Desktop Engineering – How to Calculate Dynamic and Static Load Ratings

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Introduction

When comparing bearing suppliers, engineers are often left with few options other than to compare dynamic load ratings and corresponding life calculations. Of course, we can look at steel and manufacturing quality; but if we are comparing sources of similar quality, those items may not provide a large contrast. It often surprises people to learn that bearing capacities are calculated values, not tested values. Lately, however, a trend is emerging for bearing suppliers to increase their ratings for higher performance bearings that have premium features such as higher quality steel and specilaized heat treatment. Bearing companies are under intense competitive pressure to make every feature add to the dynamic capacity of their bearings because it is very well understood that an increase in capacity adds to the bottom line. As a result, it is important that the end user develop a keen understanding of how capacity ratings and subsequent life calculations are generated in order to make a true comparison, or be left to comparing the claims of well-heeled marketing departments.

Dynamic Capacity

Nearly every calculation surrounding bearing life begins with the dynamic capacity, C_r. C_r is the equivalent load that would result in an average service life of one million revolutions. The formula is imperfect and standard bearings aren't designed to handle 100% of C_r; but, those are annoying nuances we have to understand and live with. In the past, most bearing companies would follow the dynamic capacity formula to ISO or ABMA standards and then increase the resulting life calculation by some factor based on heat treatment and other premium features in addition to the increase that would come through ISO 281 or 16281 using a-iso factors. The issues that end users have with this is that enhanced life calculation factors are not always shown on the bearing print and the print is the primary legal document that exists between the end user and the bearing manufacturer. Some end users interpret this as being an escape route for the bearing companies if something goes wrong – which it is not. Consequently, the competition and end users are pushing bearing companies to increase capacity ratings on the print. This is where the math starts getting fuzzy. We will walk through the formula with a couple of real bearings and determine where these numbers are coming from. The ISO/ABMA C_r formula:

$$C_r = b_m f_c \left(i L_{we} \cos \alpha\right)^{7/9} Z^{3/4} D_{we}^{29/27} \tag{1}$$

b_m	Rating factor for contemporary, commonly used, high quality hardened bearing steel in accordance with good manufacturing practices, the value of which varies with bearing type and design). ISO defines this value for tapers as 1.1
f_c	Factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material
i	Number of rows
L_{we}	Effective roller contact length
α	Bearing half angle (cup angle)
Z	Number of rollers per row
D_{we}	Mean roller diameter
D_{pw}	Pitch diameter of roller set

Figure 1 Company A (Left), Company B (Right).

Figure 2 Measuring major and minor inner raceway diameters for D_{pw}.

Figure 3 **Measuring roller length for L_{we}.**

Figure 4 **Measuring roller ends for D_{we}.**

For this example, let's look at two high-quality competitors, each producing their own design of the HM804846/10, a popular inch-series tapered roller bearing. We'll refer to these as Company A and Company B.

Right off, we see b_m is defined by ISO as 1.1. For i, both bearings have 1 row; $i = 1$. The bearing half-angle, α, will be provided by the manufacturer, so we can skip that measurement. Both of these bearings are around 20° (though a side-by-side comparison clearly shows they are not identical angles). Z, the number of rollers, is easy enough to count — both have 18 rollers. The remaining values — f_c, L_{we}, D_{we} and D_{pw} — are often not provided, but we can physically measure these features. Customer models will typically leave off just enough features to prevent an accurate measurement. We could get fancy and have these set up on a CMM and measure to 3 decimal places, but if you glance at the load ratings in the catalog you will see everything is rounded to the nearest 500 N. None of these factors will change your results greater than the rounding error if you are within 0.5 mm of accuracy. This sounds like a job for calipers.

We will skip f_c for now because that is a tabulated value which we need two of our other unknowns for. Let's start with the effective roller length L_{we}. ABMA defines L_{we} as:

The theoretical maximum length of contact between a roller and that raceway where the contact is shortest. NOTE: This is normally taken to be either the distance between the theoretically sharp corners of the roller minus the roller chamfers, or the raceway width excluding the grinding undercuts — whichever is the smaller.

The roller chamfer can be be hard to identify with the naked eye, and will usually involve a little guesswork.

Usually, the L_{we} will be 1-1.5 mm shorter than the entire length of the roller. We can check ourselves before we are done, so don't worry too much about your estimate for now. For a 21 mm roller, an L_{we} of 19.5 mm is a good guess.

Now on to D_{we} — the mean roller diameter. This is very straightforward; measure the large diameter at the bottom and the small diameter at the top and average the values for D_{we}.

The final measurement, D_{pw}, is also fairly simple. D_{pw}, the pitch diameter of the roller set, is the theoretical centerline that the rollers run on. This is measured in similar fashion as were the rollers; measure the large and small diameters of the inner ring raceway; take the average to find the diameter in the center, and then add 1 D_{we} to get the pitch diameter at the center of the rollers, at the center of the raceway.

With those values measured, we can now find f_c, which is a tabulated value based on the quotient.

$$\frac{D_{we}\cos\alpha}{D_{pw}} \tag{2}$$

For example, Company A

D_{we} = 10.2

D_{pw} = 71.2

α = 20

The quotient calculates to:

$$\frac{10.2\cos 20}{71.2} = .135$$

$$\therefore f_c = 87.4$$

$\frac{D_{we}\cos\alpha}{D_{we}}$	f_c
0.01	52.10
0.02	60.80
0.03	66.50
0.04	70.70
0.05	74.10
0.06	76.90
0.07	79.20
0.08	81.20
0.09	82.80
0.10	84.20
0.11	85.20
0.12	86.40
0.13	87.10
0.14	87.70
0.15	88.20
0.16	88.50

Figure 5 **f_c Table.**

Let's compare our values and results:

Plugging these values back into the formula:

$C_r = 1.1 \cdot 87.4(1 \cdot 21.3 \cdot cos20)^{7/9} 18^{3/4} 10.2^{29/27}$

C_r Company A: 104,675 N

C_r Company B: 106,144 N

If your calculated value is more than 1% different than the published value, adjust the L_{we} until the calculated C_r matches the book value.

HM804846/10		A	B
Material constant (ISO value is 1.1)	b_m	1.1	1.1
Geometry depenent factor	f_c	87.4	87.4
number of rows	i	1	1
Effective roller contact length	L_{we}	21.3	21.6
bearing half angle (cup angle)	α	20	19.37
Number of rollers per row	Z	18	18
Mean roller diameter	D_{we}	10.2	10.2
Pitch diameter of roller set	D_{pw}	71.2	71.1

Figure 6 Measured values for Co.'s A and B.

Static Capacity

By definition, the static capacity C_{or} is the calculated maximum-recommended static load value which loosely represents the yield point of the bearing steel. Ideally, this value should represent peak stress levels around 4,000 MPa — the ISO-recommended stress limit. Just due to geometry, the highest stress will occur on the inner ring/roller interface. The ball-ball contact between the inner ring and roller has a smaller contact area than the ball-socket contact pattern on the outer ring. C_{or} is a useful maximum load value if you don't have bearing software to calculate actual stress values. The benefit with using stress values is that the effects of crowning can be taken into account, and if the bearing has premium heat treatment features that produce a harder surface, stress values up to 4,200 MPa or higher may be permissible. Comparing catalog values of C_{or} can be very useful because there are no places to add non-standard factors; the formula is completely based on geometry. If you need a quick comparison for the physical amount of steel contact between two different bearings, forget C_r — C_{or} is what you want to compare.

The other good news is, if you collected your C_r values, you already have everything you need to calculate C_{or}.

$$C_{or} = 44\left(1 - \frac{D_{we}\cos\alpha}{D_{we}}\right) iZ L_{we} D_{we} \cos\alpha \tag{3}$$

C_{or} Company A: 139,926 N

C_{or} Company B: 142,337 N

If a bearing company wanted to increase the static rating on paper for a premium bearing, they could easily justify using a 4,200 MPa as a baseline for their rating, though it is not standard ISO/ABMA practice and not a fair comparison to another company that is strictly following ISO standards.

Let's compare all of our calculated values next to the published catalog values for both companies.

The calculated values for Company A came within 1% of the published values. However, something is quite different with Company B; the published C_r is 38% higher than our calculated value and the published C_{or} is 10% higher than our calculated value. Company A claims to have similar quality and performance as Company B, but we certainly cannot ignore the fact that Company B has a 41% higher C_r and a 12% higher C_{or}. This is a significant difference between two relatively similar bearings. What is going on here?

Company B claims that they have lab-tested proof to show that their increased C_r is legitimate and they do not want to be held to an artificially low ISO or ABMA formula, and therefore do not adhere to the standards. On the other hand, Company A claims that they are able to add a performance factor to the calculated L_{10} life that will give them nearly the same calculated life as Company B. Let's revisit the basic L_{10} formula so that we can play along:

$$L_{10} = \left(\frac{C_r}{P}\right)^{10/3} \tag{4}$$

	Company A	Company B
C_r Calculated	104,675	106,144
C_r. Published	104,000	147,000
$C_r\,\Delta$ %	-0.6%	38%
C_{or} Calculated	139.926	142.37
C_{or} Published	140.000	157.000
$C_r\,\Delta$ %	0.1%	10%

Figure 7 Calculated vs. published values.

Where L_{10} is measured in millions of revolutions and P is the applied load. Mathematically, an increase of X in C_r does this:

$$L_{10} = \left(\frac{X \cdot C_r}{P}\right)^{10/3}$$

While a performance factor does this:

$$L_{10} = X \cdot \left(\frac{C_r}{P}\right)^{10/3}$$

Because C_r is raised to the exponent of 10/3, a small increase nets large increases in calculated L_{10}. Let's see what type of performance factor a 38% increase in C_r would yield:

$$X = (1.38)^{10/3}$$
$$X = 3.2$$

This means that company A could multiply their calculated L_{10} by 3.2 times and effectively match the results of Company B. Company A states they are comfortable going with a performance factor of 2.6, but not 3.2 (Note: Until recently, Company B had a C_r of 141 kN that was exactly equivalent to a 2.6 performance factor. Two completely separate companies *coincidentally* had performance factors of 2.6). What the end users want the bearing companies to do is take the 2 or 2.6 performance factor and increase C_r by that amount on the print rather than just increasing the calculated L_{10}. For example, a performance factor of 2 would mean:

$$2 = \left(\frac{X \cdot C_r}{P}\right)^{10/3}$$

Let $C_r/P = 1$, then $X = 1.23$. This means that every 23% increase in dynamic capacity doubles the calculated life. End users want to see $1.23 \times C_r$, rather than $2 \times L_{10}$. The perceived benefit is that the increased C_r is shown on the print — which is a *legal document*. The risk in doing this for the bearing companies is that, right or wrong, some engineers are accustomed to designing to rules of thumb based on the published C_r. If C_r is artificially increased on the print, these practices may very easily result in a bearing that is under-designed for the application in terms of operating load and peak static stress.

The increased rating for C_{or} is easier to explain. As mentioned earlier, if you calculate the load required to reach a higher-than-ISO-recommended peak stress value of 4,000 MPa, you can easily justify the higher rating on the print. Though again, this is not standard practice.

From here it becomes difficult to make a rational decision, because there seems to be a lot of subjectivity going on with the calculations. We have tested vs. calculated dynamic load ratings, performance factors that have questionable origins, and less-than-obvious methods of increasing static load ratings. Recall the earlier statement that the static load rating calculations can be valuable for comparison. If we only compare *our* calculated static capacities (recall, true steel on steel contact area) we see a marginal difference of only 1%.

With that, we absolutely know that we have similar amounts of surface contact area. Armed with the knowledge that we have comparable geometry between the two bearings, the only real performance difference should be in the rolling fatigue performance of the steel. Again, we are assuming these are both top-shelf companies, so bearing design, manufacturing quality, surface finishes, etc., *should be* comparable. All of the fancy calculation methods beyond this point are useless for comparing these two bearings; only dyno or field performance tests over the entire loading range will conclusively separate the two. These formulas are easy to set up in a spreadsheet format that will facilitate future comparisons and provide real insight when dealing with your bearing suppliers.

Conclusion

There is an undeniable level of comfort when you see a huge capacity rating on a print that puts your safety factors well into "good night's sleep" territory. It can be argued that both Companies A and B have valid points in the way they handle the premium features. One does not want to be held to capacity ratings that they can outperform by 50%, and the other does not want to deviate from the standards.

The main point of this article is to show that load ratings are based on simple formulas that you can calculate on your own. You should ask a prospective supplier if their capacity ratings and life calculations are based on ISO 281:2007 and ISO 76:2006. If not, you need to completely understand how and why they are using their value. Likewise for any performance factors added to the calculated L_{10} life; double-check their work and ask questions. Secondly, a supplier is not off the hook just because they don't put their performance factor on the print. If their calculations are well-documented with all of the latest information you gave them, their analysis is a legal form of communication (though be forewarned — contamination levels, temperatures, alignment, roundness of shaft and bores…all of the factors that go into ISO 281 are subject to review). Finally, capacity ratings are pushed from an engineering and marketing perspective. Companies are expected to live up to their ratings, but with the wide scatter of failure points in any type of fatigue test, it can be difficult to pinpoint a true 20% difference during bench or field testing with a limited number of parts. We need to account for genuine high-performance features on our bearings because we use those factors in our designs. Just be sure that you know how to compare the different methods being used to account for those features. **PTE**

References

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