

# Analysis of Failures of High Speed Shaft Bearing System in a Wind Turbine

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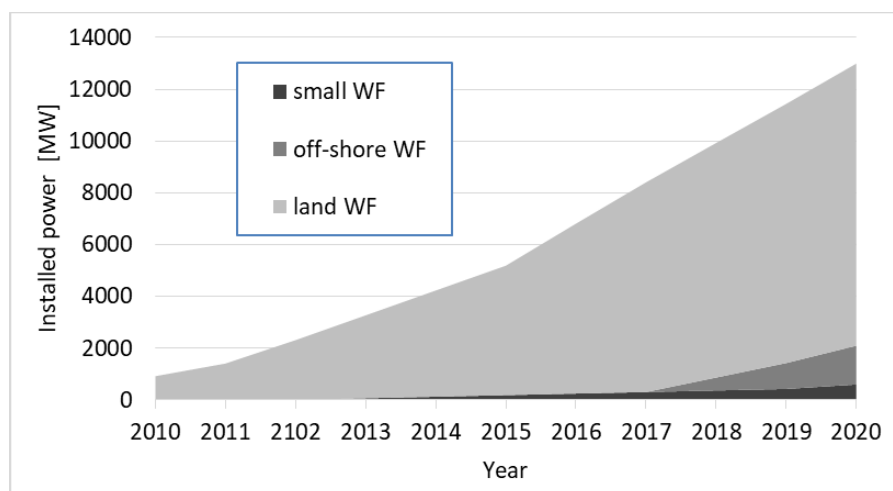
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**Abstract:** During the operation of wind turbines with gearbox of traditional configuration, consisting of one planetary stage and two helical stages high failure rate of high speed shaft bearings is observed. Such a high failures frequency is not reflected in the results of standard calculations of bearing durability. Most probably it can be attributed to atypical failure mechanism. The authors studied problems in 1.5 MW wind turbines of one of Polish wind farms. The analysis showed that the problems of high failure rate are commonly met all over the world and that the statistics for the analysed turbines were very similar. After the study of potential failure mechanism and its potential reasons, modification of the existing bearing system was proposed. Various options, with different bearing types were investigated. Different versions were examined for: expected durability increase, extent of necessary gearbox modifications and possibility to solve existing problems in operation.

**Keywords:** wind turbines; gearbox failures; bearing failures; high speed shaft bearing systems; design modification

## 1. Introduction

In recent years wind power has been the sector with the fastest growth in comparison to other sectors. Average year growth rate amounted to 20% in Europe and 30% in the USA in 2000-2010 decade. Also in Poland a fast growth of installed power has been observed and is expected (Figure 1) [1].

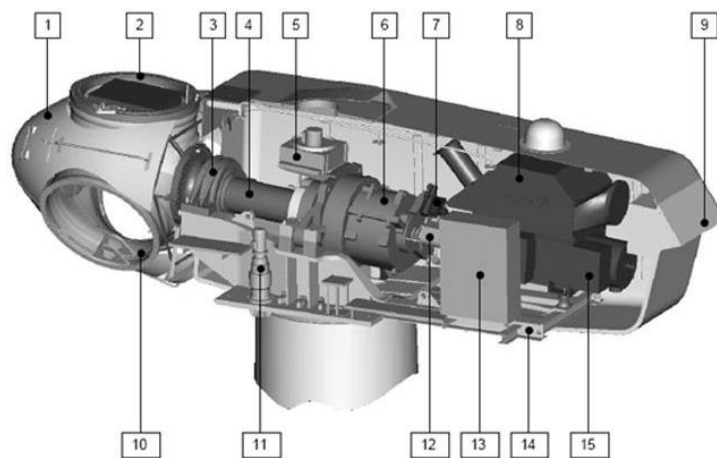


**Figure 1.** Increase in power installed in Polish wind farms [1]



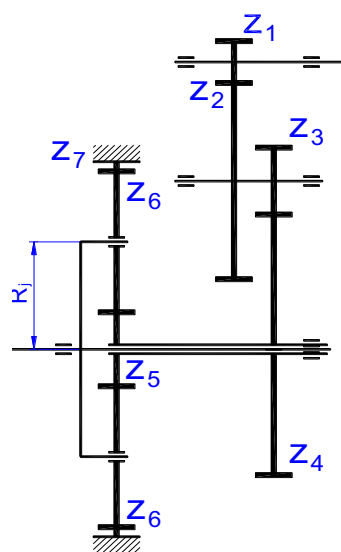
Despite of modern character of wind farms equipment, wind energy sector faces durability problems with serious failures occurring much earlier than theoretical durability assumptions. Scale of the problem can be illustrated by an example of press release in which Siemens in 2014 informed that its profit has been decreased by 54% because of customers claims dealing with inadequate durability of wind turbine components [2]. Also in Poland the owners and operators of wind farms face serious durability problems so they undertake various steps to find sources of problems and improve the situation.

Most of the wind turbines are of gearbox type with the drivetrain configuration such as presented in Figure 2 [3]. Power from the rotor is transmitted to the generator (15) through the rotor shaft (4) – connected with the multiplying gearbox (6) of the ratio of approximately 100, by the friction shaft-hub joint and further by the flexible coupling (12) and composite sleeve (not marked on the drawing). A disc brake (7) is situated at the gearbox output (and at the generator input).



**Figure 2.** Typical configuration of a wind turbine drivetrain [3]

For the analysis of the durability issues it is important to give a brief introduction into the structure of the gearbox, as well. The most common configuration is shown in Figure 3 [4].

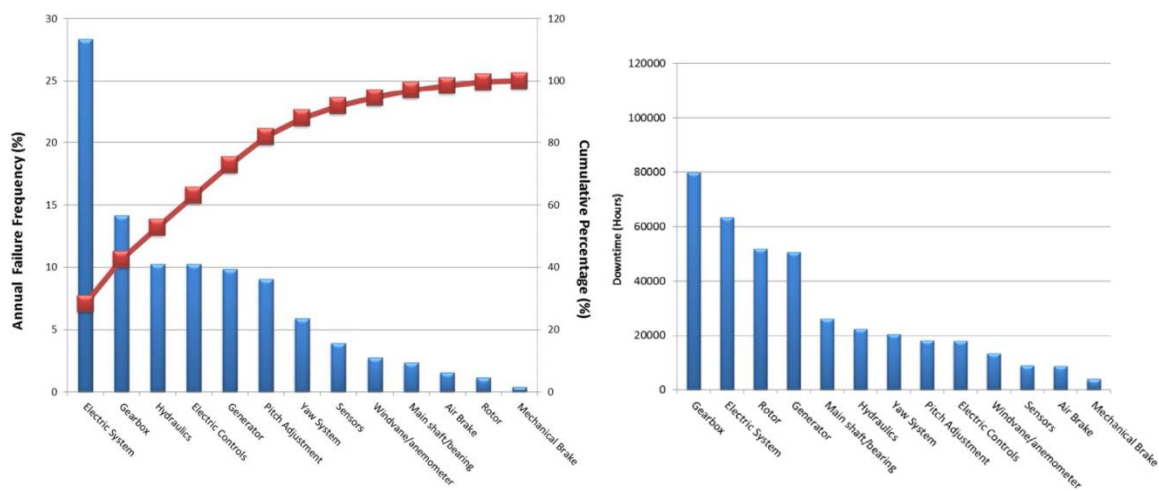


**Figure 3.** Kinematic diagram of the gearbox [4]

It consists of one planetary stage and two stages of helical gears. The slow speed input shaft is an integral part of the carrier of the planetary gears, the outer ring ( $z_7$ ) is a stationary part connected to the housing. With the use of planetary gear ( $z_6$ ) the torque is transmitted to the central wheel ( $z_5$ ) connected with a 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 IMS shaft meshed with pinion  $z_1$  which is integral part of HHS.

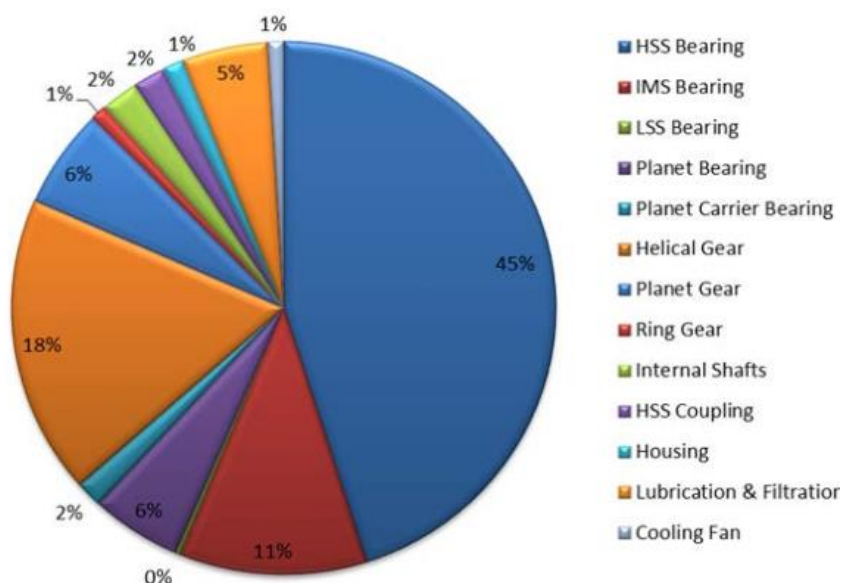
## 2. Failure statistics and mechanisms

The consequences of wind turbine failures are very expensive, and numerous research institutes are conducting wide range of studies aimed at solving existing issues. One of the findings was that, although the failures of electrical equipment failures are the most frequent, the gearbox failures are responsible for the largest downtime and cost (Figure 4) [5]



**Figure 4.** Failure frequency and downtime caused by particular wind turbine component [5]

Out of the gearbox failures these of the high speed shaft (HSS) rolling bearings are most common, which was confirmed by data collected by NREL in 2014 presented at Figure 5 [6].

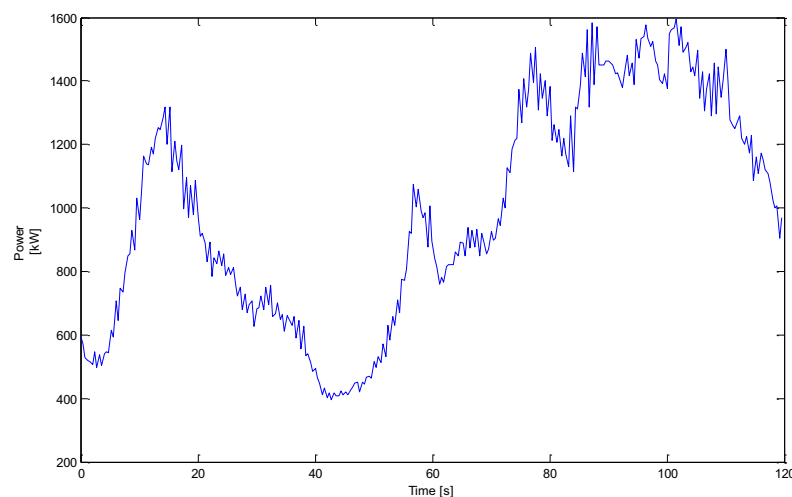


**Figure 5.** Failure rate of particular elements of wind turbine gearbox [6]

In the analysis of data of occurring failures made by other researchers, it was found that:

- 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 [7].
- 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% [7]
- The highest number of gearbox failures originate from bearings failure, even using the best bearing design practices and calculation of rating life, results are not reflected in field data of existing mechanisms [7]. Existing failures show atypical failure mode for the bearings, not taken into account in considered calculation methods [7].

Extremely unsteady operating conditions are characteristic for wind turbines and it is considered to be one of the reasons of frequent failures [8] – Figure 6 shows an example of changes of power generated by a wind turbine with four peaks of over 1000 kW and one drop to 400 kW in just 120 seconds.

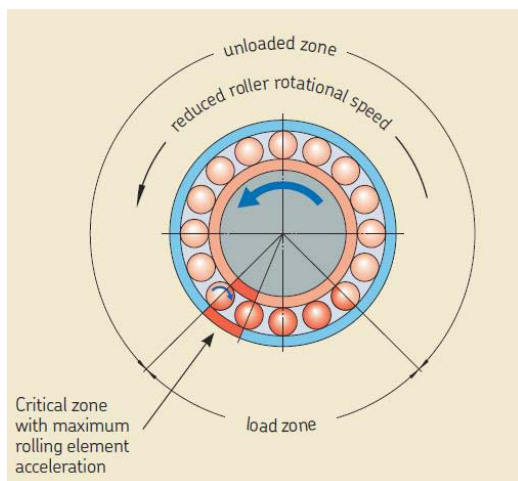


**Figure 6.** Example of power output changed in a wind turbine in 120 s [8]

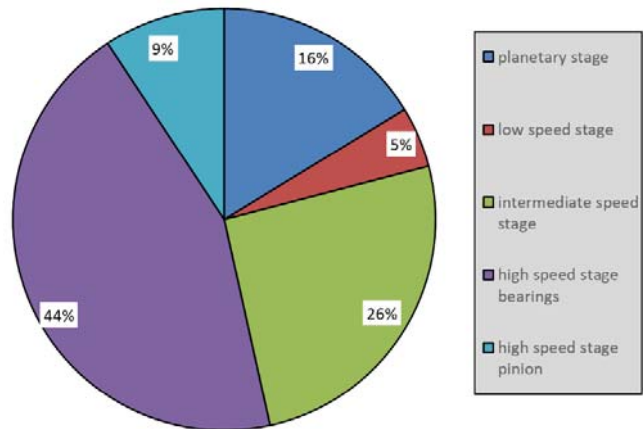
Roller element bearings are vulnerable to the conditions of loading and unloading, which causes sliding of the rolling elements [9]. As explained in Figure 7 [9] the rolling elements leaving the loaded zone are disengaged from the inner raceway due to centrifugal forces and their rotational speed is lowered considerably. Then entering the loaded zone again the elements become accelerated, but their inertia opposes such acceleration, so the elements slide with respect to the raceways and excessive heat is generated causing overheating, structural changes and even adhesion in metallic contact areas. This failure mechanism is often referred to as smearing or adhesive wear. Structural changes of bearing material caused by repeated overheating is also one of the failure modes characteristic for wind turbine bearings, known as WEA (white etching areas) [10]. As pointed out by Erichello [10] there are several hypotheses for the root cause of these failures including not only sliding, but also impact loads, hydrogen embrittlement; electrostatic discharge, corrosion fatigue, and adiabatic shear. None of these hypotheses has been proven, and it is a field of current research.

The sliding of the roller elements affects larger bearings and bearing with high rotational speeds, which is the case of HSS bearings in the gearboxes of wind turbines. Avoiding too loose fits and too low loads are the methods of preventing this failure mode. In the bearing design procedures,

specifying minimum bearing load is the only means of preventing sliding of the rolling elements [11]. In the machines with fluctuating loads fulfilling these specifications may be impossible.



**Figure 7.** Mechanism of bearing failure caused by sliding of rolling elements [9]



**Figure 8.** Percentage of failure of particular turbine elements in the analysed Polish wind farm [4]

The other way in which unsteady operating conditions affect failures of the machines is increased level of machine vibration. Changes in condition monitoring signals due to changes in load can be larger than changes resulting from developing failures, which makes it more difficult to detect them at early stages than in machines in steady operating conditions. The delay in detecting failures increases their extent, downtime and cost.

An analysis of the reasons of failures in 1.5 MW wind turbines installed in one of Polish wind farms presented in Figure 8 showed that the percentage of particular reasons was similar to that observed worldwide, with 44% of the failures occurring in the high speed shaft, mostly in the bearings [4].

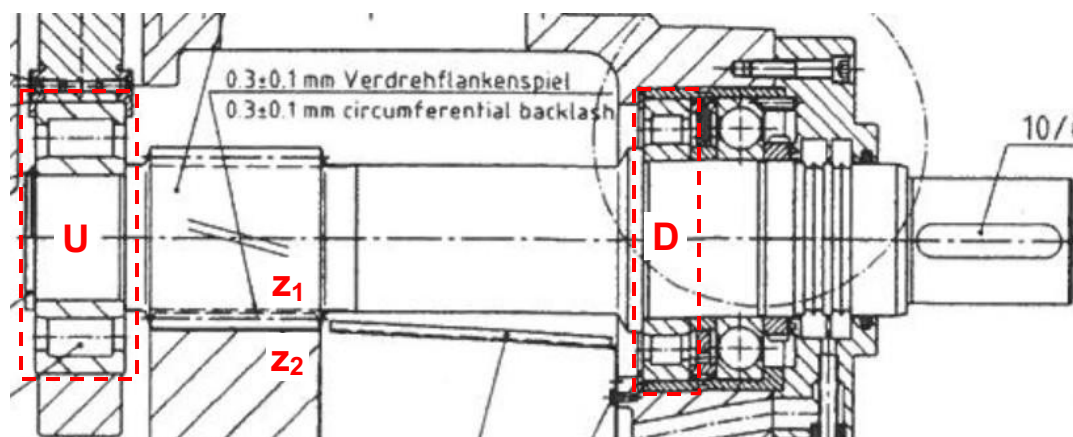
### 3. Current design

Figure 9 shows the high speed shaft with its bearing arrangement of the studied wind turbine. 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 a system of two bearings: a second cylindrical roller bearing (D- Downwind) and four-point contact bearing, playing a role of locating bearing. In operation it was found that the downwind cylindrical roller bearing has the highest failure rate. Larger number of failures in comparison to upwind bearing is not consistent with higher theoretical rating life  $L_{10}$  shown in Table 1.

**Table 1.** Theoretical durability and recorded number of failures of the HSS bearings

	Durability $L_{10}$ [years]	Number of failures
Upwind bearing	2,4	3
Downwind bearing	9,4	16

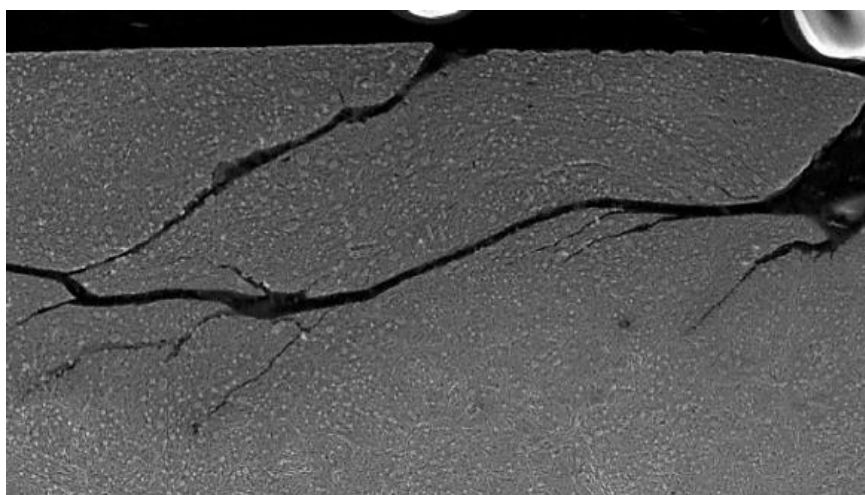




**Figure 9.** Existing bearing system of the high speed shaft in the analysed wind turbine [4]

Inconsistency may be caused by the failure mechanism not considered by the standard bearing durability calculations, i.e. by smearing (adhesive wear) caused by sliding of the rolling elements. Sliding of the rolling elements is the reason of overheating of the bearing rings and structural changes of the bearing material. Plastic deformation of the material (Figure 10), being possible only at elevated temperature was observed in the microscopic image of the destroyed bearing ring [4]. This may indicate smearing caused by sliding of the rolling elements as a possible failure mechanism in the analysed case.

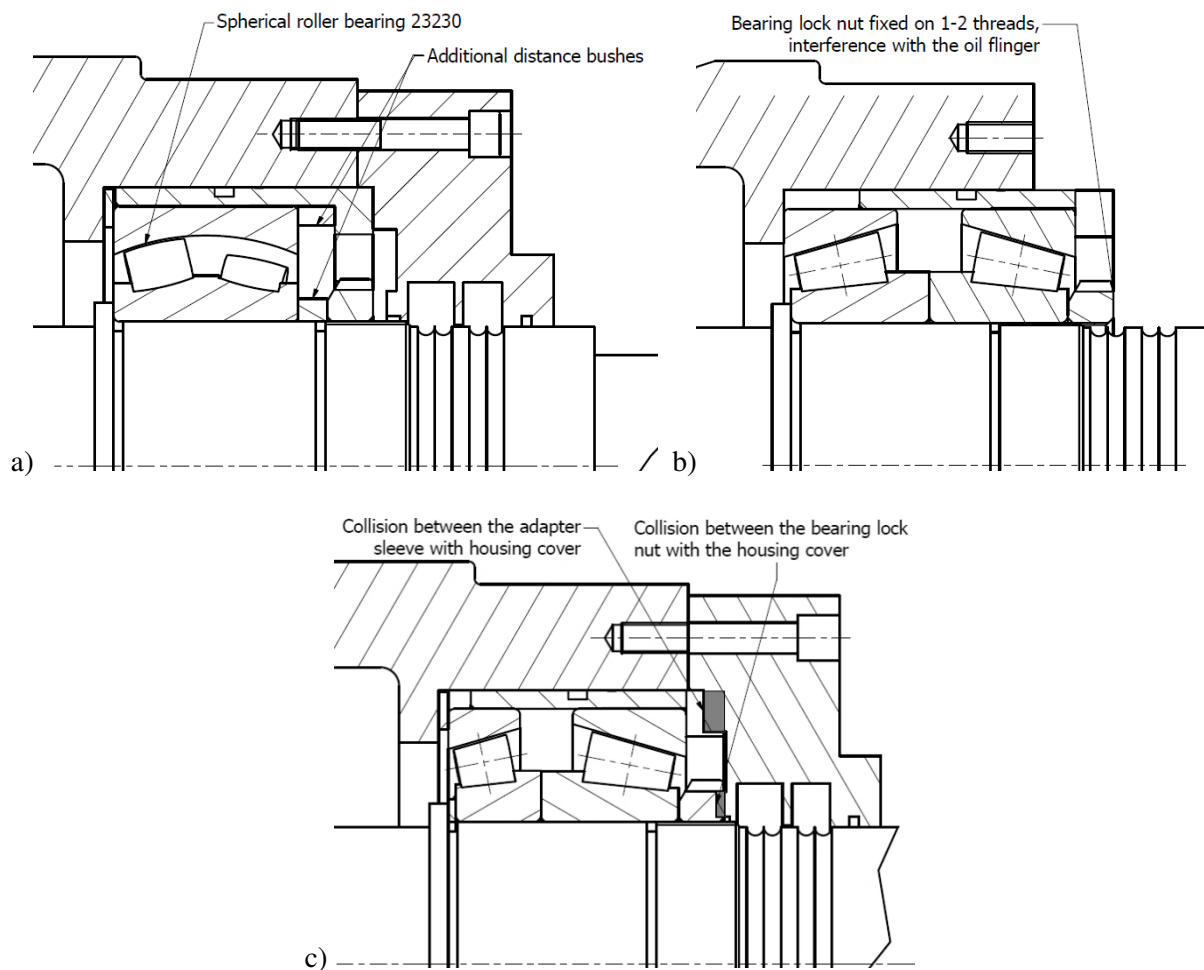
In the literature several remedies are suggested [12], including DLC coated bearing rollers, cleaner steels and lower roughness of bearing rings. It is also pointed out that pre-loaded taper roller bearings exhibit less roller sliding than cylindrical roller bearings.



**Figure 10.** Plastic deformations under the surface of destroyed bearing ring [4]

## 4. Proposed modifications

Because of large number of failures and evidence that smearing is a probable reason of failures a redesign of the bearing system of the high speed shaft was considered. Quite an obvious assumption of the planned retrofit was minimizing necessary modifications of the gearbox design. Especially changes in the gearbox housing, shaft with pinion should be limited. Design changes should comprise small bearing mounting elements, such as bearing bushing, gaskets, housing cover. Various possible bearing configurations were considered utilizing bearings of the outer and inner rings diameters unchanged, i. e. shaft diameter of 150 mm and housing diameter of 270 mm. Three of the proposed versions with satisfactory life rating values are shown in Figure 11 [13]



**Figure 11.** Proposed bearing arrangements: a) Spherical roller bearing 23230 – version A, b) Set of two identical taper roller bearings 32230-A in X arrangement – version B, c) Set of two different taper roller bearings 32230-A, 30230-A in X arrangement – version C [13]

As it can be seen in the figure, only one design (Case A) with spherical roller bearing 23230 (Figure 11 a) fits into the housing without any modification concerning the housing and the cover, this design only requires using an additional distance sleeve, but its use does not provide conditions for limited sliding of the rollers, because in two row spherical roller bearing loaded with substantial axial load and small radial load one row of rollers is unloaded, so despite good theoretical rating life this version was discarded.

Version B - a pair of 32230-A bearings shown in Figure 11 b can not be used because of excessive size in axial direction. Bearing lock nut is improperly mounted - most probably it would be fixed only on 1-2 threads and the nut would interfere with the oil flinger. Furthermore, bearing from the side of the lock nut, is partially seated on the threaded area itself, which will not provide proper fit. At the figure, the housing cover is not presented, due to other existing conflicts.

In case C incorporating a set of two tapered roller bearings of different size, shown in Figure 11 c, there is an expected collision between the bearing lock nut and the adapter sleeve with the housing cover. However, modification of the shape of the housing cover can be done, so the mounting will be possible. Other possibility to accomplish such mounting, is to design special bearing lock nut and an adapter sleeve, to avoid interference with the cover. Bearings in X arrangement are shown in the figure. 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 would not ensure proper operation of the flinger seal arrangement.

## 5. Conclusions

The gearboxes of the analysed turbines demonstrated similar problems to those reported worldwide – high failure rate, especially of the roller element bearings of the high speed shaft, and inconsistency of the rated life predictions with the operational data. The results of the study on wear patterns, including microscopic analysis of the material structure of the broken bearings suggested that smearing and adhesion due to sliding of the rolling elements may be one of the important reasons of premature failures. In such case, according to the literature, the use preloaded sets of taper roller bearings is one of the solutions. Out of presented concepts, preferred version is the one of bearing pair consisting of two bearing of different widths. It can be characterized by satisfactory durability, high probability of 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. Now the retrofit project is on its way and will be implemented in one of the gearboxes in which a major repair will be carried out.

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