



Experimental Validation of Models for the Structural Simulation of Crossed Roller Wire-Race Bearing

Iñigo Martín¹ , Iker Heras²  , and Josu Aguirrebeitia² 

¹ Siemens Gamesa Renewable Energy, 48170 Zamudio, Spain

² University of the Basque Country (UPV/EHU), 48013 Bilbao, Spain

iker.heras@ehu.eus

Abstract. Wire-race bearings are a special type of bearings where rolling elements run over steel wires embedded within the rings, allowing the use of lighter materials for the latter. In previous research works, and motivated by the capabilities of these components and the lack of bibliography on the matter, the authors developed different approaches to simulate their structural behaviour, mainly focused in the flexibility of the system, the load distribution and the static load capacity. In these works, detailed finite element models were used as a reference in order to develop simplified mechanical models of the system and evaluate their capabilities and limitations. Nevertheless, no experimental tests were performed to assess the accuracy of these finite element simulations. In the current work, a test campaign is carried out in order to experimentally validate the previously developed models. As a conclusion, finite element analyses show a satisfactory correlation with experimental results, thus validating the approaches proposed by the authors in previous works.

Keywords: wire-race bearings · crossed roller bearings · experimental testing · finite Element Analysis

1 Introduction

Wire-race bearings are a non-conventional type of slewing bearings, where the rolling element runs over race-shaped wires embedded within the rings. While the wires must be manufactured in hardened steel, this conception of bearing allows choosing a lighter material for the rings, such as aluminium, bronze, carbon fibre, plastic or even 3D printed rings. This involves weight savings up to 60–65% according to [1–3] and the consequent inertia reduction. Figure 1 shows the cross section of two wire bearings with rollers of different diameter, where the mentioned components and the design principle can be observed.

These bearings are available in a wide range of sizes, from few centimetres until two meters [1], and they can face external loads such as axial and radial forces as well as tilting moments, at high or low speed. The election of aluminium rings provides a good performance absorbing shock loads and travelling vibrations since it has a low elasticity

modulus and more damping due to the additional contacts, which leads to a reduction of brinelling and striation in the raceways. Applications where weight savings, low inertia and shock loads are an important issue, the election of wire-race bearings is an option to take into account. Nowadays, some examples of implementation of this kind of bearing can be found in medical equipment, robotics, aeronautics and military applications. Another advantage of the race shaped wires is that they can be placed anywhere, even embedded in the design of the machine. This helps to reduce the mounting space and increases the compactness. This layout provides also maintenance and reparation profits allowing to replace the wires when they suffer race damage instead of replacing the whole bearing.

Despite the patent of wire-race bearings was registered by Erich Franke in 1936, little relevant research about them could be found until late 2010s, which made it difficult to characterize these components or foresee their mechanical response. The first meaningful contributions on the topic were made by Gunia and Smolnicky [4–6] and Dindar et al. [7]. On the one hand, Gunia and Smolnicky focused their work on ball wire-race bearings, analysing the effect of several geometrical parameters on its structural behaviour via Finite Element Analysis (FEA). On the other hand, Dindar et al. studied roller wire-race bearings and the friction sources that actuate on them, developing a methodology to estimate the friction torque under different load conditions.

With this background, the authors started their research in the topic, focusing on the flexibility of the system, the load distribution and the static load capacity. In their first work [8], they made a first approach to the matter by comparing the structural response of conventional and wire-race ball bearings via FEA, identifying the wire twisting phenomenon, i.e. the rotation of the wire cross-section under external load, and its relevance on the performance of the component. In [9] they developed an analytical formula to calculate the wire twisting stiffness, so it could be used in future analytical approaches, like the one proposed in the previous IFToMM congress [10] for ball wire-race bearings. Similarly, but focusing on roller wire-race bearings, the authors proposed a simplified non-linear model in [11], which was later linearized in order to implement in FEA [12]. This way, the approach in [11] can solve the load distribution problem numerically in a fast way under the assumption of rigid rings (considering only local contact deformations), while the FE model in [12] allows considering ring deformations but at the expense of a higher computational cost. Finally, the authors carried out a sensitivity analysis to study the effects of the main geometrical parameters on the performance of both ball and roller wire-race bearings with the aim of offering guidelines to the designers [13].

So far, in all the research works in the field of wire-race slewing bearing analysis carried out by the authors [8–13], detailed finite element models were used as a reference in order to develop simplified mechanical models of the system and evaluate their capabilities and limitations, as well as to study the effect of the design parameters. Nevertheless, no experimental tests were performed to assess the accuracy of these finite element simulations. In the current work, a test campaign is carried out in order to measure the force-deformation behaviour of roller wire-race bearings and thus experimentally validate the previously developed models.

2 Materials and Methods

2.1 Specimens

The design of the bearings to be tested is an important task, since they must be able to provide relevant results and also meet the characteristics of the test bench. An experimental test campaign is expensive and tedious, so everything must be planned in order to obtain the maximum return. In this sense, two specimens of two different bearing designs were considered to be tested in order to obtain more diverse results. Both bearing designs must be similar, since the clamping tools will be the same for both of them; but they must be also different enough to obtain sound conclusions. For this reason, both bearings have the same external ring geometry and the only difference between them lies on the geometry of the wire-roller-wire subset (see Fig. 1). To this purpose, several FE analyses were performed until both bearing geometries were different enough but within the design space considered for the development of the analytical model [12]. The geometry of the rings was obtained with the standard ring geometry defined in [14]. The dimensions for the two considered bearing geometries are summarized in Table 1 where, according to nomenclature in [13], D_{pw} is the bearing mean diameter, D_w is the roller diameter, λ is the wire diameter, Rf is the raceway factor, α is the contact angle, H is the dimension of the housing for the wire-roller-wire set, N_R is the number of rollers and C_{0a} is the axial static load capacity (calculated by the analytical model in [11]). To give a visual representation of the bearing designs, the cross-section of both alternatives are represented in Fig. 1. As pointed out in the table, four bearings were tested, two for each geometry. It is worth also mentioning the layout of the assembly bolts, which were arranged in such a way that the tightening process were easily performed from the upper side.

Table 1. Geometrical data of the tested bearings (according to nomenclature in [13]).

Bearing ID	D_{pw} (mm)	D_w (mm)	λ (mm)	Rf (-)	α ($^\circ$)	H (mm)	N_R (-)	C_{0a} (mm)
1 and 2	400	14	8	0.8	45	19.00	72	652.6
3 and 4	400	16	10	0.4	45	25.56	62	660.3

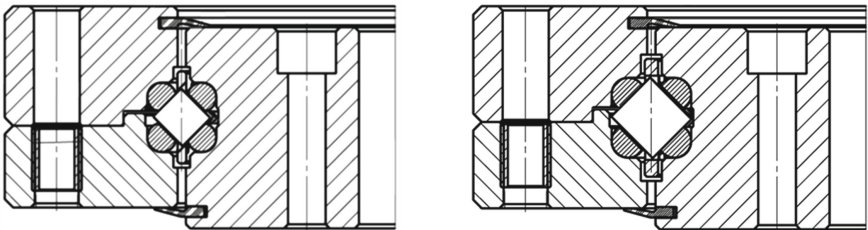


Fig. 1. Cross section of the tested bearings: bearings 1–2 (left) and bearings 3–4 (right).

2.2 Test Bench Setup

The Servosis ME 405-30 test bench was used for the experimental tests. This machine was designed to perform tension-compression tests on different kind of specimens, depending on the installed clamping tools. A picture of the test bench with its main components is represented in Fig. 2. Regarding the test bench features, its mechanical transmission system formed by two spindles can apply a maximum load of 300 kN. The distance between the columns allows to test a bearing with an external diameter up to 600mm. In terms of control, the PCD2K software was developed specifically by the manufacturer for these kind of benches. This software allows performing the tests by the introduction of loads or displacements, depending on the convenience. The load cell placed in the middle of the machine bridge provides the reaction force in real time and the encoder that measure the deformations have a $10\mu\text{m}$ resolution.

Specific clamping tools were designed and manufactured for the bearings under study. Besides, external measurement devices were arranged because of two reasons. First, bearing deformations are very small and the machine measuring system do not have enough resolution. Second, the machine measures the deformation of the whole kinematic chain. The development of the clamping tools may seem a simple task, but they should be as light and stiff as possible. Light tools simplifies the assembly process and the stiffness avoids measurement errors. Besides, any misalignment between the tools can introduce non-desired loads. With all this in mind, some FE analyses were carried out to achieve the final design, which can be seen in Fig. 3. In this figure, the whole test assembly is represented. The lower tool is made of one piece, which is connected to the test machine by means of a male-female joint locked with a pin. This connection is relevant since the load is transmitted through the pin. The upper tool is similar but more complex; it was made of two parts and an axial spherical bearing. The installation of an axial spherical bearing aims to reduce problems caused by lack of parallelism and prevent the introduction of undesirable tilting moments. The three parts that compose the clamping tools were manufactured in AISI 1042 steel finished with a bluing treatment. Regarding the necessary additional measurement equipment to record only the deformation of the bearing, four Mitutoyo IP42 543-470B dial indicators were arranged every 90 degrees, as shown in Fig. 3. These dial indicators had a micrometric resolution and a maximum permissible error of $\pm 3\mu\text{m}$.

2.3 FE Model

The reference FE model used by the authors in their previous research works for the simulation of roller wire-race bearings, and therefore the one to be validated now, is the one described in [13]. The load case to be studied in this research work is a compression axial force. Therefore, and taking advantage of the cyclic symmetry of the system in this case, only one-half of the sector corresponding to one roller is considered. Figure 4 shows the boundary conditions and the mesh for the FE model (symmetry boundary conditions are not represented). Note that bolts are not modelled, since it is assumed that the joint is not going to open or slide.

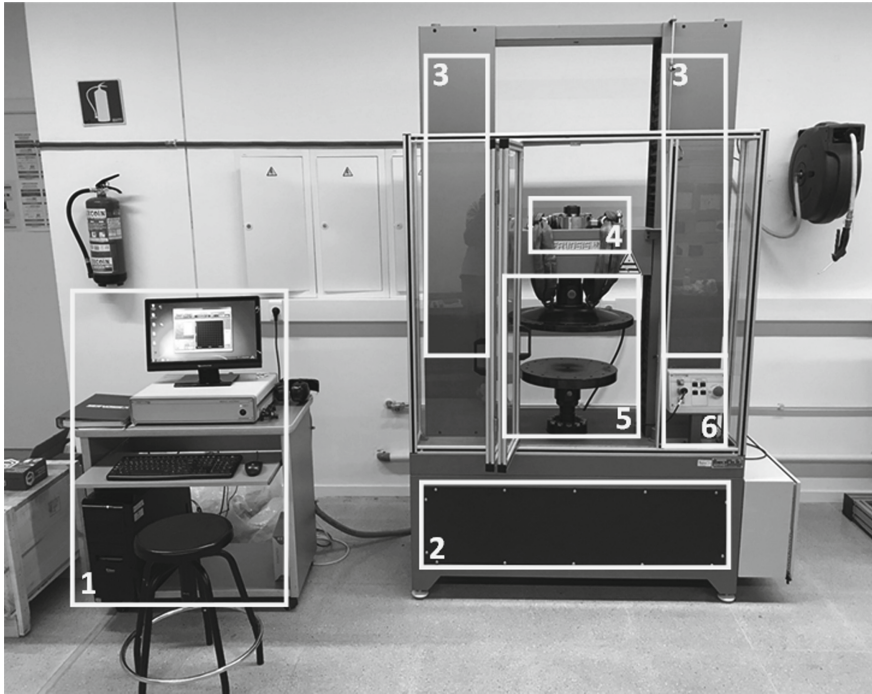


Fig. 2. Servosis ME 405–30 test bench with the clamping tools: 1) controller; 2) motor and transmission; 3) spindles; 4) load cell; 5) clamping tools; and 6) manual control.

2.4 Experimental Test Campaign

Once the bearings and the components of the test assembly are ready, let's define the tests to be performed. First, it is important to study if the bolt preload has influence on the FE and analytical models. For this reason, there were two groups of tests: with no bolt preload and with bolt preload. No bolt preload tests consisted on bolts tightened with a wrench until the eventual tool-bearing gaps were suppressed. The same tests were also performed applying a bolt preload equivalent to the 50% of the maximum allowable preload. M8 and M10 bolts of grade 8.8 were used in the inner and outer rings, respectively. Inner ring bolts were tightened having the bearing simply supported on the lower tool, but the outer ring bolts were tightened while the bearing was supporting a load of 10 kN. This procedure was established to allow the upper clamping tool to accommodate the spherical bearing and recover the centred position before the bolt tightening.

Regarding the sources of uncertainty related with the assembly process, it is important to control irregular or low bolt preload (non-preloaded bolts case) in conjunction with bearing misalignment. In order to control these phenomena, each bearing is unassembled and reassembled in a different position before each test. To evaluate this, it was considered enough to perform three tests for each bearing and tightening methodology; this makes a total of six assemblies per bearing.

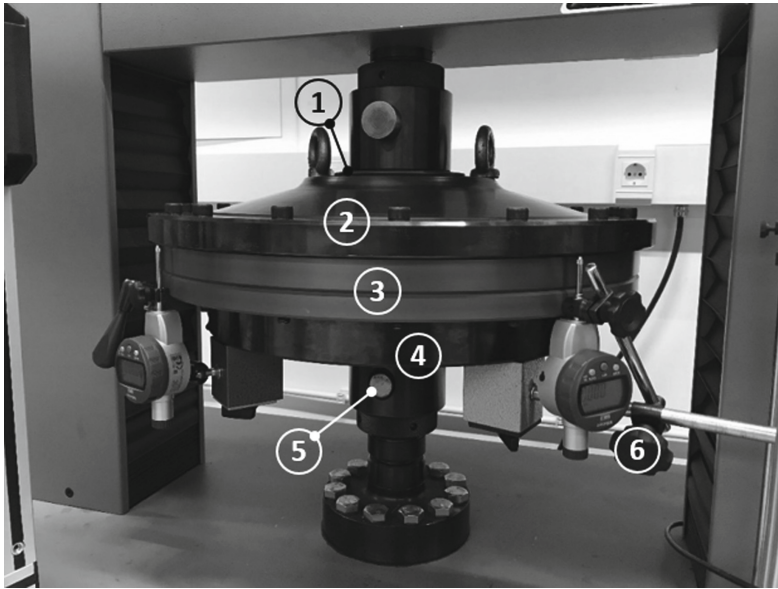


Fig. 3. Test assembly: 1) axial spherical bearing; 2) upper tool; 3) tested bearing; 4) lower tool; 5) mounting pins; and 6) dial indicators.

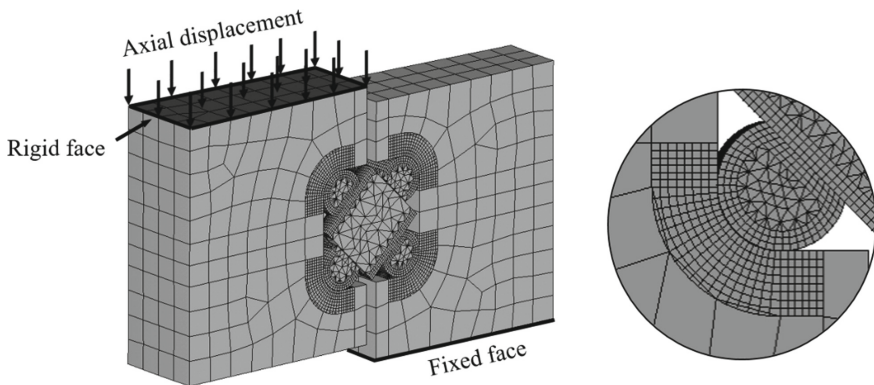


Fig. 4. FE model: boundary conditions and mesh.

Repeatability for each assembly is another phenomenon that has been taken into consideration. For that purpose, each loading process is repeated four times for each assembly. The first loading process aims to accommodate the bearing for the test and the other three to assess the repeatability of the results. The first loading process in all cases retrieved non-realistic results, so it was discarded. This way, possible measuring or loading errors can be easily discriminated. Each loading process consists on introducing a compressive load by steps, reading the deformations on the dial indicators. The maximum load was established on 200 kN, since the tests were conceived to be non-destructive, and

greater loads may generate excessive permanent deformations on the races. However, it is a relevant load value considering that the axial static load capacity of both bearing designs is around 650 kN. Besides, according to manufacturers, these kind of bearings usually do not exceed the 20% of the static load capacity during operation.

Figure 5 offers a summary of the tests to be performed for each bearing. There are two bearing designs and two specimens per design. The same tests were performed with and without bolt preloading. Three tests were planned for each bearing and preload case, where the bearing is reassembled on the test rig for each test. Each test consists on four loading processes. In summary, each bearing is assembled 6 times and subjected to 24 loading processes, making a total of 24 assemblies and 92 loading processes.

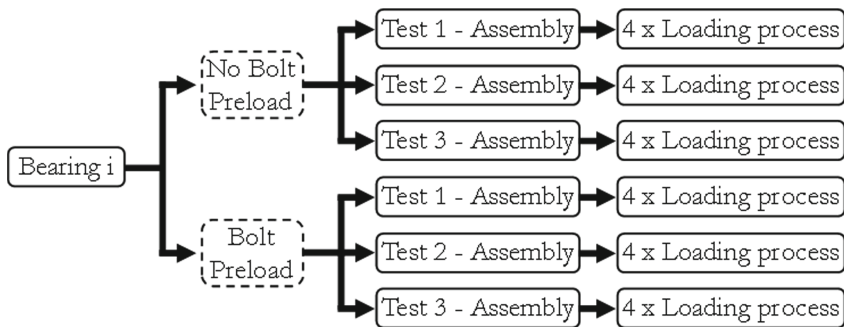


Fig. 5. Experimental test campaign schema for one bearing specimen

3 Results

Figures 6 and 7 show the stiffness comparison between the experimental tests and the FE results without and with bolt preload. The area that represents experimental test measurements considered all the obtained results with the exception of the first loading process of each test, because of the axial spherical bearing settlement. It was verified that, after this first loading process, the other three retrieved the same results for every assembly. Regarding the effect of the assembly process, it was observed that, after disassembling each specimen and putting it back in the test bench, the measured results offered some dispersion. Nonetheless, the main cause of the dispersion observed in the experimental results is the different stiffness behaviour showed by the bearings corresponding to the same design (i.e. differences between specimens 1 and 2, and between 3 and 4). A deep research on the cause of this difference was not carried out. However, there are some aspects that can promote this behaviour. For instance, a deficient lubrication would impede the correct wire-twisting, generating a rise in the stiffness and a higher contact pressure on one wire-race edge. Of course, manufacturing errors can also influence the structural behaviour, since more conformal wire-ring contact would retrieve a stiffer behaviour.

The tests proved that an adequate bolt preload has a relevant effect on the stiffness behaviour in comparison with low and non-controlled bolt preload. Thus, the lack of bolt

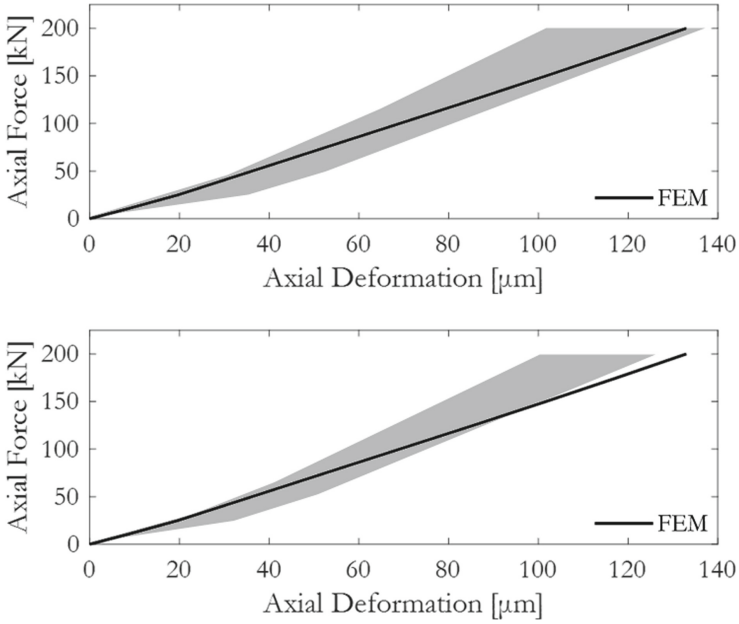


Fig. 6. Validation for bearings 1-2: with no preload (upper) and with preload (lower).

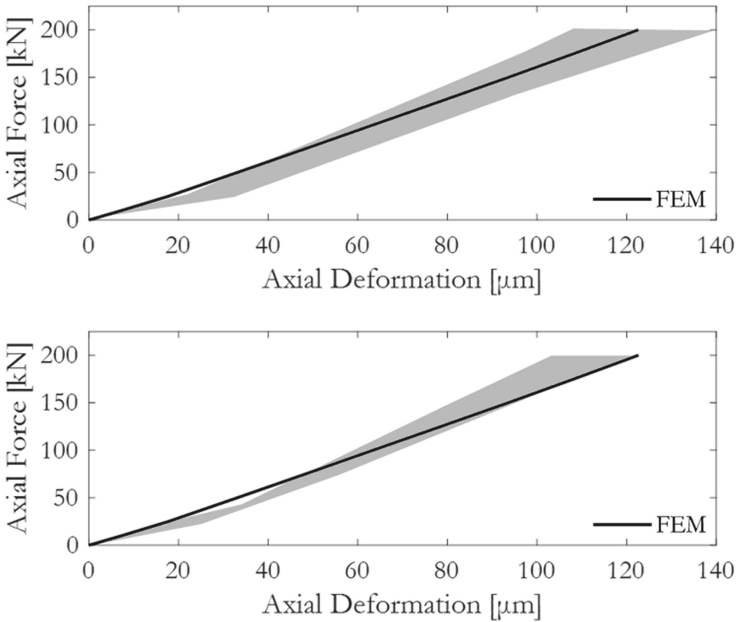


Fig. 7. Validation for bearings 3-4: with no preload (upper) and with preload (lower).

preload not only leads to a more scattered results, but also to a more flexible structural behaviour. The non-linear behaviour that show the experimental measurements was a non-expected effect, since the FE results always show a linear behaviour all along this load range. This effect could be also related with the manufacturing errors, promoting some rollers to be more loaded than others at the beginning of the loading process. It is not until a higher load level when all the rollers make contact and the linear behaviour starts.

Regarding the FE-experimental correlation, FE results are inside the area created by all the experimental results. According to this, it can be stated that the FE models used previously by the authors are experimentally validated.

4 Conclusions

The main conclusion that arises from the current work is that the FE models previously developed by the authors for the simulation of the structural behaviour of roller wire-race bearings are experimentally validated. Since the accuracy of the simplified approaches proposed in previous works was checked using these FE models as reference, the justification of their capabilities and limitations is now properly founded.

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