DESIGNING LARGE DIAMETER, CLOSELY COUPLED 2-ROW TAPERED ROLLER BEARINGS FOR SUPPORTING WIND TURBINE ROTOR LOADING

Douglas Lucas, Thierry Pontius
Introduction

Direct drive wind turbine generators require use of large diameter, closely spaced two-row tapered roller bearing to support the complex schedule of rotor loading. These bearings are ideally suited for such applications because of their ability to carry heavy radial and thrust loading in various combinations; because their design is built around the concept of zero slip which minimizes wear over long periods of operation; and because their internal clearance can be optimized with preload to help insure excellent system stability even in the heaviest of wind conditions. Generally speaking, such a bearing is custom designed for a specific wind turbine to survive at least 20 years. Such a reliability level requires extraordinary measures to be taken during the design process to insure that engineering consideration is given not only to the fatigue life requirement, but also to efficiency, structural integrity, handling, mounting, adjustment, lubrication and sealing. This paper will explore these criteria and steps taken to design a bearing suitable for a 1.5 MW wind turbine generator.

Mainshaft Bearing Comparison

Modular wind turbine generator (WTG) designs employ speed-increasing gearboxes. Spherical roller bearing (SRB) pillow blocks adjacent the rotor and input shaft bearings in the gearbox housings support the loads and torque to these gearboxes. A typical modular configuration is shown below in Figure 1.

Another design is the direct drive generator design. This style WTG eliminates the gearbox and often incorporates a large diameter three-row roller bearing design. A cross-section of the three-row roller bearing design is shown in Figure 2. The 3-row roller bearing, referred to in this paper as a 3-row CRB, uses two rows of cylindrical thrust rollers and one radial cylindrical row. This bearing was originally used in slow moving or oscillating slewing bearing applications and has been adapted to some WTG applications.

The primary features and functions of the 3-row CRB include:

- The two rows of thrust rollers support the axial loads (Fx) and overturning moments (My, Mz). The Germanischer Lloyd coordinate system in Figure 3 is used to define the important load directions.
  - These bearings have a very high overturning moment capacity and maintain a very high axial stiffness.
  - The radial bearing row supports the radial loads (Fy, Fz).
  - The radial rows have a moderate radial load carrying capacity.
  - The radial row is typically manufactured to have clearance, while the axial rows are designed to be lightly pre-loaded.

Figure 1 - Typical Gearbox Wind Turbine Design

Figure 2 - Example of a Typical Three-Row Roller Bearing

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• The operating bearing pre-load is sensitive to amount of surface wear.
• The 3-row CRB is typically produced pre-sealed and grease lubricated.
• Retainers typically separate the rollers.
• Special features and mounting configurations are available for various applications.

An alternative to the 3-row CRB is to use a close-coupled two-row tapered roller bearing or 2-row TRB (see Figure 4).

The primary features and functions include:
• Only two rows of rollers are used to support all combinations of radial, thrust, and moment loading with a proven true-rolling motion design to minimize skidding and the associated wear.
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• These bearings have a high overturning moment capacity and can maintain a high axial stiffness, while maintaining some compliancy.
• The 2-row TRB will be pre-loaded to optimize the load zones, and improve the bearing L10 fatigue life and dynamic stiffness. It can be pre-set at the bearing factory before installation into the wind turbine.
• Because the bearing is close-coupled, it can be manufactured as separate components, or it may be unitized into a non-separable assembly to simplify handling and installation.
• The 2-row TRB can be designed to be pre-greased and sealed, similar to the 3-row CRB.
• Full roller complements without retainers (see Figure 5) are available to increase the maximum dynamic bearing capacity. Coatings may be added to further increase bearing life.
• The bearings can be designed with normal sections for the tapered double outer (TDO) or tapered non-adjustable (TNA) type configurations for lowering the bearing costs.
• Special features and mounting configurations are available for various applications.

Bearing Load Sharing

True-rolling motion is designed into the 2-row TRB to minimize sliding or skidding between the rollers and the raceways. This reduces the potential for raceway surface wear leading to excessive clearance in the bearing.
The load zone is the angular loaded portion of the raceway measured in degrees. Figure 6 shows the loaded roller in a 180° load zone. Assuming unchanged external loads, as the bearing load zone decreases, the number of rollers sharing the load decreases, the rollers will have a higher contact stresses, and bearing fatigue life decreases. Load zone may decrease as a result of increased bearing clearances caused by wear and may result in lost traction between the rollers and races. These traction losses may result in sliding or skidding rollers. The occurrence of some non-fatigue related bearing failures may be eliminated by pre-loading the 2-row TRB and increasing the load zone. It will also eliminate stress concentrations that result from mis-alignment of the rolling elements as a result of bearing clearance, something a 3-row CRB does not do.

**Design of the 2-ROW TRB**

A typical close-coupled tapered roller bearing for mainshaft applications is composed of a double outer race [A] (or cup), and two inner races [B] (or cones), two rows of rollers [C], and a retainer [D] (cage) for each roller row as shown in Figure 7. Depending on the design, the bearings might not utilize a cage, in which case a full complement of rollers are used instead. Additionally there can be some means of unitization to hold the bearing together as one piece for installation. The intersection of the bearing centerline and the angled dashed lines in Figure 7 define the effective bearing spread for counteracting the overturning moments. www.bearing.sg

There are many design consideration required for the 2-row TRB for mainshaft applications. These details must be carefully considered and balanced in order to obtain a bearing that is optimized for performance, price, and manufacturing. The primary features of the bearing that must be considered in the design phase are discussed below.

- **Mean Pitch Diameter**

  This is the average of the bore and outside diameter of the bearing. In many cases the customer limits the available space for the bearing. However a bearing with a large mean diameter will provide an increased fatigue life and increased resistance to overturning moments.

- **Included Cup Angle, E**

  This parameter will be balanced based on the available envelope and will be optimized for performance. All portions of the load cycle, including extreme loads, are analyzed to get the best performance. The cup angle will define how well the bearing will perform against the overturning moments by defining the effective bearing spread. It will also affect the axial load capacity, fatigue life, operating efficiency, maintenance of load zone, setting, handling and mounting, and fatigue life. Radial capacity reduces with an increasing angle. A reasonable range for a 2-row TRB in a mainshaft application is 60° - 90°. A small-
er cup angle will be more compliant to make initial bearing setting less critical, however the effective bearing spread would also be reduced. Figure 8 shows how the axial preload varies with differing cup angles and bearing setting values. The larger cup angle (135°) is significantly stiffer than the smaller cup angles (90° and 60°). As the included cup angle approaches a thrust bearing (180°), the stiffness will be extremely high and the bearing preload will be very sensitive to wear.

![Figure 8 - Bearing Axial Stiffness for Various Included Cup Angles](image)

- **Included Roller Angle, F**

  The included roller angle should be minimized to reduce the induced thrust load on the cone rib, reduce the rib-roller contact stresses, and reduce the rib-roller sliding friction that generates heat and powerloss. With these considerations the design will be less prone to scoring and scuffing in the rib-roller contact areas. Again a balance must be achieved between the roller angle, bearing pitch diameter, and cup angle to achieve a small roller angle. A larger mean pitch diameter and smaller cup angle will both allow for smaller roller angles and still keep the apex converging on the bearing centerline.

- **Mean Roller Diameter**

  The mean roller diameter, the average of the large and small end diameters, effects the bearing fatigue life. A smaller diameter roller will increase the number of rollers in a bearing, increase the roller surface speed, and will result in an increased film strength. However it will also reduce the bearing capacity because of increased stresses from reduced Hertzian contact areas. With higher speeds, the rollers will be subject to an increased number of fatigue cycles which will reduce the bearing fatigue life.

  - Full Complement of Rollers

    Using a full complement of rollers, ref. Figure 5, is considered in applications where unitization of the bearing is used or where the size of the bearing is so large that typical bearing retainers or cages may not be feasible. The increased number of rollers increases the bearing capacity. The number of rollers affects the natural frequency of the bearing and typically a prime number of rollers is used. The full complement design is limited by speed, but the limitation can be overcome by the addition of coatings to the roller to increase the scuffing resistance.

  - Race Wall Sections for Support Structures

    Wall sections typically are maintained for manufacturing reasons to reduce distortion. However with 2-row TRB’s used as support structures also need to be analyzed with FEA to reduce the effect of bolt clamp loads on race profiles or deformation from high external loads. These races of the support structure bearings are not pressed into housing or onto shafts, but are attached by bolt or other means to the surrounding parts.

**Bearing Fatigue Duty Cycle**

The bearing fatigue duty cycle received from the customers can have a significant influence on the size and geometry of the mainshaft bearing designs. A concern is that adding conservatism by oversimplification of the duty cycle will result in a negative cost structure. Some manufacturers use hundreds of conditions in the duty cycle, some use tens of conditions, and others yet will use a single condition duty cycle.
Table 1 shows the wide discrepancy in the number of conditions in the duty cycle data from wind turbine manufacturers. Clearly some variation in bearing size and life calculation can be attributed to design variations in blade diameter and wind turbine location. Figure 9 shows the bearing life depletion rate of the first 100 conditions of differing duty cycles. It is shown that duty cycle “C” depletes 58% of the bearing life in 100 conditions and cycle “E” depletes all of the bearing life in just 25 conditions.

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**TABLE 1 - 1.5 MW VARIATION IN DUTY CYCLE CONDITIONS FOR VARIOUS WIND TURBINE MANUFACTURERS**

Duty cycles are typically generated by using design programs to model the wind turbine system, typically with an output at 20-Hz. The high frequency of data provides a vast number of snapshots of the system, even for short time intervals. All this data must be sorted and binned in useful categories, using the arithmetic average bin value, for fatigue analysis. A 5-second excerpt of data from the graph has been added in Table 2 to show the variation of the data. Variation in this short time is graphically shown in Figure 10. The complete data is then sorted into bins and the time durations in each bin is summed to determine the percent time each condition contributes to the duty cycle. This rainflow counted load spectrum is then used in an advanced bearing fatigue calculation program with Miner’s Rule to determine the bearing L10a fatigue life.

**TABLE 2 - TABULAR DATA OUTPUT FROM DESIGN PROGRAM**

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Bin size should be determined methodically for the speed and loads by understanding the effect on the bearing system. The recommended order of importance of the data for proper bearing is:

1. RPM

The bearing speed will affect the development of the film thickness (lambda ratio, the ratio of the film thickness to the surface finish) and ultimately affect the predicted bearing life.

2. Pitch Moment, My

The pitch moments contribute significantly bearing life reduction. Wind speed vertical distributions create these high moments and adjust the loading on the bearing rows in the XZ-plane that are a result of the rotor mass.

3. Yaw Moment, Mz

The yaw moments are perpendicular to the pitch moments and may be either clockwise or counterclockwise with respect to the tower axis. Consideration of the yaw moments can be simplified into absolute values without having a large effect on the bearing fatigue life calculation.

4. Radial Load, Fz

The radial load is relatively constant as a result of the weights of the rotor hub, blades, and generator.

5. Axial Load, Fx

In many cases the axial load is relatively constant for most condition in the duty cycle. A small number of coarse bins can be used.

6. Radial Load, Fy

The yaw loads are small compared to the pitch loads and are not as critical to the bearing fatigue life depletion.

**Advanced Life Methods**

There are many assumptions and simplifications that are used in a TRB bearing catalog life calculation. This includes:

- Bearing load zone is 180°
- Lubrication factor does not include surface finishes.
- Bearing alignment is assumed to be less than 0.0005-rad.

Advanced life methods can be used to more accurately determine the bearing fatigue lives to obtain the most bearing for the given resources. It is suggested that wind turbine manufacturers contact their approved bearing suppliers for advanced bearing life analysis. There are several life adjustment factors included in advanced bearing analysis in SYSx, a proprietary finite element based computer simulation software of the author’s company.

- Load Zone Factor, a3k

As the bearing loading changes, the load zone will change. In multiple condition duty cycles, the load zone can change dramatically and will affect bearing performance. This factor takes into account the change in roller loading on bearing life.
Figure 11 shows that a reduction in bearing preload on the unseated bearing will lead to a reduction in load zone for a range of conditions. One might conclude to increase the dimensional preload beyond 0.30-mm to ensure both rows are well seated under the heaviest loads. The preload would need increased significantly to dramatically increase the load zone above 110°.

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Figure 12 suggests that an increase in preload over the optimum -0.30-mm would lead to a reduced bearing system life. This shows that it is very important to have a proper duty cycle. An incorrect setting specification resulting from an improper duty cycle or simplified set of conditions may lead to reduced bearing system life and excessive heat generation.

- Low Load Factor

Predicted bearing life will be more accurately calculated for low loads when a stress life response factor (low load factor) is used. This factor will more accurately predict the life of a lightly loaded bearing as a result of a change in the Weibull slope resulting from a low stress state.

- Lubrication Factor, a₃l

Bearing fatigue life can be adjusted to either improve the life in the case of higher lambda ratios or reduce the life in case of low lambda ratios. It is important then to look at these lubrication conditions to adjust the bearing design.

- Alignment Factor, a₃m

The shaft alignment cannot be guaranteed in a wind turbine application because of the high loads and overturning moments. Mis-alignments will increase edge stresses in roller bearings, resulting in a GSC spall (geometric stress concentration). Race profiles can be recommended to remedy this situation.

An FEA approach has been added to SYSx to provide further optimization and accuracy to the bearing life simulation by predicting race out-of-round deformation conditions that change the bearing load zone and potentially decrease bearing fatigue life. Because the loads will effect the bearing race displacement, it is critical to evaluate applications were the race or housing sections are thin. The 2-D FEA mesh generator builds a compliance matrix for the race or housing; this is used to determine the out-of-round or ovalization of the bearing races. Figure 13 shows a classical 360° load zone distribution for uniform, circular races.
Using the 2-D Housing Modeler for the same application, the roller-race load zone changes as shown in Figure 14. The irregular shaped circular line is the deformed raceway, while the radial lines represent the roller loads. The load distribution, race deformation, and bearing life are strongly affected by the compliance matrix.

Reliability Requirements

There have been many bearing life requirements from various customers. Some have used 150,000 hours, while others have used 175,000 or even 200,000 hours for an L10 life (90% reliability).

As seen in Table 3, the required calculated L10 fatigue life for a 20 year design life increases with increasing reliability requirements in order to obtain the required reliability of 150,000 hours, assuming a Weibull slope of 1.5. Also shown in a 30-year life design. The Reliability Factor, a1, is multiplied by the L10 to attain the Ln life of 150,000 or 225,000 hours for a 20 or 30 year life, respectively.

It is important to understand that the reliability requirements are defined for failure by fatigue spall. There are other types of bearing failures that may occur in the application that are not protected against by traditional fatigue durability analysis. These include, but are not limited to:

- Scoring
  Scoring may occur on a roller bearing if the end of the roller contacts an improperly lubricated flange, high rib contact stresses, or improper contact geometry.

- Skidding
  Skidding will typically occur if there are insufficient traction forces between the roller and the raceways as a result of low bearing pre-load, a low load zone, or an off-apex design.

- Brinelling and False Brinelling
  Brinelling results from permanent deformation or yielding in the part. False brinelling is commonly seen when the rollers are not rotating and oscillate back and forth along the direction of the rotational axis of the roller.

- Structural Issues
  Structural issues may be related to sections of the inner or outer raceways that may be used as structural member to transmit the load instead of using a housing or shaft to transfer the load.

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Table 3 - L10 Life Requirement for Various Reliabilities
• Heat and seizure

Improperly defined bearing setting and improperly mounted bearings can result in excessive heat generation and bearing seizure.

Retainers and Unitization

There are a limited number of options for roller retaining devices. Pin style cages, polymer cages, and roller separators have not been accepted in the wind turbine industry due to reliability concerns. A machined “L” style steel cage may be too costly to manufacture for large diameter bearings. Some viable options for the 2-row TRB are:

• Precision cut “L” style cage

• No cage - full roller complement

The precision cut cages are similar to the traditional stamped steel cages used on smaller bearings, however they are manufactured by other means, which may include:

• Full machining

• Forming technology

• CNC controlled precision cutting

The size of the bearing retainer makes it difficult to close the cages around the rollers like traditional stamped steel cages. In some cases a means of axial retention is added to hold the rollers in place after assembly. The inner race assembly can then be handled separately from the outer race without a need of unitization.

If no cage is used, then the full complement of rollers typically requires bearing unitization to retain the rollers. There are several design considerations when using a full complement of rollers:

• Maximum allowable speed is limited to prevent metal transfer from roller to roller / race.

• Diamond-like coatings (DLC) on rollers will allow for increases in speed and will enhance bearing performance by altering the surface finish and improving the lambda ratios. The bearing life will be improved, particularly in low lambda conditions, by reducing adhesive metal transfer.

• Unitization will simplify bearing setting, installation and removal, and may help eliminate incidental damage to rollers during turbine assembly.

Lubrication

The primary consideration as with all bearing applications, is that there is a sufficient oil viscosity to maintain proper film strength. Because of the low speed of the mainshaft bearings, grease is a very viable alternative and is typically used in mainshaft pillowblock designs. Oil will work just as well for the 2-row TRB mainshaft bearings.

Although grease may result in a thinner film thickness, it is the preferred option for direct drive WTG applications. It will have a lower chance of leakage, will not migrate at easily, and will exclude containments more effectively than oil.

Common considerations for the grease selection process include:

• Higher viscosity (ISOVG 460 or 320) is better for maintaining good film strength.
• Synthetic base oil with high viscosity index (VI) will provide better lubrication over a larger temperature range.

• Excellent water, rust, oxidation, and corrosion resistance is important for extended grease lives.

• Low temperature operation and pumpability may be required in some applications.

Seals

Sealing is more critical in direct drive generator wind turbines. The generator may be damaged if oil or grease enters it. Typically non-contacting labyrinth seals are used to seal bearings and gearboxes in wind turbines. The problem with labyrinth seals is that they control the rate of leakage but do not eliminate it. Therefore it is most effective to have a series of seal types working together to eliminate leakage of grease or oil from the bearing systems. Conventional polymer sprung lips seals will work but may not last 20 years; provisions should be made to periodically replace them.

By using a unitized, 2-row TRB in the mainshaft position, a fewer number of seals may be required and the seals can be easily incorporated into the bearing assembly at the manufacturing plant.

Conclusion

As presented in this paper, there are many considerations that must be taken into account when designing a mainshaft bearing to last 20 years in a wind turbine application. Understanding of the WTG system is important for ensuring that the proper bearing selection was made. This paper has presented many of the criteria that need to be examined during the selection and design of the mainshaft bearing for a direct drive WTG. Advanced modeling techniques will help ensure that the bearing is efficiently and properly selected for the application without being overly conservative and adding prohibitive cost. The closely spaced two-row tapered roller bearing was shown to give advantages over the typical designs currently available due to the ability to carry heavy thrust and radial loads and maintain a tight connection between the rollers and the races through pre-load.