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The contributions regarding to the behavior of bearings in wind turbines

Panait Paula-Gabriela
Doctoral School of Mechanical and Mechatronics Engineering,
Maritime University
Constanta, Romania
paula.panait72@yahoo.com

Panaitescu Mariana, Scupi-Alexandru-Andrei, Panaitescu Fanel-Viorel
Engineering Sciences in Mechanics and Environment Department, Maritime University of Constanta
Constanta, Romania
panaitescumariana1@gmail.com, ID ORCID 0000-0002-8610-9554

Abstract—**With the growth of the wind industry in the world and the commissioning of turbines with powers higher than 1 MW, higher loads and increased stresses have a strong effect on the durability of the bearings that equip the main shaft and the kinematic transmission of wind turbine gears. As a result, the wind industry is demanding greater durability of the bearings that equip the main shaft and the kinematic transmission with gears, and manufacturers are striving to come up with solutions for these requirements. The novelty in this work consists in the identification of some solutions, as well as improvements of the existing solutions for the construction of some mechanical systems for the existing wind turbines. We present below some constructive bearing solutions that are effective in the wind turbine industry. The research in the paper consists of: stress analysis of the load-bearing structure for the axial radial bearing with straight rollers with single ring, some aspects about lubrication of blade bearing in wind turbine and defects in blade bearings.**

Keywords— wind turbine, bearing, stress, lubrication, simulation, defect, sustainable solution

I. Introduction

Globally the main global bearing manufacturers are SKF (Sweden), Timken (USA), Schaeffler Group (Germany), NTN (Japan), Iljin (Korea), NSK (Japan), JTEKT (Japan) [1]-[13]. From the analysis of the available works, but especially from the analysis of the activity reports of the major bearing manufacturers, a series of common concerns emerge in the field of bearing design, namely [1]-[13]:

• The development of mathematical models to estimate and increase the durability of bearings, different software applications are made based on the literature to estimate the durability of bearings.

•Development of profiles for the active surfaces of the bearings

• Development of new thermal treatments (bainite treatment) to increase the durability of bearings

• Development of special bearings for electric cars with reduced rolling resistance and special covers to isolate electrically.

• Design of large size bearings for wind turbines

• Development of smart bearings with sensor integration.

The safe and reliable operation of engineering systems is of great importance for the quality of production, as well as for the preservation of human life and health and the environment. With the growth of the wind industry in the world and the commissioning of turbines with powers higher than 1 MW, higher loads and increased stresses have a strong effect on the durability of the bearings that equip the main shaft and the kinematic transmission of wind turbine gears. Failures and premature decommissioning occur earlier than expected and for many wind farm operators. As a result, the wind industry is demanding greater durability of the bearings that equip the main shaft and the kinematic transmission with gears, and manufacturers are striving to come up with solutions for these requirements.

The novelty in this work consists in the identification of some solutions, as well as improvements of the existing solutions for the construction of some mechanical systems for the existing wind turbines.

II. Main Bearing Configurations Used In Wind Turbines

For the benefit of the wind industry, the reliability of the main shaft bearings must be improved [14]-[16]. Market demand motivates the development of new technical solutions to replace standard radial thrust roller bearings in three-point bearing turbines, such as wear-resistant radial roller bearings and two-point roller bearings, radial tapered roller bearings in a row, restressed [17]. For this reason, this paper presents some constructive solutions that are effective in the wind turbine industry (Fig. 1)[16], [18]- [21].

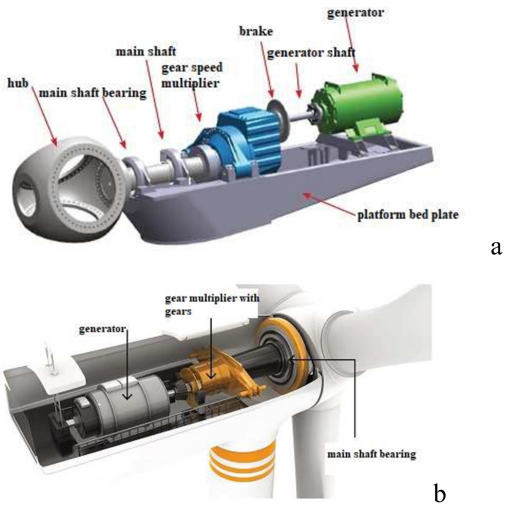

Fig. 1. Construction solutions for the main shaft bearing with three and four support points [21]

The use of a single double-row radial slewing roller bearing as the main shaft bearing in power wind turbines has changed - although in the past it was the preferred construction solution, operators are now looking for a better solution. [15] The main reason is the premature failure observed in this type of bearings, mainly due to micro-exfoliation (tread fatigue). Although there is no officially stated maximum limit, the percentage ratio of axial to radial loads considered acceptable for a double-row radial roller bearing is approximately 25%.

In the four-point bearing (Figure 1.b), the main shaft is supported by the torsion arms of the gear multiplier and two main bearings in front of the gear multiplier. These main bearings are often double-row angular contact roller bearings, but other design arrangements are commonly used, including cylindrical and tapered roller bearings.

Another improved construction solution presented in the paper is the wear-resistant double-row radial roller bearing, which uses special coating technology in combination with improved work surface finishes. They significantly reduce the shear stresses and the interaction between the micro-asperities of the working surfaces [21]. For a double-row radial roller bearing operating in a three-point bearing turbine the operating conditions of the main shaft bearing are not at all favorable for the generation of the lubricant film (Fig. 2) [21]. As a result, a wear-resistant double-row oscillating roller bearing was developed using a special coating technology in combination with improved finishes [20]- [22].

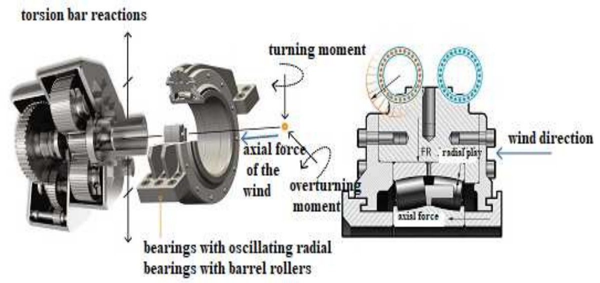

Fig. 2. The double-row oscillating roller radial bearing [21]

Wear-resistant bearings have raceways protected against micro-exfoliation, significantly reducing shear stresses and the interaction between the micro-asperities of the working surfaces.

The specially designed surface coating is composed of a durable and unique layer of amorphous hydrogen carbide reinforced with tungsten carbide. These coatings have a hardness two to three times higher than steel, between one and two micrometers thick and have a reduced coefficient of friction with steel.

With a specially designed surface coating, the coating deposited on the bearing rollers is intended to grind and repair raceways imprinted with hard contaminating particles during bearing operation. The improved surface finish also increases the thickness of the lubricant film, which means a more efficient separation of the micro-asperities of the working surfaces in contact. Combined, these improvements reduce the shear stresses that cause wear (Table 1)[21].

Table 1. Characteristics of the wear-resistant two-row radial barrel oscillating roller bearing.

Description of technology	Benefits
roller finishing	low roughness low tensions and contact between micro asperities
special surface coverage of the rollers: thickness 1 µm amorphous hydrogenated carbide reinforced with tungsten carbide	increases wear resistance and durability, increases resistance to hard particle contamination
optimized internal geometry- tread and roller compliance	reduces the tension on the rollers, reduces the risk of tilting the roller, creates favorable traction
2-part cage- made of brass, mechanically processed	reduces operating forces

Working to solve the problems associated with cylindrical roller oscillating radial bearings, engineers have found a new solution for three-point bearing turbines, represented by the double-row tapered roller radial thrust bearing, mounted with preload. In field studies, the double-row tapered roller axial radial bearing had reduced wear, induced deformation/load in the gear reducer with lower gears (no additional load on the torsion arms), and increased system stiffness [15], [20].

III. Rolling Elements Used In Load-Bearing Structures

A. *Stress analysis of the load-bearing structure*

In this paper the stress analysis of the load-bearing structure for the axial radial bearing with straight rollers with single ring [8], [21] was tested (Fig.3).

Fig. 3. Axial radial bearing with straight rollers with single inner ring

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Input data: inner ring bearing, diameter d = 90mm; outer ring diameter, D = 180mm, bearing cage, 130 mm diameter, - 10 rollers/cylinders with a diameter of 12 mm evenly distributed radially. Material used: structural steel [21]. It was imposed as a condition for the initiation of the simulation that the inner ring does not move (Fig. 4)[21].

Fig. 4. The shape of the row axial straight roller bearing

The shape of the bearings was modeled axially with straight rollers in a row.

IV. Finite Element Method And Numerical Simulations

In the following were made numerical simulations using finite element method and ANSYS FLUENT software.

After simulation the following results were obtained:

- The force on the outer surface of the outer ring (Fig. 5)- value of 2.05 • 10^5 Pa and it is uniformly distributed;

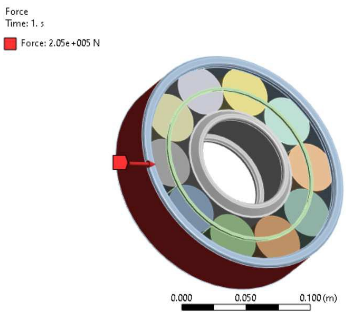

Fig. 5. The force on the outer surface of the outer ring

- The shear stress on outer ring, on the cage and on the inner ring (Fig. 6);

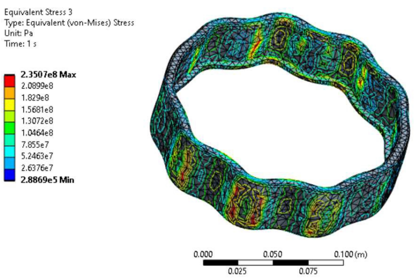

Fig. 6. The equivalent stress on outer ring

- The equivalent tension (von Misses) on rollers.

After the simulations, the following aspects can be concluded:

- The rollers bear the highest tension compared to the rings or the cage (the maximum value obtained 2.3. 10^8 Pa vs. 3.3.10^8 Pa; this indicates that the rollers must be made of a better/better performing steel than the other elements.

- The total deformation has a value of 0.06 [mm] and appears on the outer ring (Fig 6.1);

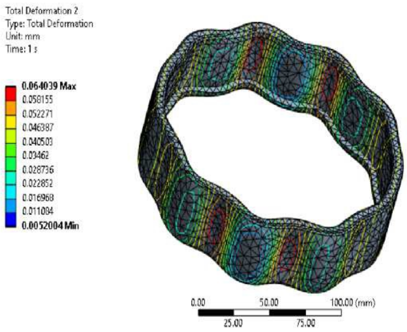

Fig 6. Total deformation on outer ring

- The lifetime of the bearing, expressed in number of cycles, is at least minimum 540 cycles (minimum on cage) up to 10^6 cycles ; after this value the break appears. Goodman mean stress theory was used [9] .

Goodman equation is:

where σ – local stress [MPa]; S – nominal elastic stress [MPa]. The lifetime of the bearing, expressed in number of cycles, is at least 2200 cycles (Fig.7).

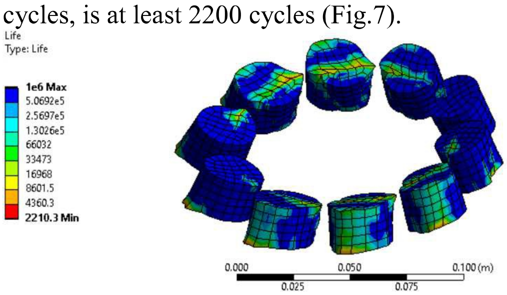

Fig. 7. Bearing life simulation

Since the rollers bear the greatest stresses during the operation of the turbine, compared to the ring or the cage, it is recommended that the material for them be better steel than the other elements (Fig. 8). Experimental research has shown that the use of such a double-row radial tapered roller thrust bearing, pre-stressed during assembly, reduces the axial load in the gear speed multiplier by 67% compared to the double-row tapered roller radial bearing [14], [19], [21], [25], [26].

Fig. 8. Comparison between axial radial bearing with two-row tapered roller and axial oscillating roller bearing with two rows [14]

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Al-based ceramic materials undergo intense wear processes at temperatures higher than $800\,^{0}$ C.

V. Lubrication of Blade Bearing in Wind Turbine

A simplified simulation of oil lubrication/circulation inside the bearing was made in ANSYS Fluent with the following assumptions

- Thermal expansion is not taken into account;

- Bearing vibration is not taken into account;

- The rollers rotate around the axis of the bearing, not around their own axis;

- The rollers inside the bearing are positioned symmetrically at a distance from the inner and outer rings (which in reality does not happen due to vibrations).

A. Lubrication simulation

- Input data: inner ring bearing, diameter d = 90mm; outer ring diameter, D = 180mm, bearing cage, 130 mm diameter, - 10 rollers/cylinders with a diameter of 12 mm evenly distributed radially (Fig. 9).

The oil fills and lubricates the bearing and does not flow due to the caps between the outer and inner ring [23].

Fig. 9. The shape of the chosen bearing with cover .

The mathematical model of the simulation is the turbulent k-epsilon model and it based on 3 elements: turbulent kinetic energy k (1), dissipation epsilon (2) and the energy (3):

(1)

(2)

(3)

where

(4)

(5)

(6)

(7)

with k_{eff} - effective conductivity; J_j - fluid diffusion flux j; S_h - the heat due to chemical reaction; α_i - is the volume fraction of the phase i.

Equation (3) becomes:

(8)

where h is enthalpy;

Equation (9) is for ideal fluids and (10) for real fluids:

; (9)

; (10)

where

. (11)

If the simulation does not take thermal variations into account, (3), (8), (9) and (10) are ignored.

B. Results of simulation

Simulations were made for 15 rpm (Fig. 10) and 700 rpm (Fig.11), with the rollers inside the bearing symmetrically positioned at a distance from the inner and outer rings:

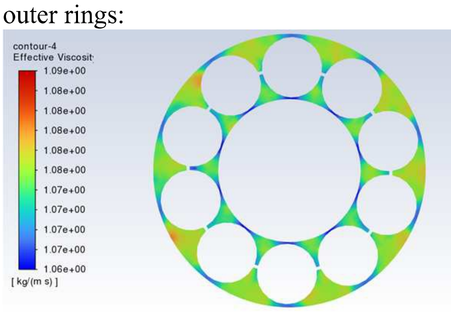

Fig. 10. The dinamic viscosity inside the bearing-15 rpm

In Fig. 10 an increase of 3% of dynamic viscosity can be observed on the inside the bearing neat the outer ring due to the increase in speed up from 0.06 m/s to 0.135 m/s and implicitly the dynamic pressure up to a value of 8 Pa.

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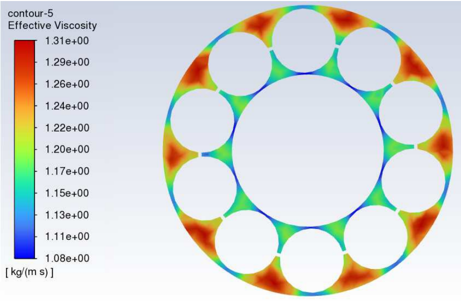

Fig. 11. The dinamic viscosity inside the bearing-700 rpm

In the Fig.11 an increase of 25% of dynamic viscosity can be observed on the inside the bearing near the outer ring due to the increase in speed and implicitly the dynamic pressure. The speed gradient increase from approx. 3 m/s to 6.5 m/s and maximum dynamic pressure is about 17 800 Pa.

VI. Blade Bearing Defects

The maintenance research study was carried out in 3 wind farms in the south-east of Romania, consisting of GE 2.5 MW-100 turbines. The turbines were put into operation in 2010. In the meantime, blade bearing defects [23] were observed in 7 of the 8 turbines.

The type of bearings analyzed is *Double Row Ball Bearings* (the inner ring is bolted to the blade flange and the outer ring is bolted to the hub).

For the laboratory investigation of a segment of the damaged bearing a segment of inner ring, outer ring, cage and balls (Fig. 12 to Figure 15) samples were taken that also included areas with less severe damage to perform the hardness measurement.

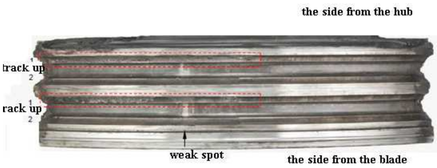

Fig. 12. Inner ring segment (top and bottom halves of raceways 1 and 2, in red - areas with major defects)

Fig. 13. Outer ring segment (top and bottom halves of raceways 1 and 2, in red - areas with major defects)

Fig. 14. Cage segment

Fig. 15. Ball patterns

On both raceways (upper and lower) of the outer ring, the blade-side raceway halves (marked "2" in Fig.12 and Fig. 13) are severely damaged by a pronounced crack. This damage is limited to a circumferential portion of the segment (dotted red line in Fig. 13). Also were analyzed cage segment (Fig.14) and ball patterns (Fig. 15).

The cage (Fig. 14 and Fig. 16) shows severe abrasion and circumferential grooves in the area of contact with the annular edges due to friction. The pockets of the cage show wear from contact with the ball. The balls have a regular surface (Fig. 15), no scratches or deformations are visible.

Fig. 16. Balls and cage (the cage has severe abrasion and detachment of material on the circumference)

In the zone of pronounced rupture (Fig. 17), the outer edge of the track halves "2" is deformed and the split extends to the edge. The opposite track halves (marked "1") show less damage, nicks and leveling of the surface with some small cracks.

Fig. 17. Outer ring-tread with severe exfoliation

At the inner ring, exfoliation was observed in the halves of the raceways (Fig. 18) and weak points on the lower raceway (Fig. 19).

Fig. 18. Inner ring-exfoliation in the halves of the raceways

Fig. 19. Inner ring- weak points on the lower raceway

Hardness measurements were also carried out for the outer ring and balls and it was found that the difference in hardness of the surface of the raceway and the balls is approx. 6 HRC and not in the range of 8 HRC as stated in the standard [19], [23].

The outer ring and ball investigated samples comply with the material specification [20], [24]-[26]. The microstructure of the ball and the outer ring, as well as its purity, meet the required quality. The hardness and surface hardness depth of the outer ring is in accordance with common guidelines. The hardness of the ball is in the upper range. However, the difference in hardness is not considered to be a contributing factor to the deformation damage.

Aspects related to the influence of vibration on the reliability of mechanical systems in the wind turbine and early failure prediction remain of interest for future studies and research [20], [21], [22],[25], 26] .

VII. Conclusions

The global interest in increasing the reliability of complex mechanical systems, implicitly of wind turbines, leads to the concern of finding the best performing solutions for each component sub-assembly of the complex system.

Thus the market demand motivates the development of new technical solutions to replace the standard radial radial bearings in turbines with three-point bearings, such as wear-resistant radial roller bearings and double-row tapered roller axial radial bearings, pre-stressed.

The present paper tries to touch some of the influences that affect the optimal operation of wind turbine bearings and highlight reliable solutions for this field.

Thus it was highlighted that the lubrication of the bearings has a defining impact in their sustainable operation as well as the complex stress mode of the bearings, as a result of the influence of atmospheric conditions on the operation of the wind turbine.

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