

Thermally Safe Operating Speed—DIN 732

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This article is Part III in a series of articles on speed rating of bearings. Part I appeared in the September 2022 issue ("Ball Bearing Limiting Speeds"), and Part II appeared in the October 2022 issue ("Ball Bearing Thermal Speed Rating").

Happy New Year! We are finally on to our 3rd and (maybe) final installment of speed ratings. For those playing along at home, DIN 732 is essentially the second part of ISO 15312. In ISO 15312, we solved the *Thermal Speed Rating*—which you could think of as a generic safe operating speed under normal conditions. DIN 732 defines a *Speed Ratio* which is a factor multiplied to the *Thermal Speed Rating* to calculate the *Thermally Safe Operating Speed*. This can be greater or less than 1 depending on conditions. Neither ISO 15312 or DIN 732 are stand-alone documents, they must be used together in order to calculate a meaningful result.

When we think about speed in terms of thermal limitations, the two primary drivers are load and lubrication. Lubrication is obvious, but perhaps not as obvious, are loads. Ball bearings can generate a lot of heat and become quite inefficient at high loads. It is a common notion that a ball bearing is always more efficient than a taper roller bearing. Intuitively, it makes sense. Tapers have a huge frictional sliding surface where the bottoms of the rollers and the rib meet. At first glance, it appears that a ball has almost no friction—a hardened steel ball on an equally

hardened raceway. The difference comes in surface pressure and the losses come in the form of hysteresis (and a few other drivers); the small elastic yielding of the ball and raceway under load. Under heavy loads, it is no longer just near frictionless rolling on a hard surface. As loads increase, eventually the ball bearing losses will surpass tapers. Tapers are designed to take huge loads, spreading a load over the full length of the roller. As a result, hysteretic losses are much lower. Under continuous high loads, like you might find in some industrial applications, a roller bearing or sometimes even a larger ball bearing can be more efficient in terms of rolling losses. In automotive, we are always trying to find the right balance between highway efficiency and high load capability.

Now, please don't go out and buy bulldozer bearings for your EDM design because Norm said they were more efficient—but you get the idea. There are tradeoffs with every design that must be calculated and not just assumed.

SKF has a nice, expanded explanation on loss mechanisms here:

[evolution.skf.com/using-a-friction-model-as-an-engineering-tool-3/](https://www.skf.com/using-a-friction-model-as-an-engineering-tool-3/)

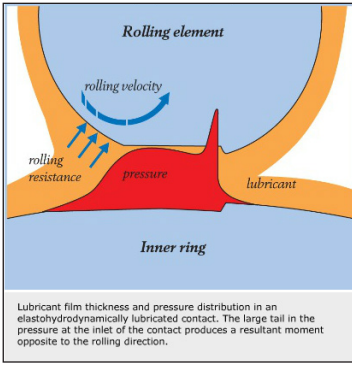


Figure 1 — Courtesy SKF.

As a recap, we first covered the catalog speed rating, otherwise known as the limiting speed. This is the mechanical based speed limitation based on cage strength and friction assumptions. If the bearing is sealed, there is a lower speed rating also listed in most catalogs alongside the limiting speed. The second rating we discussed was the thermal speed rating $n_{\theta r}$ within the context of ISO 15312. This is the speed rating which has predefined conditions for load, ambient temperature and lubrication type that will maintain a bearing temperature of 70°C. This speed rating on its own is not incredibly useful, but is described as an ancillary value for the *Thermally Safe Operating Speed* n_{θ} as defined in DIN 732. As we go through this process, it becomes clear why the limiting speed and thermal speed ratings are set values—they are based on pre-defined conditions. The *Thermally Safe Operating Speed* is a dynamic value that includes variables for load, temperature, viscosity and oil flow. We will start with the definition:

Thermally safe operating speed

$$n_{\theta} = n_{\theta r} \cdot f_n$$

For the thermal speed rating $n_{\theta r}$, we will use the value that we calculated from ISO 15312 in our previous session. Recall $n_{\theta r}$ for a standard 6205 is 13,300 rpm. Now we just have the speed ratio to deal with, which is still a bit of a handful. When the Lubricant Parameter K_L is defined within the normal operating range of $0.01 \leq K_L \leq 10$ and likewise with the Load Parameter K_P , $0.01 \leq K_P \leq 10$; the speed ratio f_n can be calculated as follows.

Speed ratio
490.77

$$f_n = \frac{490.77}{1 + 498.78 \cdot K_L^{0.599} + 852.88 \cdot K_P^{0.963} - 504.5 \cdot K_L^{0.055} \cdot K_P^{0.832}}$$

Where:

Lubricant factor

$$K_L = 10^{-6} \cdot \frac{\pi}{30} \cdot n_{\theta r} \cdot \frac{10^{-7} \cdot f_0 (v \cdot n_{\theta r})^{2/3} \cdot d_M^3}{\dot{Q}}$$

Load factor

$$K_P = 10^{-3} \cdot \frac{\pi}{30} \cdot n_{\theta r} \cdot \frac{f_1 \cdot P_1 \cdot d_M}{\dot{Q}}$$

Let's start by solving the total dissipated heat flow \dot{Q} :

$$\dot{Q} = \dot{Q}_S + \dot{Q}_L + \dot{Q}_E$$

Where:

$$\dot{Q}_S = k_q \cdot A_S \cdot \Delta\theta_A$$

In this case, the heat emitting bearing seating area, A_S , is the same as A_r used in ISO 15312. Again using our 6205 example:

$$A_S(6205) = \pi \cdot 15(52 + 25) = 3628.54 \text{ mm}^2$$

$\Delta\theta_A$ is the difference between operating temperature and ambient temperature. For ease of comparison, we will use the operating condition defined in ISO 15312 as 70°C. If we are operating at room temperature, $\Delta\theta_A = 47^\circ\text{C}$ (23°C to 70°C).

Schaeffler provides the chart for the k_q value. Below, the 1 line is for axial bearings and 2 is for radial bearings which includes the 6205 ball bearing. This gives us about 0.32×10^{-6} . Our description states we should have a value between $0.2 - 1.0 \times 10^{-6}$ which seems reasonable. There are a lot of unspecified factors here depending on architecture which is characterized by the large gray area. The faster the system can self-cool, the larger this value is.

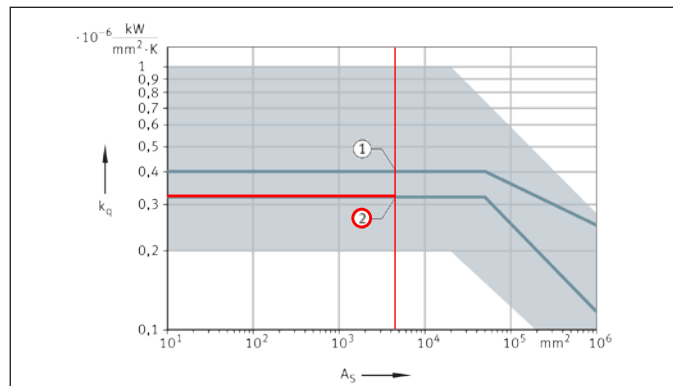


Figure 2 — <https://medias.schaeffler.us/en/speeds>

$$K_L = 10^{-3} \cdot \frac{\pi}{30} \cdot 13,300 \cdot \frac{10^{-7} \cdot 2(12 \cdot 13,300)^{2/3} \cdot 38.5^3}{390}$$

$$K_L = 0.12 \text{ (Within the required range of } 0.01 - 10)$$

Therefore:

$$\dot{Q}_S = 0.32 \cdot 10^{-6} \cdot 3628.54 \text{ mm}^2 \cdot 47 \text{ }^\circ\text{C}$$

$$\dot{Q}_S = 0.054 \text{ kW}$$

Now on to heat flow dissipated by the lubricant.

$$\dot{Q}_L = 0.0286 \frac{\text{kW}}{\text{l/min} \cdot \text{K}} \cdot \dot{V}_L \cdot \Delta\theta_A$$

\dot{V}_L is simply the oil flow. These numbers are always somewhat of a guess if you have a passive oil system. In these cases, you can just pick a reasonable middle of the road value ~ 0.25 l/min. There are published formulas that will give you estimates on needed oil flow rates based on speed, load and temperature.

Now we have:

$$\dot{Q}_L = 0.0286 \frac{\text{kW}}{\text{l/min} \cdot \text{K}} \cdot 0.25 \frac{\text{l}}{\text{min}} \cdot 47^\circ\text{C}$$

$$\dot{Q}_L = 0.336 \text{ kW}$$

Now we can solve the total dissipated flow rate. For this exercise, we will just set the external heating / cooling factor \dot{Q}_E to zero. If you have a heater or cooler, you will add that value here.

$$\dot{Q} = \dot{Q}_S + \dot{Q}_L$$

$$\dot{Q} = 0.058 + 0.358 = 0.39 \text{ kW (390W)}$$

Now we move to the lubricant factor:

$$K_L = 10^{-3} \cdot \frac{\pi}{30} \cdot n_{\theta r} \cdot \frac{10^{-7} \cdot f_0 (v \cdot n_{\theta r})^{2/3} \cdot d_M^3}{\dot{Q}}$$

The bearing factor f_0 is a table value that can be found in ISO 15312 and DIN 732 and referenced in public literature. In our case, the 6205 ball bearing has $f_0 = 2.0$ (for oil bath or recirculation).



v is the kinematic viscosity at operating temperature. For continuity, we will keep the value of 12 mm²/s as defined in ISO 15312. Recall that we defined our operating temperature at 73°C. Regardless of where you set your operating temperature, just make sure it is consistent. Again, we revisit our ISO 15312 results for the thermal speed rating $n_{\theta r}$: 13,300 rpm. d_M is the mean bearing diameter (6205): (52+25)/2 = 38.5 mm

Now on to the load factor K_p .

$$K_p = 10^{-3} \cdot \frac{\pi}{30} \cdot n_{\theta r} \cdot \frac{f_1 \cdot P_1 \cdot d_M}{\dot{Q}}$$

Most of these variables have now been defined—we just have f_1 , the *other* bearing factor and P_1 , the *decisive* load. f_1 is as a simple formula in DIN 732: $0.0009 \cdot (P_0/C_0)^{0.5}$ P_0 is the applied equivalent static load and C_0 is the static load rating. If axial loads are light, P_0 can be estimated as the applied radial load. For the 6205 C_0 , we will use the SKF rating of 7.8kN. For P_1 we will continue using the ISO 15312 parameter for ease of comparison – which was 5% of Cor (7.8kN) or 390N. Now we have $f_1 = 0.0009(0.39/7.8)^{0.5} = 0.0002$.

Now we can solve K_p .

$$K_p = 10^{-3} \cdot \frac{\pi}{30} \cdot 13,300 \cdot \frac{0.00023 \cdot 390 \cdot 38.5}{390 \text{ W}}$$

$$K_p = 0.0123^*$$

It is coincidental that P_1 and \dot{Q} are the same value. Units are included for clarity.

*Schaeffler also provides a chart for basic configuration for reference. We can see here that for light load, the K_p factor can easily run off the chart > 1. If we use the ISO 15312 value of 390N, this would give us a C_0/P (7.9/.39) of 20. Something doesn't seem to be lining up between the chart and the formula—the formula result seems too low. We will use our software to skip the guessing game, but for meantime, let's use the old college trick of setting the variable to 1 as a placeholder.

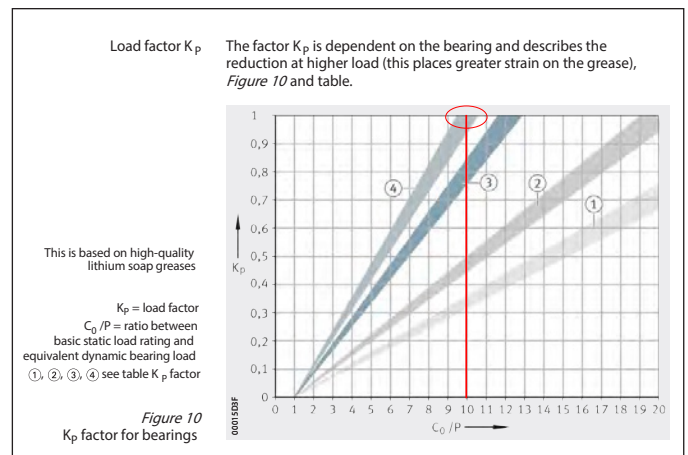


Figure 3—https://www.schaeffler.com/remotemedien/media/_shared_media/08_media_library/01_publications/schaeffler_2/catalogue_1/downloads_6/g11_de_en.pdf

Let us try solving f_n with K_L : 0.12 and K_p : 1.

$$f_n = \frac{490.77}{1 + 498.78 \cdot K_L^{0.599} + 852.88 \cdot K_p^{0.963} - 504.5 \cdot K_L^{0.055} \cdot K_p^{0.832}}$$

$$f_n = \frac{490.77}{1 + 498.78 \cdot (0.12)^{0.599} + 852.88 \cdot (1)^{0.963} - 504.5 \cdot (0.12)^{0.055} \cdot (1)^{0.832}}$$

$$f_n = 1.26$$

This answer doesn't seem terrible. We can do a little reverse engineering in *Masta* to see if our K_p assumption holds any water. Using all of the same operating assumptions from ISO 15312:

$$f_n = 24,535/16,558 = 1.48$$

DIN 732 Speed Rating			
Inner Ring Angular Velocity (rev/min)	ω_i		2000
Outer Ring Angular Velocity (rev/min)	ω_o		0
Relative Angular Velocity (rev/min)			2000
Maximum Grease Speed (rev/min)	N_g		18000
Maximum Oil Speed (rev/min)	N_o		18000
Thermal Limiting Speed (rev/min)			24535.8008
Thermal Limiting Speed f_0	f_0		1
Thermal Limiting Speed f_1	f_1		0.000199
Thermal Reference Speed (rev/min)			16558.3839
Dynamic Equivalent Load (kN)			0.39
Lubricant Windage and Churning Temperature (°C)	ⓘ		70
Air Convection Heat Dissipation (kW)	ⓘ		0.07257

Working Schaeffler's f_n chart backwards, we see that the K_p value should have been around 0.69. I don't have a quick answer on how to get that number from the formula. This may need some clarification with our ISO folks.

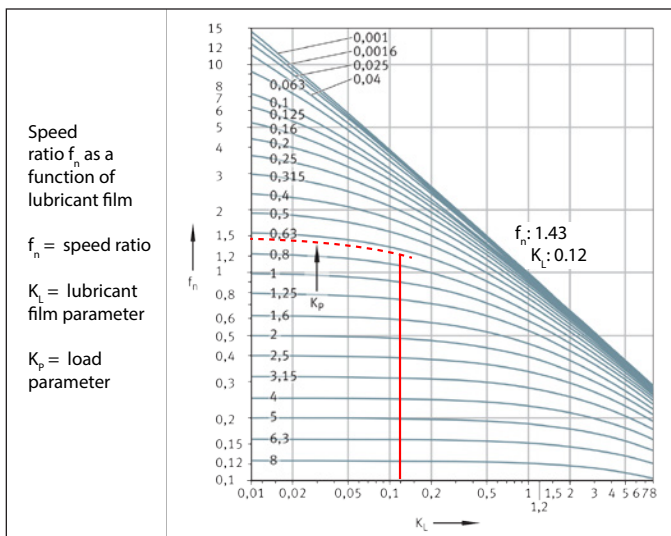
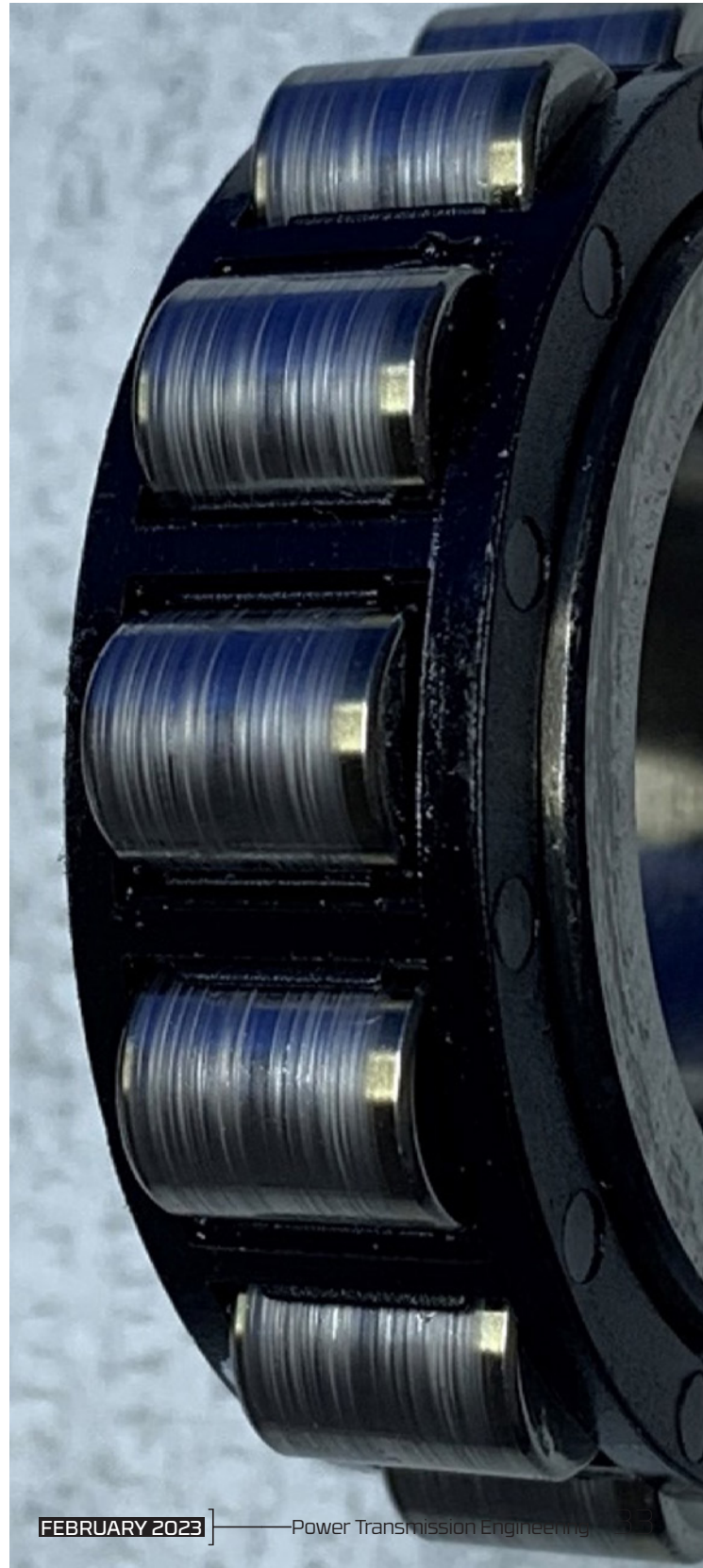


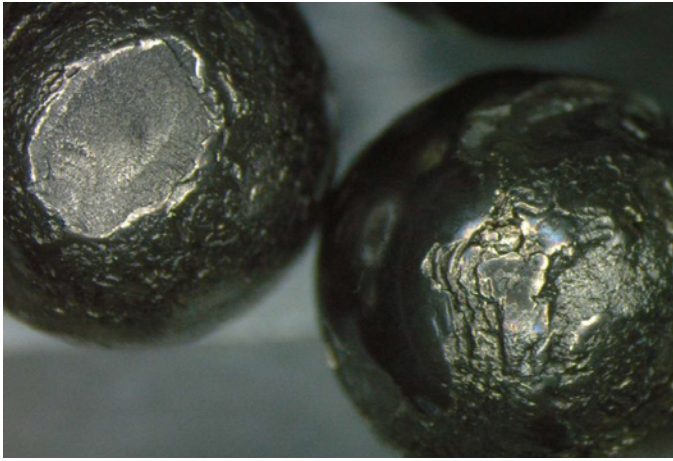
Figure 4—<https://medias.schaeffler.us/en/speeds>

We can solve that problem another day. Now let's solve our thermally safe operating speed with the software derived speed ratio:

$$n_g = n_{gr} \cdot f_n$$

$$n_g = 13,300 \cdot 1.48 = 19,684 \text{ rpm}$$

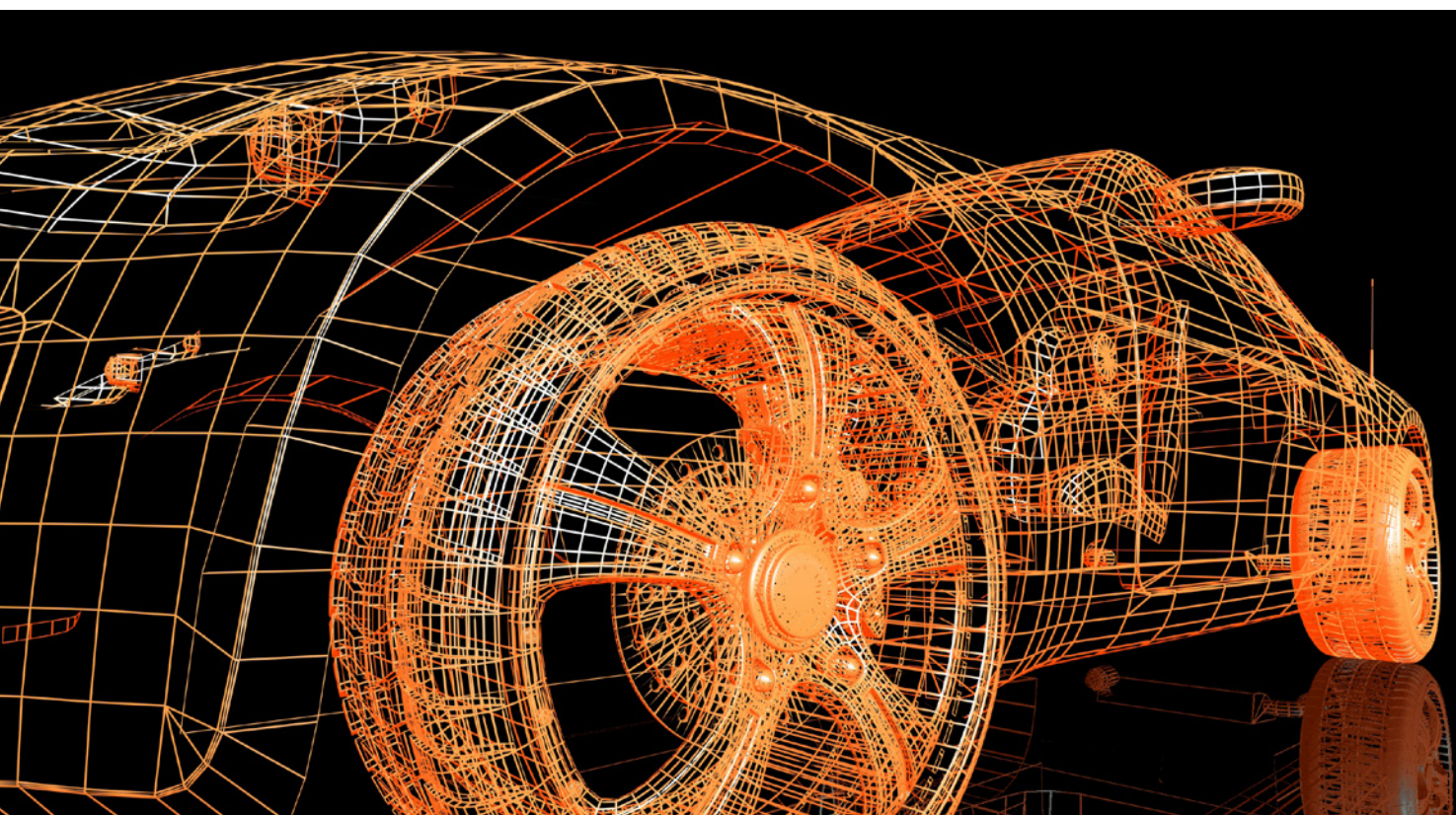




Alright, this is looking more promising. According to this standard, under the given conditions, we should be able to run this 6205 at 19,684 rpm continuously without overheating. *Continuously* is an important note because this means we can likely overshoot this value for brief periods. Of course we must always be cognizant of the mechanical limitations of the cage.

I thought it would be fun to compare some supplier's website results and *Masta* to see if our calculations come close. Many suppliers do not include thermal ratings—these are some of the few that do.

All results using standard 6205, 70°C, 390N Radial load, normal clearance, viscosity 12 mm ² /s	Our Results (using an SKF 6205 load ratings)	Masta	Schaeffler	NSK	SKF
Thermally safe operating speed $n\theta$	19,684	24,535	14,000	14,000	37,800
Thermal Speed Rating $n\theta_r$	13,300	16,558	14,400	14,000	28,000
Limiting Speed (catalog)	18,000	18,000	19,600	15,000	18,000
Comments	Used Masta to solve speed	SKF 6205	Most Conservative	Coincidence that $n\theta = n\theta_r$. Values diverge w/ changing variables	Likely need ceramic balls and custom cage to hit this $n\theta$



I am pretty satisfied with these results. Our hand calculated numbers are not as aggressive as *Masta*, but higher than the limiting speed—just what I would expect. Schaeffler and NSK are on the conservative side, but I strongly suspect you would receive better numbers with an application review. You should consult your SKF representative about their numbers.

PTE

Nomenclature

- n_{θ} —Thermally safe operating speed RPM.
- $n_{\theta r}$ —Thermal speed rating.
- f_n —Speed ratio.
- N_R —Frictional power kW.
- Q' —Total dissipated heat flow W.
- M_R —Total frictional torque Nmm.
- f_0 —Bearing factor for frictional torque as a function of speed.
- ν —Kinematic viscosity of the lubricant at operating temperature mm^2/s .
- f_1 —Bearing factor for frictional torque as a function of load.
- P_1 —Decisive load: radial load for radial bearings, axial load for axial bearings N.
- Q_s —Heat flow dissipated via the bearing seating surfaces kW.
- Q_L —Heat flow dissipated by the lubricant kW.
- Q_E —Heat flow. For heating by external source (+), for cooling by external source (-).
- K_L —Lubricant parameter.
- K_p —Load parameter.
- K_q —Heat transfer coefficient $10^{-6} \text{ kW}/(\text{mm}^2 \cdot \text{K})$ Heat transfer coefficient, as a function of the bearing seating surface. This is dependent on the housing design and size, the housing material and the installation position. For normal installation, the heat transfer coefficient for bearing seating surfaces up to $25,000 \text{ mm}^2$ is between $0.2 \cdot 10^{-6} \text{ kW}/(\text{mm}^2 \cdot \text{K})$ and $1.0 \cdot 10^{-6} \text{ kW}/(\text{mm}^2 \cdot \text{K})$.
- A_s —Heat-dissipating bearing seating surface mm^2 .
- $\Delta\theta_A$ —Difference between mean bearing temperature and ambient temperature K.
- V_L —Oil flow l/min.
- $\Delta\theta_L$ —Difference between oil inlet temperature and oil outlet temperature K.
- B —Bearing width mm.
- d —Bearing bore diameter mm.
- d_M —Mean bearing diameter $(D + d)/2$ mm.
- D —Bearing outside diameter mm.
- d_1 —Outside diameter of shaft locating washer mm.
- D_1 —Inside diameter of housing locating washer mm.
- T —Total width of tapered roller bearing mm.



Norm Parker is currently the Sr. Specialist Roller Bearings—Propulsion Systems at Stellantis. He's been contributing to PTE's "Bearings with Norm" blog since 2014.