Bartosz BASTIAN*, Rafał GAWARKIEWICZ*, Michał WASILCZUK*

ANALYSIS OF THE POSSIBILITIES OF HIGH SPEED SHAFT BEARING SYSTEM DURABILITY INCREASE

ANALIZA MOŻLIWOŚCI ZWIĘKSZENIA TRWAŁOŚCI ŁOŻYSKOWANIA WAŁU SZYBKOOBROTOWEGO PRZEKŁADNI TURBINY WIATROWEJ

Key words:

wind turbine gearbox, bearing failures, bearing system design, retrofit, increase of durability

Słowa kluczowe:

przekładnia turbiny wiatrowej, uszkodzenia łożysk, projektowanie węzłów łożyskowych, modernizacja konstrukcji, wzrost trwałości

Abstract

During the operation of wind turbines with a gearbox of traditional configuration, a high failure rate of high-speed shaft bearings is observed. Such a high failures frequency is not reflected in standard bearing durability calculation methods, which can be attributed to atypical failure mechanism. To avoid observed problems in the 1.5 MW wind turbine, the modification of the existing bearing system is proposed. Multiple options, utilizing various bearing

* Faculty of Mechanical Engineering, Department of Machine Design and Automotive Engineering, Gdańsk University of Technology, ul. G. Narutowicza 11/12, 80-233 Gdańsk, Poland.

types were investigated. Different versions were examined for a potential durability increase, the extent of necessary design modifications, and possibility of solving existing problems in operation.

INTRODUCTION

According to the data presented in **Fig. 1** (1), the most common wind turbine gearbox failures are those of the third – high-speed (HS) stage, being responsible for 64% of all failure cases (1).

The consequences of gearbox failures are very expensive, and numerous research institutes are conducting wide range studies aimed at solving existing issues. One of the elements of these studies is the analysis of data of occurring failures. In the observations made by other researchers, the following situations were observed:

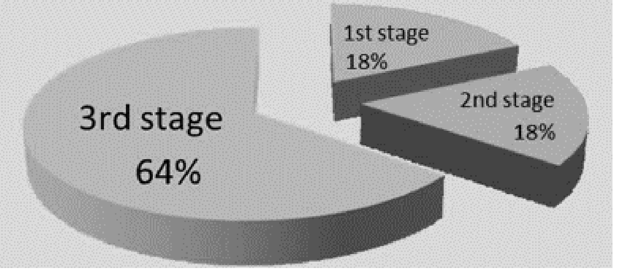

Fig. 1. Failure rate of 3MW PL-PL-PA gearbox (1)
Rys. 1. Awaryjność przekładni 3MW PL-PL-PA turbin wiatrowych (1)

- Most of the turbine gearbox issues are generic in nature and are not related to any specific wind turbine or gearbox manufacturer. Failures can be connected with converged construction solutions. Therefore, failures may be associated with design assumptions rather than with calculation methods used for evaluation of the durability of individual components (2).
- In most of the cases, gearbox failures do not originate as gear failure or gear-tooth design deficiencies, but as bearing failures. The observed failures appear to initiate at specific bearing locations, and later advance into the gear teeth as bearing debris and excess clearances cause surface wear and misalignments. Transmission failure due to gear related quality issues is assessed at up to 10% (2).
- Most of gearbox failures originate from bearing failures, even using the best bearing design practices and calculations of rating life, and the results are not reflected in field data of existing mechanisms (2). Existing failures show an atypical failure mode for the bearings, which not taken into account in considered calculation methods (3).

The high failure rate of bearings mounted on high-speed shafts is also confirmed by data collected by NREL presented in **Fig. 2** (4).

The aim of this paper is to propose modifications to the existing bearing system of high-speed shafts of 1.5 MW wind turbines in which bearing failures were found to appear more frequently than are assessed by rating life calculations (3).

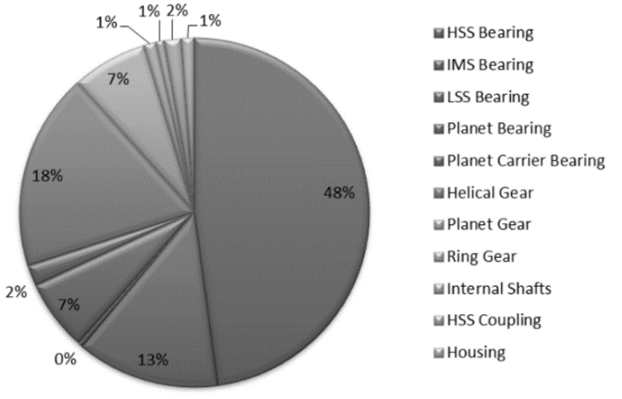

Fig. 2. Failure rate of particular elements of wind turbine (4)
Rys. 2. Częstotliwość uszkodzeń poszczególnych elementów turbin wiatrowych (4)

GEARBOX STRUCTURE

The analysed gearbox is of typical configuration for wind turbine gearboxes. It contains one planetary stage and two stages of helical gears. The kinematic diagram of the gearbox is presented in **Fig. 3**. The slow speed input shaft is an integral part of the carrier of the planetary gears, with outer ring (z_7), which is

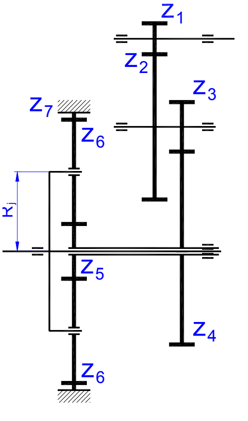

Fig. 3. Kinematic diagram of the gearbox (5)
Rys. 3. Schemat przekładni zębatej (5)

part of the housing, with the use of planetary gear (z_6) that transmits torque to the central wheel (z_5) connected with a hollow slow speed shaft. On the same shaft, helical gear z_4 is mounted. From this gear, torque is transmitted to gear z_3 of an intermediate shaft (IMS). High-speed shaft (HSS) is driven by gear z_2 of the IMS shaft meshed with pinion z_1 that is integral part of HHS. Technical specifications of the gearbox are presented in **Table 1** (5).

Table 1. Data concerning gearbox and separate gears (5)
Tabela 1. Dane dotyczące przekładni oraz kół zębatych (5)

gear:		z_1	z_2	z_3	z_4	z_5	z_6	z_7
No. of teeth	[-]	25	116	26	102	22	41	-104
Ratio	[-]	4.640^{-1}		3.923^{-1}		5.727^{-1}		
Total ratio	[-]	104.254^{-1}						
Power	[kW]	1660						
Rotational speed	[rpm]	17.3/1800						
Lubricant	[-]	synthetic Mobilgear SHC XMP 320						

Figure 4 presents a high-speed shaft with an existing bearing arrangement. The shaft is located on three bearings: first, on the left hand side, there is a cylindrical roller bearing (designated as U – Upwind), providing radial support; and, on the right hand side, there is system of two bearings: a second cylindrical roller bearing (D- Downwind) and four-point contact bearing used as a locating bearing. In operation, it was found that the downwind cylindrical roller bearing has the highest failure rate, which not consistent with the results of calculations of the basic rating life. Rating life calculations were presented in (3).

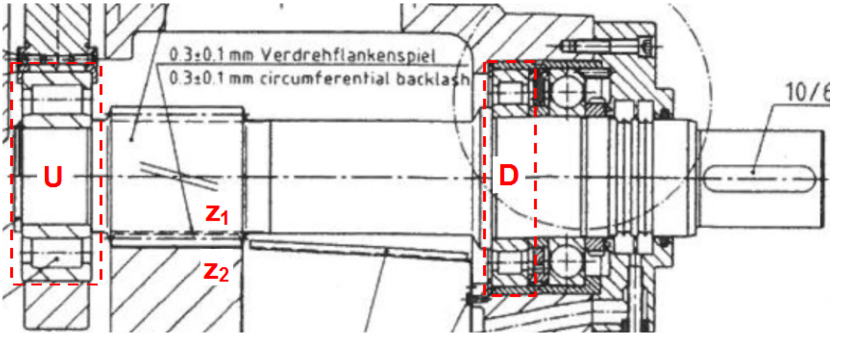

Fig. 4. Existing bearing system of high speed shaft (5); z1 and z2 – gears of: high speed and intermediate speed shafts; sides: U – Upwind and D – Upwind
Rys. 4. Istniejące rozwiązanie łożyskowania wału szybkoobrotowego (5); z1 i z2 – koła zębate wałów: szybkoobrotowego i średnioobrotowego; strony: U – „pod wiatr" i D – „z wiatrem"

EXISTING ISSUES, POSSIBLE SOLUTIONS

According to previous examination (3), some of existing bearing damage can be classified as (micro)pitting and spalling. Such failure modes can occur in the case when rolling elements lose contact with bearing races. This can occur in wind turbines during the change of rotational speed, especially during its decrease, when, for short periods of time, load in the gear can decrease to zero. In such conditions, bearing rolling elements start sliding with respect to the race instead of roll. Sliding of the bearing elements is a widely recognized reason for bearing failure. To avoid such situations, the use of preloaded pairs of tapered roller bearings is recommended (6). Other solutions also will be considered, according to information contained in **Table 2** (7).

Table 2. Bearing types used in wind turbine gearboxes (7)
Tabela 2. Typy łożysk wykorzystywanych w przekładniach (7)

Area	Bearing Type	
High-speed shaft	Fixed-side	SBR, CBR, TBR, BB, 4PCBB
	Free-side	SRB, CRB, BB
Intermediate shaft	Fixed-side	SBR, CBR, TBR, 4PCBB
	Free-side	SRB, CRB
Low-speed shaft	Fixed-side	SRB, CRB
	Free-side	SBR, CBR, FCCRB
Planetary gear	SRB, CRB, FCCRB, TRB	
Carrier	FCCRB, SRB, TRB	

SRB – Self-aligning Roller Bearing, CRB – Cylindrical Roller Bearing,
FCCRB –Full Complement Cylindrical Roller Bearing, TRB – Taper Roller Bearing,
BB –Deep Groove Ball Bearing, 4PCVBB – Four contact ball bearing

ANALYSIS OF POSSIBLE SOLUTIONS

To compare the suitability of the application of individual bearing types, the basic bearing rating life equation (according to ISO 281:1990 (8)) was used (under assumption of 90% reliability), not considering special material selection, or dynamic load conditions. Additionally, operating temperature was assumed not to exceed 120°C. For such conditions, the rating life equation expressed in operating hours takes the following form:

$$L_{10h} = \frac{16660}{n} \cdot \left(\frac{C}{P} \right)^p$$

Maximum rotation speed of the shaft is set at 1800 rpm. The highest radial and axial load for the studied bearing arrangement is 33.3 kN and 26.2 kN, respectively. It is worth noting that the proportion of axial to radial load, in this case, is substantial, which supports the use of bearings suitable for this load ratio.

In the use of taper roller bearings, to avoid complete unloading of one of the bearings, it is advisable to apply preload in such systems. The preload value, assuming bearings of similar dimensions and having similar stiffness, should be set (acc. (9)) to half of the existing axial load. Therefore, axial load for the taper roller bearing used in calculations will be increased by 13.1 kN.

Additionally, the variable load value acting upon the system, depending on the wind speed, was considered. Based on the wind speed data in West Pomeranian voivodeship (**Fig. 5**) (10), and the power of wind turbine as a function of wind speed (**Fig. 5**) (5), five working ranges were assumed and presented in **Table 3**.

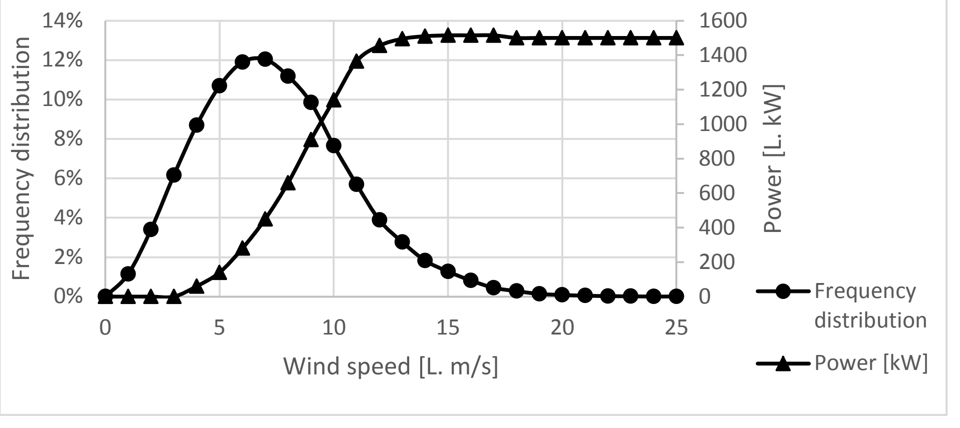

Fig. 5. Wind speed data for Western Pomeranian voivodeship at height of 50 m (10) and power curve for FL MD-77 wind turbine (5)

Rys. 5. Rozkład prędkości wiatru na wysokości 50 m dla obszaru woj. zachodniopomorskiego (10) oraz krzywa mocy dla turbiny wiatrowej FL MD-77 (5)

Table 3. Impact of wind speed on wind turbine load

Tabela 3. Wpływ prędkości wiatru na obciążenie turbiny

Wind speed [m/s]	**Turbine output**	**percentage of time [%]**
0-4	0% – turbine stopped	18
4-7	25%	32
7-10	50%	29
10- 11	75%	11
over 11	100%	10

Table 4. Calculation data and results of bearing systems taken under consideration
Tabela 4. Dane do obliczeń oraz wyniki rozważanych zespołów łożyskowych

Type and bearing designation		e	X	Y	C_r [kN]	L_{10h} [years]	Remarks
D	Existing cylindrical roller bearing D (subjected to radial load only) NKE NU230–E–M6–C3	n/a	1	0	510	54.15	-
U	Existing cylindrical roller bearing (U) – operating correctly	n/a	1	0	930	14,05	-
A	Spherical roller bearing 23230	0.25	0.67	3	1010	28.89	4
B	Bearing pair of taper roller bearings 32230-A + 30230 A	0.44	0.4	1.38	740	12.75	5
C	Bearing pair of taper roller bearings 2x 32230-A	0.44	0.67	2.32	1269	13.68	2, 3
E	Bearing pair of taper roller bearings 2x 31330-A	0.83	0.67	1.23	1355	87.78	2, 3
F	Self-aligning ball bearing 1230-M	0.22	0.65	4.49	180	20 [h]*	1,4
G	Spherical roller bearing 22230	0.33	0.67	4	1280	6.08	1, 4
H	Bearing pair of taper roller bearings 2x 30230-A	0.44	0.67	2.32	798	2.91	1

Remarks
1 – low rating life
2 – dimension issues/limited functionality of construction parts
3 – need of considerable design modification
4 – not preloaded bearing – risk of further existence of the sliding problems
5 – feasible solution
* – due to very low value, unit of hour was used

In **Table 4**, data of investigated bearing arrangements are presented with the results of rating life calculations in comparison to the existing design. The life rating of the existing cylindrical roller bearing (U) should be considered as the expected value of rating life of various proposed design versions. Keeping in mind that in real operation cylindrical roller bearings (U) displays an acceptable durability and that the bearing arrangement at the generator side (D) is subjected to premature failure, which is not reflected by calculations. This situation, when the observed durability is different from predictions is an indirect proof that the reasons of failure are different from those considered in standard calculation methods.

Results presented in the table show that a self-aligning ball bearing (F) is a highly inadequate solution. The rating life of one of the spherical ball bearings (G) and one of the pairs of tapered roller bearings (H) is also not suitable, as its durability is lower than the one of bearing U. The use of these design versions may lead to frequent need to change of the bearings.

Spherical roller bearing (A) and two sets of tapered roller bearing (B and C) exhibit rating life lower than the one of existing bearing set D, but it is greater or comparable with values of rating life of bearing U – allowing to further investigate cases. The rating life of tapered bearing set (E) is significantly higher than other versions, and it is almost twice as high compared to the existing design.

INSTALLATION CONSIDERATIONS OF SELECTED BEARING SETS

Except the rating life calculation, during selection of version, other factors should be taken into account. One of these is the consideration of existing mounting conditions and the dimensions of other parts of the unit. The cost of the gearbox design modification imposes the use of the versions characterized by little or no necessary changes to the housing, if possible. Design changes should be limited to small bearing mounting elements, such as the bearings' bushings, gaskets, or housing cover. Figures below present considered bearing set configuration, with minimal housing adjustments. Considering those guidelines, solutions were proposed **(Fig. 6)**.

As it can be seen, only one solution (A) with spherical roller bearing (**Fig. 6a**) fits into the housing without any modification concerning the housing and the cover, and the design only requires using an additional distance bush.

In case (B), incorporating a set of two tapered roller bearings of different size (**Fig. 6b**), there is an expected collision between the bearing lock nut and the adapter sleeve with the housing cover. However, a change in the shape of the housing cover can be done, so the mounting will be possible. It is also possible to accomplish this mounting by designing a special bearing lock nut and an adapter sleeve to avoid interference with the cover. Bearings in X arrangement are shown in the figure. The O-type arrangement is not presented, because an increased axial length of such a bearing arrangement would make the use of a lock nut impossible, and this would not ensure proper operation of the flinger seal arrangement.

Version C, which is a pair of 32230-A bearings, cannot be used without substantial changes. With the bearings arranged in X arrangement, the bearing lock nut is improperly mounted. (It is probably fixed only on 1-2 threads (**Fig. 6c**)). Furthermore, the bearing from the side of the lock nut is partially mounted on the threaded area itself, which will not provide proper fit and the nut interferes with the oil flinger. Bearings in O arrangement would result in even worse conditions due to increased axial length. In the figure, the housing cover is not presented, due to other existing conflicts.

The version E, utilizing a pair of bearings 31330 (**Fig. 6d**) provides substantial increase in calculated rating life, in comparison to the existing

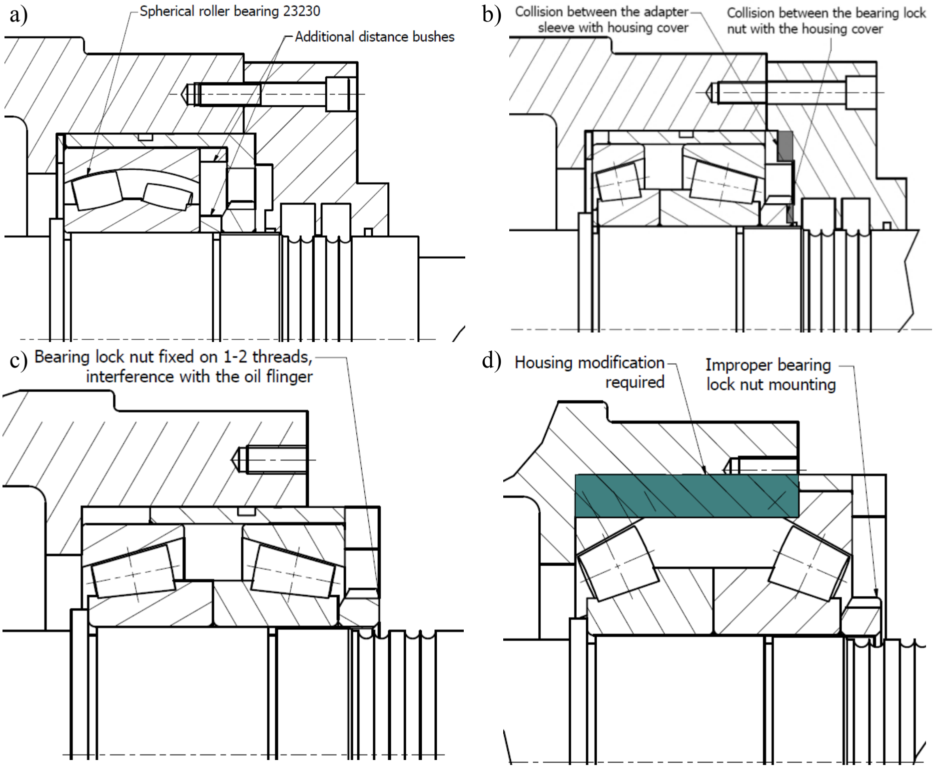

Fig. 6. a) Spherical roller bearing 23230 – version A, b) Taper roller bearing pair 32230-A, 30230-A, X arrangement – version B, c) Taper roller bearing set 2x 32230-A, X arrangement – version C, d) Taper roller bearing set 2x 31330-A – version E

Rys. 6. a) łożysko baryłkowe 23230 – propozycja A, b) zespół łożysk 32230-A i 30230-A, układ X – propozycja B, c) zespół łożysk 2x 32230-A, układ X – propozycja C, d) zespół łożysk 2x31330A – propozycja E

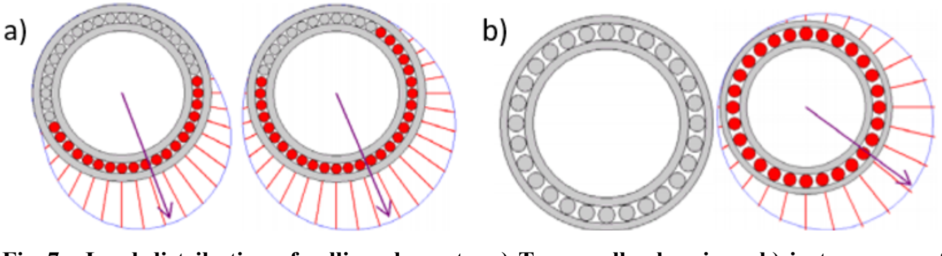

Fig. 7. Load distribution of rolling elements: a) Taper roller bearings, b) in two races of spherical roller bearing (left – upwind row, right – downwind row) (11)

Rys. 7. Schemat rozkładu obciążenia elementów tocznych: a) w parze łożysk stożkowych, b) w obu rzędach w łożysku baryłkowym dwurzędowym (po lewej – łożysko upwind, po prawej – łożysko downwind) (11) assembly. As seen in the figure, there exists a need to enlarge the outer diameter of the bearing set, involving housing modification. Taking into account the need of using an adapter sleeve, inside which bearing would be mounted, the interference with the existing design elements are likely to occur, e.g., visible threaded holes. Moreover, the problems existing in previous cases are still present: improper lock nut mounting, mounting the bearing on the threaded part of the shaft, and improper operation of the oil flinger.

According to the literature, temporary unloading of the bearing, causing sliding of the rolling elements with respects to the bearing race, is mentioned as the origin of the existing problems of wind turbine gearbox bearing failures due to (micro)pitting or spalling. Because of this, design versions with simultaneous load on both bearings in the arrangement are preferred. **Figures 7a** and **b** show the load distribution on the bearing races subjected to the axial load of tapered roller bearings (a) and spherical roller bearings. Distribution of load with axially loaded bearings indicates the advantage of the use of preloaded taper roller bearings.

Summary

The bearing system of high-speed shafts in wind turbines exhibits operational problems due to the high ratio of axial to radial load and the low contact angle of the bearings existing on the market. Additionally, it is advisable to use preloaded sets of taper roller bearings to eliminate existing problems, which are believed to result from rolling elements sliding occurring in the bearings with too low loads. Out of presented concepts, the preferred solution is the one of bearing pairs consisting of two bearings of different widths. It can be characterized by acceptable durability and the high probability of the elimination of the existing sliding problem. Necessary modifications of the gearbox are relatively small, and they do not require machining of the housing. The new parts can be prepared earlier and the turbine retrofit can be carried out during the nearest turbine maintenance stop without dismantling the whole gearbox from the turbine nacelle.

REFERENCES

1. Halse C.: Wind Turbine Drivetrain Development, Romax Technology Inc., USA Wind Technical Center, 2012.
2. Musial W., Butterfield S.: Improving Wind Turbine Gearbox Reliability, 2007 European Wind Energy Conference, Milan, Italy, May 7–10, 2007.
3. Wasilczuk M., Gawarkiewicz R., Libera M., Wasilczuk F., Kinal G.: Bearing systems of wind turbines – maintenance problems (in Polish). Tribologia 4/2015(4-2015):187-198 · 10/ 2015.

4. Sheng S.: Report on Wind Turbine Subsystem Reliability – A survey of Various Databases, National Renewable Energy Laboratory, June, 2013, http://www.nrel.gov/docs/fy13osti/59111.pdf .
5. Gawarkiewicz, R., Wasilczuk F., Wasilczuk M.: Drivetrain of a Wind Turbine Machine Dynamics Research, 2015, Vol. 39, No 2, 83–95.
6. Doll G.L., Tribological challenges in wind turbine technology, Wind Turbine Tribology Seminar, Broomfield, Colorado, USA, November 15-17, 2011.
7. Ninoyu N. Yagi S.: Technical trends in wind turbine bearings, NTN Technical Review No.76, 2008, 113–120 .
8. ISO 281:1990. Rolling bearings – Dynamic load ratings and rating life
9. SKF, Preload in ball and tapered roller bearing systems (in Polish). http://www.skf.com/pl/products/bearings-units-housings/roller-bearings/principles/design-considerations/selecting-clearance-preload/bearing-preload/preload-in-bearing-systems/index.html.
10. The International Renewable Energy Agency, Global Atlas for Renewable Energy, http://irena.masdar.ac.ae/?&tool=dtu:gwa&map=103.
11. Marks C.S., Matthew B.T.: Bearing selection techniques as applied to mainshaft direct and hybrid drives for wind turbines, Timken Technical Paper, 2009.

Streszczenie

Podczas eksploatacji turbin wiatrowych o tradycyjnej konstrukcji przekładniowej obserwuje się wysoką awaryjność łożysk tocznych wału szybkoobrotowego. Ta wysoka awaryjność nie znajduje odzwierciedlenia w standardowych obliczeniach trwałości, co spowodowane jest nieuwzględnieniem nietypowego mechanizmu ich niszczenia. W celu uniknięcia obserwowanych problemów w turbinie o mocy 1,5 MW zaproponowano modyfikację istniejącego układu łożyskowania. Rozważano kilka propozycji z użyciem różnych rodzajów łożysk. Przeanalizowano poszczególne rozwiązania pod kątem: potencjalnego zwiększenia trwałości, skali niezbędnych zmian konstrukcyjnych w przekładni, a także ze względu na rozwiązanie istniejących problemów eksploatacyjnych.