision and power density, there are few alternatives that can compete with the tapered roller bearing, aka TRB.

Below is a basic 1-5 rating chart showing the benefits and drawbacks of the most popular bearing selections (there are many more). In terms of load and abuse-handling capability, the TRB earns its place as the king of bearings.

If you are a new designer or application engineer for TRBs, the “how-to” guides are abundant but tend to talk in generalities. Usable information can be difficult to find. This guide attempts to address some common trouble areas that are not always obvious to the casual user. This article will frequently refer to the bearing catalogs, so it will be useful to have one handy for reference.

Intro to TRBs

Becoming familiar with TRB nomenclature will become useful as you walk through the application process. For the purpose of this guide we will limit the discussion to radial, single-row tapered roller bearings with stamped cages, as opposed to the various paired arrangements and cages which are more common in heavy industry.

TRBs have separable cups and cones and are usually sold individually. It

would not be the first time that somebody started building prototypes to find they only ordered half of the bearing. Matched sets are available for higher precision, but are difficult to implement in high-production operations because the cup is often installed into the housing as a completely separate operation from the cone. Cups and cones from different suppliers should not be mixed. Even though the part numbers and appearance may be identical, the crowning of the contact surfaces between the cone, cup and rollers are designed to match and are not standardized.

The cone of the TRB will always come assembled with the rollers and is referred to as the cone assembly. The assembly consists of the cone, rollers and cage. For small to medium



Figure 1 TDO-style dual-row TRB. (courtesy RKB)



Figure 2 Single-row TRB. (courtesy Schaeffler Group)

	Tapered Roller Bearing *	Ball Bearing	Angular Contact Ball Bearing*	Cylindrical Roller Bearing	Needle Roller Bearing
Radial Load	1	3	3	1	1
Axial Load	1	4	2	5	0
Combined Radial/Axial Load	1	3	2	5	0
Power Density **	1	4	3	2	2
Vibration & Impact Capability	1	3	3	2	4
High Speed Capability	3	1	1	2	1
Low Torque	3	1	1	2	1
Rigidity	1	4	2	3	3
Misalignment Capability	3	2	2	4	5
Preload Capability	1	5	1	0	0
Overall Ease of Use	5	1	5	3	3

1 = Best, 5 = Worst, 0 = Not Capable, * Preloaded matched pair, ** Load rating/mass

sized applications, the cage will always be stamped strip-steel of various grades. For large bearings, where the size of the stamping becomes impractical, pin-type, machined steel or brass cages may be utilized. Wheel hub assembly tapered bearings often use polymer cages but are not yet available in single-row tapered bearings (but we have been assured for the last decade that they are “almost” ready).

TRBs can only take a thrust load in one direction and therefore need to be paired with a mating bearing for most applications. Occasionally a TRB is paired with an angular contact ball bearing for various reasons; but overwhelmingly, two tapers are paired together. The two are often different sizes, depending on the loading situation.

Sizing up Your System

Likely, one of the first things you will receive as the bearing application engineer is the duty cycle and a schematic of the system, and/or you will be asked to help develop one. Early in the design stage, it is not uncommon for this to change weekly or even daily, so set up a system that is easy to make iterations with. By the end of a design cycle you may easily have over 100 iterations of a design. You will thank yourself later for utilizing good organization early on.

It is fairly common practice for the bearing engineer to reduce the duty cycle down to a workable size of 10–15 steps through equivalent damage calculations, which are beyond the scope of this article. It is important to capture adequate parts of every section of the duty cycle to ensure you don’t average out a problem area. For instance, while you may only have 100k cycles and a very low speed and high torque, this section may cause trouble down the road if ignored. While you are working on the duty cycle, it is also very useful to have a separate reduction of only one step; that is one single equivalent load, speed and life requirement for the entire duty cycle, in addition to the 10–15 step reduction. This makes for an easy way to calculate an estimated load rating needed for the bearing. An example of this is shown later.

After you have the duty cycle ready and have enough information to calculate loading, it is time to get down to business. Early on, you should determine the critical characteristics of the design. Assuming top priority is always meeting the basic durability functions—Is stiffness a critical function? Is efficiency important? Will your lubrication become contaminated? Are there safety concerns with failure? Do you expect impact loading? All of these questions will guide you to what level of quality, safety factors and material type you may need.

If you have a bearing software suite, most of the work for determining bearing loads will be spent in modeling your system. Don’t be discouraged if you don’t have software, as many complicated systems are still calculated by hand. If you don’t have the (substantial) budget for modeling software and you frequently design similar systems with several iterations, it may be worth taking the time to make your own program. A simple program to calculate axle loads on each bearing can be made in *Excel VBA* forms in a few days with very limited programming knowledge (one way is to just open *Excel*, hit alt-F11 and you are off to the races). Many free sources are available online for training and reference. Otherwise, you can just plug-and-chug formulas into a spreadsheet.

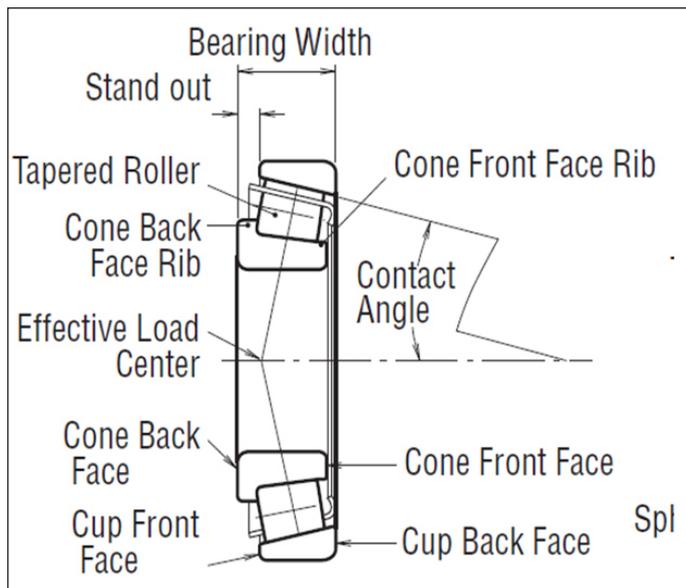


Figure 3 TRB features.

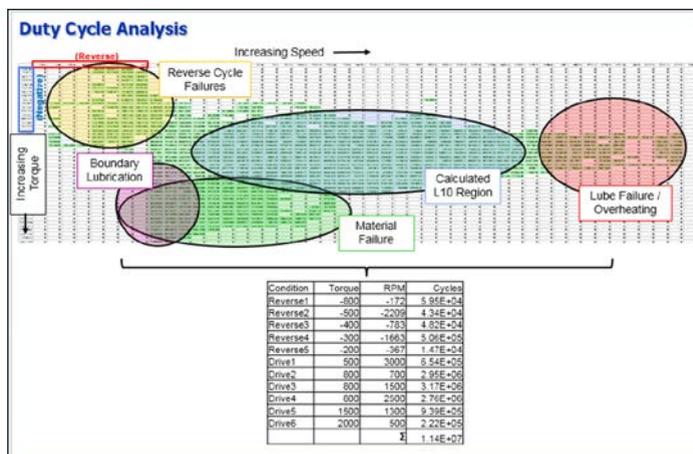


Figure 4 Reducing the duty cycle into workable pieces.

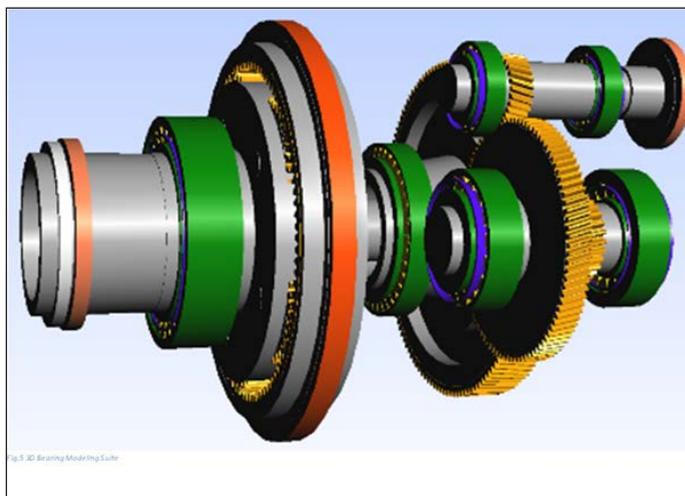


Figure 5 A 3-D bearing modeling suite.

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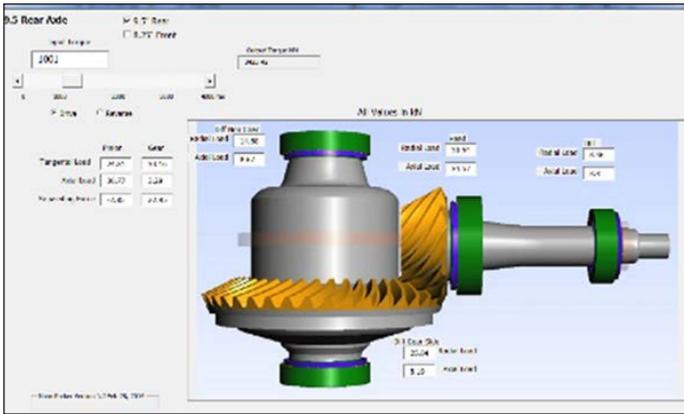


Figure 6 Excel VBA form.

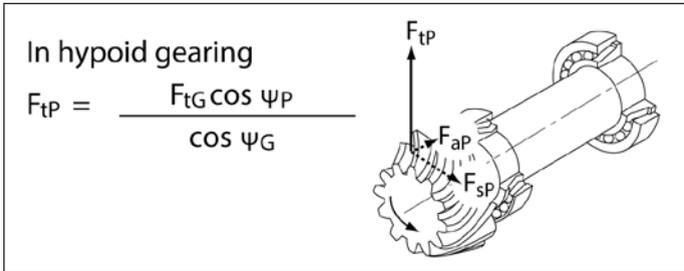


Figure 7 Timken hypoid calculations section.

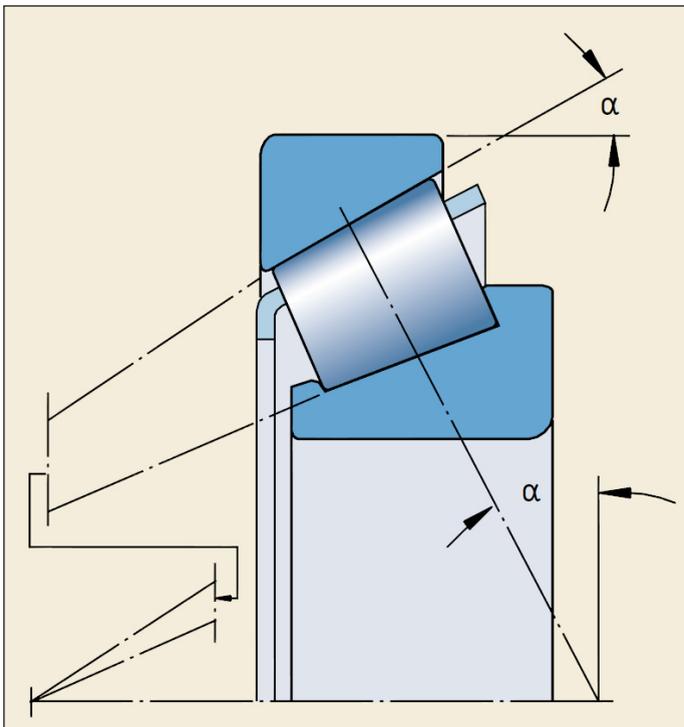


Figure 8 TRB contact angle. (courtesy SKF)

Most bearing catalogs have a nice section outlining how to calculate bearing loads for various systems. This is a tedious and time consuming process, but when executed properly, is accurate enough for most purposes.

For fatigue, or L_{10} calculations, we only need to be concerned with radial and axial loading. The direction of the radial load and the resultant between radial and axial loads will be used in other areas, but is not needed for L_{10} .

After your loading calculations are set up, the next step is to select representative bearings. This will be an iterative process, so your best guess is sufficient at this point.

As you browse your bearing catalog, you have likely noticed there are inch and metric series bearings. There is actually a third type hidden in the inch series section known as the “J” series. The “J” series is basically metric-sized and tolerance-bearing, with an inch style numbering system. Most major bearing companies produce all three series, though non-U.S. companies tend to produce more metric sizes while large domestic suppliers (particularly for axles) have a wider offering of inch and “J” series. *Just a note of caution here:* the inch and metric series bearing tolerances are in opposite directions. Metric and “J” series bearings always have a unilateral negative tolerance (e.g. 40 mm +0/-0.012), while inch series always have a unilateral positive tolerance (e.g. 40 mm +0.013/-0). It is not uncommon for both series to be in the same assembly; if this is overlooked for fitting, headaches will ensue.

Now we can get in the ballpark for the minimum dynamic load rating. If you binned down your duty cycle to one step, now is an excellent time to use it. We can easily calculate a minimum dynamic load rating based on that value (See Example 1).

Example, Part 1.

Using standard catalog formulas and equivalent damage methods for a pinion head bearing, you have calculated an equivalent load of 10 kN for a duration of 2.5×10^9 cycles at 1,000 rpm. What is the minimum dynamic load rating to meet this criteria?

Having a single step value of 10 kN allows us to take a good guess for a bearing we might need by using the basic L_{10} formula: $L_{10} = (C/P)^{10/3}$

Plugging in our values and solving for the dynamic load rating C, we have (L_{10} is measured in millions of cycles):

$$2500 = (C/10,000N)^{10/3}$$

$$\rightarrow C = 10,000N(2500hr)^{3/10} = 104,564N$$

Our minimum anticipated dynamic load rating will be roughly Cr: 105 kN

Contact Angle α

Now we have a dynamic load rating target, but we have no idea what contact angle α is needed. There are a few different strategies for selecting a good angle, but all are centered on finding the ratio between radial and axial loads. When we refer to the contact angle, we are always referring to the angle of the cup in relation to the shaft or centerline (Fig. 8).

Generally speaking, if the radial (F_r) and axial (F_a) loads are equal, a decent contact angle would be in the 20-25° range. The greater the radial / axial load ratio, the lower the contact

angle should be and vice-versa. Ranges are typically 10-12° on the low end and 25-40° on the high end. If we had no axial load, we could flatten the bearing out completely and change over to a cylindrical bearing ($\alpha=0^\circ$). If we had no radial load, we could change to a pure thrust bearing ($\alpha=90^\circ$).

If the ratio of radial to axial loading remains constant (e.g. gearbox), this is an easy step because there is only one ratio. After the radial and axial loads are calculated, they can be resolved on the X-Y plane to give the loading direction in relation to the bearing contact angle (Fig. 9).

$$\tan^{-1}(Fr/Fa) = \text{resultant loading angle}$$

It might be tempting to conclude that the complement of the resultant would be the perfect contact angle, as it represents the plane normal to the load. There are a few issues with this; first, TRBs are about twice as stiff in the radial direction as the axial, so we want to take advantage of that by putting more load in the radial direction. Secondly, the steeper the angle, the larger the induced axial load (explained in the following section), which places more load on the opposite bearing. Finally, if there is a reverse load, it will likely have a completely different resultant load and needs to be accommodated. For these reasons, it turns out that approximately half of the complementary angle is often a decent contact angle to start working with.

Induced Axial Load F_{ac}

Directly related to the contact angle is the concept of induced axial loading F_{ac} . This is purely a function of geometry of the bearings. In Figure 10, if we only apply a radial load to the larger bearing, an axial component is developed as a function of the force acting on an inclined plane. That force is transmitted, or induced, through the shaft, to the other bearing with a magnitude of:

$$F_{ac} = \frac{F_r}{2Y_1}$$

Where Y_1 is the axial load factor that can be found in the bearing catalog tables for each bearing or can be calculated as shown below:

$$Y_1 = 0.4 / \tan(\alpha)$$

A deeper dive into induced axial loading is beyond the scope of this article, but a rule of thumb is to consider that, for many TRBs, approximately one-half the radial load is induced as an axial load on the opposite bearing. If both bearings are identical and have equivalent loads, these forces will balance and have no effect on fatigue life. When the bearings are different in size and loading conditions, care must be taken to ensure the induced load

on the smaller bearing is not being overlooked as it can be substantial.

Materials

Bearing companies have been arguing for 100 years over which type of steel is best suited for different bearing applications. Outside of the U.S., the overwhelming majority of bearing steels are through-hardened. In the U.S., however; there is a substantial supply of case-carburized TRBs. Even in the U.S., the other bearing types (ball, needle, cylindrical, etc.) are usually through-hardened. What makes the U.S. unique is in part due to the early days of the industry where high-production bearing companies founded in the U.S. began using case-carburized low-carbon steel due to lack of supply for high quality through-hardened steel in contrast to Japan where through-hardened steel was more available.

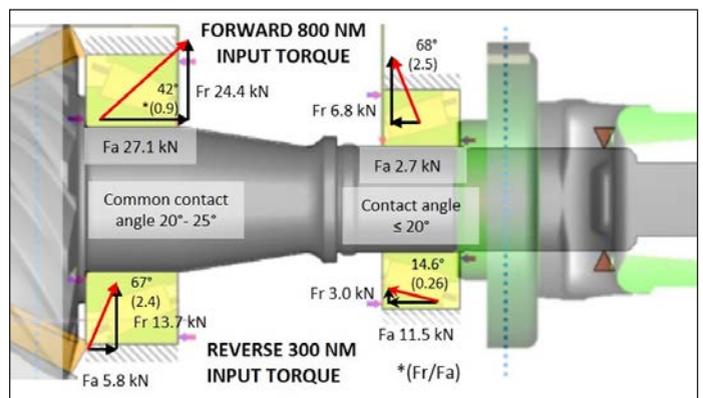


Figure 9 Pinion bearing load ratios.

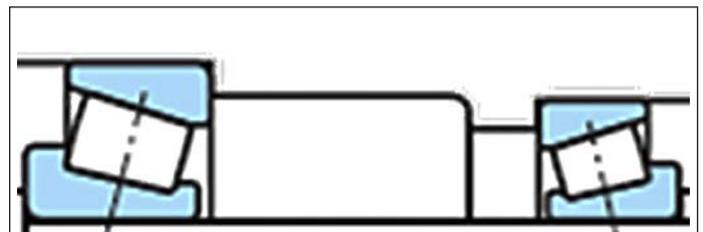


Figure 10 Induced axial load.

d	Boundary dimensions (mm)					r _{min.}	r _{1 min.}	Basic load ratings (kN)		Bearing No.		Constant e	Axial load factors	
	D	T	B	C	C _r			C _{0r}	Inner ring	Outer ring	Y ₁		Y ₀	
45.242	73.431	19.558	19.812	15.748	3.6	0.8	55.6	78.1	LM102949	LM102910	0.31	1.97	1.08	
	77.788	19.842	19.842	15.080	3.6	0.8	57.1	73.5	LM603049	LM603011	0.43	1.41	0.77	
	77.788	21.430	19.842	16.667	3.6	0.8	57.1	73.5	LM603049	LM603012	0.43	1.41	0.77	
	79.974	19.842	19.842	15.080	3.6	0.8	57.1	73.5	LM603049	LM603014	0.43	1.41	0.77	
45.618	85.000	23.812	25.400	19.050	3.6	2.4	77.2	100	25590	25526	0.33	1.79	0.99	
45.987	74.976	18.000	18.000	14.000	2.4	1.6	52.6	74.6	LM503349R	LM503310	0.40	1.49	0.82	
46.038	79.375	17.462	17.462	13.495	2.8	1.6	47.1	59.1	18690	18620	0.37	1.60	0.88	
	85.000	20.638	21.692	17.462	3.6	1.2	71.8	81.7	359A	354A	0.31	1.96	1.08	
	85.000	20.638	21.692	17.462	2.4	1.2	71.8	81.7	359S	354A	0.31	1.96	1.08	
	85.000	25.400	25.608	20.638	3.6	1.2	80.0	106	2984	2924	0.35	1.73	0.95	
47.625	88.900	20.638	22.225	16.513	3.6	1.2	74.3	87.3	369A	362A	0.32	1.88	1.03	
	88.900	25.400	25.400	19.050	3.6	3.2	87.1	112	M804049	M804010	0.55	1.10	0.60	
	95.250	30.162	29.370	23.020	3.6	3.2	104	140	HM804846	HM804810	0.55	1.10	0.60	
	96.838	21.000	21.946	15.875	0.8	0.8	80.4	101	386A	382A	0.35	1.69	0.93	
	101.600	34.925	36.068	26.988	3.6	3.2	131	159	528	522	0.29	2.10	1.16	
	104.775	30.162	29.317	24.605	4.8	3.2	109	144	463	453X	0.34	1.79	0.98	
	104.775	30.162	29.317	24.605	0.8	3.2	109	144	467	453X	0.34	1.79	0.98	

Figure 11 Standards Catalog (excerpt).

A common rule of thumb (though not written in stone) is if the application is going to experience sustained loads above 50% of the static load capacity with substantial shock loading, case-carburized steel is a good consideration. It is generally accepted that case-carburized bearings will outperform through-hardened bearings in high-performance applications, providing both are standard, non-premium products made with the same steel quality. Several enhancements can be made to either bearing to skew the results to the casual user.

The driving difference between the two materials lies in the residual stresses of the finished bearing. Fatigue theory has shown that materials with a greater net compressive residual surface stress will outperform those with low compressive or tensile surface stress.

Typically, case-carburized bearings have a high net-compressive surface stress, while the through-hardened bearing will have little if any compressive stress. In fact, if you cut through a bearing cone pressed onto a shaft, a case-carburized bearing will often clamp onto the grinding wheel or saw while a through-hardened bearing will pop open. If a substantial press fit is needed for a through-hardened bearing cone, there is a good chance the bearing cone is in tension after installation. This does not mean the bearing cannot perform well, in fact they frequently do. It simply means that if heavy loading and shock loads are present, a similar quality, case-carburized bearing would likely perform better. On the other hand, there is some argument that if the loads are low to moderate, a through-hardened bearing may perform better because the steel near the surface is often cleaner, resulting in longer fatigue life. People can (and do) go on for days about the potential benefits and drawbacks of each.

Another consideration is carbon-nitride treated bearings (CN). These come in two varieties; one is strictly a CN surface treatment, resulting in a harder surface which helps handle hard particle contamination. The other version is an actual CN case that is imparted into a medium carbon steel which has shown similar performance to a case-carburized bearing. Both applications of CN do have real and measurable benefits, though it is a premium product addition.

Now we have enough information to make a good guess at a bearing for our example application:

$$\longrightarrow C = 10,000N(2500hr)^{3/8} = 104,564N$$

Example, Part 2.

The shaft engineer informs you that his target diameter is between 47 mm – 49 mm. Your contact angle calculations indicate you will need a fairly steep angle bearing with Y_1 around 1. Additionally, the application team indicated that there will be heavy loads at times with occasional shock loading. As always, the smallest bearing that will meet the target is ideal.

The shaft size requirement already puts us into the inch series bearings if we want to stay with standard sizes. We are likely going to consider case-carburized bearings due to potential shock loading. The smallest bearing that would meet this requirement of 104 kN with Y_1 of 1.1 is the HM804846/10 with a dynamic load rating of 104 kN.

Installation Topics

A couple of significant items that we will not cover in this article are fitting and preload. Proper execution of your selected bearings can often be the most difficult part of the process. Do your homework and do not take these topics lightly.

Conclusion

Of course there are more sophisticated methods of calculating bearing life; however, if you utilize the information shown here along with the online calculators available, you can come close to the life calculated by sophisticated software packages. The real benefit of software comes through the time savings. At least 10× the amount of analysis can be performed with software as opposed to a spreadsheet with the additional benefit of shaft deflections, stress analysis, stiffness calculations, etc. That being said, I would never hire a new bearing engineer and have them skip the deep-seated knowledge gained by going through this process by hand. (See also Parker's Ask the Expert response on tapered rolling bearings in this issue.) **PTE**

References

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Koyo: <http://eb-cat.ds-navi.co.jp/enu/jtekt/tech/eb/eb1.asp>.

NSK: www.jp.nsk.com/app02/BearingGuide/html/bearing.html?PageCode=1005&LangID=1&ActCode=1

Schaeffler: www.schaeffler.us/content.schaeffler.us/us/products_services/inafagproducts/calculating/Calculation_and_testing.jsp

SKF: www.skf.com/us/knowledge-centre/engineering-tools/index.html

Timken: www.timken.com/en-us/knowledge/engineers/Pages/default.aspx

Norm Parker is the bearing technical specialist for the driveline division at General Motors LLC. Located onsite at the Milford (MI) Proving Grounds, he is regularly tasked with testing theoretical models in the real world, in real time. With his bachelor and master degrees in mechanical engineering from Oakland University (Rochester, Michigan), Parker has developed a keen interest in the academic, commercial and engineering aspects of the bearing industry. Prior to joining GM, he rose through the ranks of traditional bearing companies; by so doing he acquired invaluable experience in working with some of the largest customers — with the toughest applications and demands — on the planet. Parker plans to continue expanding his expertise and providing substantial personal contributions to bearing technology through metallurgy, design and processing.

