Dynamics of Tapered Roller Bearings: Modelling and Analysis

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Summary: The understanding of the dynamic performance of roller bearings, used in railway vehicles for instance, is fundamental to support the evaluation of the bearings performance via monitoring systems. The vibration output of the axleboxes is the measurable outcome of the bearing dynamic response, under operating conditions, that is characterized in this work. The main goal of this work is to develop a dynamic analysis tool, referred to as BearDyn, in MATLAB®, able to handle models representative of actual railway axle bearings, by using a multibody formulation to describe the mechanical elements of the bearing and their interactions, using realistic bearing geometric data obtained by precise measurements of tapered bearings. Online contact detection is studied in steady state general geometries. A dynamic analysis program of two colliding bodies is developed to validate the contact detection methods. In BearDyn, continuous contact force models based on the Hertz elastic contact theory and modified according to experimental evidence describe the interactions between the elements. Tribological lubrication models are applied to describe the tangential forces in the presence of lubricant. Finally, the BearDyn is demonstrated in the framework of realistic train operations that include the bearing loading due to the wheel-rail contact and the supporting mechanisms. The bearing dynamic response is obtained in terms of forces, kinematic quantities and different interaction measures, in the time and frequency domain. Two visualization tools are developed using MATLAB® and SAGA, to visually identify the geometries used and contact points locations.

1 INTRODUCTION

A rolling element has a finite life. The bearing will fail due to fatigue, caused by the high cyclic stresses between the rolling elements and raceways, even if operated under ideal design conditions. As the failure of components in the axle bearing is one of the most significant factors in the safety incidents, it’s important to have monitoring and diagnosing systems to prevent accidents and reduce maintenance costs. A number of tools and techniques of condition monitoring can be used to check a series of properties, such as Vibration analysis, Acoustic monitoring, Thermography and Lubrication sampling [1], [2]. The practical application of railway axleboxes bearings condition monitoring is done by using vehicle-based sensors. The noise and vibration output of the axleboxes of the rolling stock is in fact a measurable outcome of the bearing dynamic response, under the operating conditions, towards which the methodology developed here is aimed. Developing a computational tool that simulates the roller bearing dynamics can lead to an innovative way to predict flaws, reduce maintenance costs and better understand the behavior of the roller bearings under working conditions.

The current status in Roller Bearing Modeling is described by Gupta in [3]. In recent years, Schaeffler Technologies also developed an industrial software named CABA3D [4] for the simulation of dynamic processes in rolling bearings and Kiekbusch [5] developed dynamic simulation models using commercial multibody software ADAMS and SIMPACK with advanced lubrication models, cage modeling and elastic structures.

The method presented here is the development of a dynamic analysis tool, referred to as BearDyn which serves as the acronym for Bearing Dynamic Analysis Program, able to handle bearing models representative of the actual tapered roller bearings used in railway operations. The structure of the models and theoretical knowledge, in which the body of the work is done, is available in the work by Ambrósio [6] delivered to project MAXBE. Sequentially, Lima [7] applied the methodologies and the first version of BearDyn was developed. Lima [7] explained the formulation needed for BearDyn but did not implement all the necessary interactions for one full bearing. For this reason, the results of the
dynamic analysis were not trustworthy and since very few verification methods were used during the implementation. A study of better fitter numerical procedures was found necessary for BearDyn in order to better trust the models and successfully implement them in the Dynamic Analysis Program. For this reason, the work here developed focus on contact detection for all contacts in a Tapered Roller Bearing, and force calculation, including normal force models and more realistic lubrication models [8,9].

2 DYNAMIC ANALYSIS

Cartesian coordinates and Newton-Euler’s method are used to formulate the Equations of motion of the spatial multibody systems [10]. The kinematics of a single rigid body, is described by a set of coordinates $\mathbf{q}_i = [\mathbf{r}_i^T \mathbf{p}_i^T]^T$, in which the position of the body with respect to global coordinate system XYZ is defined by the coordinate vector $\mathbf{r}_i = [x \ y \ z]^T$ that represents the location of the local reference frame $(\xi \eta \zeta)_i$, and the orientation of the body is described by the rotational coordinates vector $\mathbf{p}_i = [e_1 \ e_2 \ e_3 \ e_4]^T$, which is made with the Euler parameters for the rigid body [10]. The complete multibody system, made of $nb$ bodies, is described by a set of coordinates $\mathbf{q} = [\mathbf{q}_1^T \mathbf{q}_2^T \ldots \mathbf{q}_{nb}^T]^T$. The multibody system generally includes a set of kinematic constraints denoted as $\Phi(\mathbf{q}, t) = 0$, which represent the kinematic joints or any type of relations between the coordinates. In the computational tool BearDyn developed for this project, kinematic constraints were not introduced as its range of application includes isolated roller bearings only. Consequently, the dynamic response of the system depends only on the forces applied to the bodies. The system is defined as a system of unconstrained bodies, where the equations of motion are used in a simplified form, for one body, described as

$$
\mathbf{M}_i \ddot{\mathbf{q}}_i = \mathbf{g}_i
$$

in which is $\mathbf{M}_i$ is the diagonal matrix with the mass and inertia of the body and $\mathbf{g}_i$, the vector with the sum of all forces and moments acting on the body. This system of equations is solved for $\ddot{\mathbf{q}}_i$. Then, in each integration time step, the accelerations vector, $\ddot{\mathbf{q}}_i$, together with velocities vector, $\dot{\mathbf{q}}_i$, are integrated in order to obtain the system velocities and positions for the next time step [11]. The vector $\mathbf{g}$ includes all the external and internal applied forces in the system, namely, the gravitational forces, the normal, hydrodynamic, elastohydrodynamic and friction forces between bearing rolling elements and the gyroscopic forces of the rigid bodies. The contact forces, presented and discussed in this work, need to be applied in the contact points identified during the contact detection process. In what follows, the vector of forces, $\mathbf{g}$, includes the forces applied in all the bodies of the system as $\mathbf{g} = [\mathbf{g}_1^T \mathbf{g}_2^T \ldots \mathbf{g}_{nb}^T]^T$, being each of the individual body force vector written as

$$
\mathbf{g}_i = \left\{ \begin{array}{c} \mathbf{f}_i \\
\mathbf{n}_{app,i} + \mathbf{A}_i^T (\mathbf{s}_i^p \mathbf{f}_i) - \mathbf{\omega}_i^T \mathbf{J}_i^T \mathbf{\omega}_i \end{array} \right\}
$$

where it is supposed that force $\mathbf{f}_i$ is applied on the contact point of body $i$, $\mathbf{n}_{app,i}$ is a vector with the moments directly applied, $\mathbf{\omega}_i$ is the body angular velocity, expressed in body coordinates and $\mathbf{J}_i$ is the inertia tensor, also expressed in body fixed frame. Therefore, when contact is detected, the coordinates of the contact points, in each surface, $\mathbf{s}_i^p$ need to be identified during the contact detection process. Afterwards, by using appropriate normal contact force models and tangential force models the vector of applied forces $\mathbf{f}_i$ is calculated. The dynamic analysis is performed by solving the system composed with the equations of motion from all bodies presented in Eq.(1) and integrating the defined vector $\ddot{\mathbf{y}}$, to get the velocities and positions of the bodies. Further detail on the general algorithm is explained in [10].
3 CONTACT DETECTION IN DYNAMIC ANALYSIS

For two moving bodies, there is always a pair of points, one in each surface, where the contact is more likely to occur. Depending on the forces acting in the system, the bodies have a translational and rotational movement, which leads to a position change of the contact point in each body, meaning that this pair of points must be updated in each time step. Over time, the program will find the two points that are in contact, or closer to each other, and identify the pseudo-interference between the surfaces. In realistic mechanical systems, it is likely that a rigid body contacts with other bodies in more than one pair of points, at the same time. For this reason, in a dynamic analysis simulation all the possible pair of contact points must be considered. It is necessary to study every pair of contact points individually, since the value of force developed in each point can be different depending on the geometry and kinematics of the body.

When applying the procedure to the case of a cylindrical body, such as a roller, approaching a generic surface. The situation is numerically perceived as contact, as illustrated in Figure 1, being the shaded volume a representation of the penetration of the roller in the surface. The contact patch between the two bodies is described as a line contact along the longitudinal direction, distributed over a small area. This means that this contact area cannot be simplified into a single point, being a line, eventually with varying interference depth. Consider now that the roller is divided in a user defined $N_{sl}$ number of strips, i.e., cylindrical segments. Now the contact problem of the complete roller can be described as $N_{sl}$ independent contact problems of thin cylinders, in which the contacting penetration depth is constant throughout the slice, or strip, described as a circular line [5]. Some surfaces of contact in a roller bearing are also described as lines, such as the cage side, cage top that contact the roller and roller small top, because of their small height to length ratio. This approach is realistic and allows for a simpler contact detection with less equations to solve.

![Figure 1: a) Roller Approaching a surface, (b) Roller penetrating a surface](image)

3.1 System of equations to solve

Surfaces in a dynamic analysis can be described as generic three-dimensional geometries or two-dimensional lines, which means that three different contact situations can develop, as represented in Figure 2. The contact can be between two generic surfaces, as seen in Figure 2 (a), a generic surface and a line, as seen in Figure 2 (b) and between two lines, represented in Figure 2 (c). Points $P$ and $Q$ represent the closer proximity points, each belonging to a generic surface on body $i$ and $j$, respectively. Furthermore, the geometries are defined in the referential $(\xi\eta\zeta)_i$ and $(\xi\eta\zeta)_j$ fixed to the centers of mass of bodies $i$ and $j$, respectively. Vector $d$ represents the distance between them, given by $d = r_P - r_Q$.

![Figure 2: a) Generic Surface, (b) Line Contact](image)

Still with reference to Figure 2, on point $P$ in body $i$ the vector normal to the surface is $n_i$ while $t_i$ and $b_i$ are the tangent and binormal vectors to the surface, respectively, forming an orthogonal basis. The same theory applies to $n_j$, $t_j$ and $b_j$ which are the vectors on point $Q$. No all these vectors are necessarily defined, depending on the contact type., i.e., in a contact with a line contact not always are vectors $n_i$ and $b_i$ formulated. Figure 2 shows the necessary vectors for each specific geometry. The
relation between the components of the different vectors defined in each body and the parameters used to define the surface depends on the specific geometry of the contacting surface. The conditions for minimal distance between points \( P \) and \( Q \) depend on the contact type, resulting in four different possible systems to solve, since for the line to line contact, two different systems can be used depending on the specific geometry formulation.

![Figure 2: Candidates to contact points between: (a) parametric surfaces; (b) a surface and a line; (c) two lines](image)

The conditions for minimal distance are,

**Surface to surface**

\[
\begin{align*}
\mathbf{n}_Q^\text{T} \mathbf{t}_P &= 0 ; \quad \mathbf{d}^\text{T} \mathbf{b}_P = 0 \\
\mathbf{n}_Q^\text{T} \mathbf{t}_Q &= 0 ; \quad \mathbf{d}^\text{T} \mathbf{b}_Q = 0
\end{align*}
\]  

(3)

**Surface to line**

\[
\begin{align*}
\mathbf{n}_Q^\text{T} \mathbf{t}_P &= 0 \\
\mathbf{d}^\text{T} \mathbf{t}_Q &= 0 \\
\mathbf{d}^\text{T} \mathbf{b}_Q &= 0
\end{align*}
\]  

(4)

**Line to line**

\[
\begin{align*}
\mathbf{d}^\text{T} \mathbf{b}_Q &= 0 \quad \text{or} \quad \mathbf{d}^\text{T} \mathbf{b}_Q &= 0 \\
\mathbf{d}^\text{T} \mathbf{b}_P &= 0 \quad \mathbf{d}^\text{T} \mathbf{t}_Q &= 0
\end{align*}
\]  

(5)

Effective contact occurs if, besides the fulfilment of the system above, penetration also exists, which is expressed by \( \delta = \mathbf{d}^\text{T} \mathbf{n}_Q \leq 0 \), otherwise the points are in close proximity, but not in contact.

The system of nonlinear equations to solve depends on the case in study, but the methods used are general, meaning that can be applied to all of them. Since the contact points are evaluated online, the method should also be fast. Solving the system of equations means finding a set of design parameters, \( \mathbf{x} = \{x_1, x_2, \ldots, x_{np}\} \), that can in some way be defined as optimal, using optimization techniques or other standard methods. The more significant methods implemented were \texttt{fsolve}, from the optimization toolbox in MATLAB\textsuperscript{®}, and Newton Raphson Method, used for solving a system of equations in an iterative process with the Jacobian matrix, the matrix of all first-order partial derivatives that can be evaluated from the analytical first order derivative or obtained from the function \( \mathbf{F}(\mathbf{x}) \) in a discrete interval \( \Delta \mathbf{x} \).

### 3.2 Demonstration cases with contact detection

Testing the alternative methods individually allows to better understand the numerical issues, allowing an easier way to interpret results, identifying problems and to favouring decisions about corrections and modifications. For this purpose, the contact detection is first tested in simple steady state geometries. Circles, spheres and cylinders are formulated and generated in random positions and contact detection is implemented to find the points of more proximity. This allows to detect numerical and formulation.
problems that are corrected before the methods are used with more complex bearing geometries.

Since the contact detection is evaluated every timestep, it is also important to test the robustness of the methods applied in a Dynamic Analysis Program. The contact between circle and cylinder is tested. First a single circle is the rolling body, simulating a coin and afterwards a cylindrical roller defined with multiple circles rolling along a cylindrical floor surface is simulated. For the normal force calculation, the Kelvin-Voigt Contact Model is used and for the Tangential Forces, the Threall Friction Model. This program allows the study of contact detection and force calculations without having to deal with the complexity of a full bearing. In the beginning of the simulation, the roller is placed close to the cylindrical surface, in a higher position so that the body starts rolling downhill due to gravity. The kinematic results are visualized in MATLAB® using two generic MATLAB® functions getframe and VideoWriter. As expected, the moving body starts rolling down and, after reaching the bottom, starts losing velocity, stopping at the same height as it started.

4 FORMULATION FOR THE BEARING CONTACT DETECTION

The most important roller contacts, in any type of roller bearing, take place between the roller side and the raceways or flange surfaces and between rollers and cage pockets. In this work, only the most common and important roller contacts are considered, reason why contact is studied only between the rollers and each of the bodies directly surrounding them and not between rollers or between cage and raceways. The contacts considered in this work, illustrated as in Figure 3,

- Contact between roller and raceways;
  - Contact between roller and inner raceway (Inner);
  - Contact between roller and outer raceway (Outer);
- Contact between roller and flanges;
  - Contact between roller top and right flange (FR);
  - Contact between roller top and left flange (FL);
- Contact between roller and cage;
  - Contact between roller side and cage pocket tops (C2.1, C4.1, C4.2);
  - Contact between roller side and cage pocket sides (C3.1, C3.2).

All bearing surfaces are characterized by geometries of revolution, i.e., they are obtained by sweeping a plane line about an axis of revolution. Therefore, the coordinates of any point in the surface can be expressed in terms of the parameters that define the planar line and the sweep angle. Contact between the raceway and the rollers is formulated using the slice method. The roller is reduced to six circles with different radius and longitudinal positions, all defined in the initialization and contact points are obtained by solving the system with three nonlinear equations. Contact between tapered roller small end and flange is described as the circular landmark that limits the roller, obtained with the same equations from the roller slice, contacting the conical surface of the left flange. Due to its spherical
shape, the roller large end contact with the right flange is between a spherical cap surface and a conical flange, making this contact different from the other flange. Previous work [7] showed this contact to be prone to convergence errors due to the large value of the curvature radius of the roller large top. This problem is due to the radius of the spherical surface being too large. The geometries don’t have limits to the domain, which make it possible for the contact point to be detected in a location where there is no realistic meaning for the contact to exist. If this situation occurs, the bodies will continue to approach and the contact forces will not be applied. In Figure 4 (a) the roller bearing with the original radius of the spherical cap is represented. To solve this problem, the radius of curvature $R_{cr}$ was projected in order to guarantee that the contact point is initially in the correct position between flange and sphere, as showed in Figure 4 (b).

![Figure 4: Roller side view with different spherical cap radius: (a) Previous value implemented; (b) new radius applied to BearDyn](image)

The roller to cage contacts are represented in Figure 5. The large top of the roller is a spherical surface, reason why its contact with the cage top can be detected as between a surface and a line. The small top of the roller is assumed as flat and the contact can be detected as two points, between a circumference located at each end of the roller and a line for the cage top. When checking for the collisions between the roller and the side of the cage pocket, the contact is line to line, since the roller side is modelled with circular slices.

![Figure 5: Contact of circle with line, as in the contact between the roller and the side of the pocket](image)

### 5 CONTACT FORCES

Once contact is detected, the forces are applied in each pair of contacting points $P$ and $Q$. Some kinematic variables are required by the models of the normal contact and friction forces. It is necessary to calculate the relative velocity between the bodies, namely its projection on the surface tangent to the contacting bodies on the points of contact and its projection on the normal vector to the surface. The velocities of the contact points $P$ and $Q$ are obtained by taking the time derivative of the position vectors, $\mathbf{r}_P$ and $\mathbf{r}_Q$, written as,

\[
\dot{\mathbf{r}}_P = \dot{\mathbf{r}}_i + \mathbf{A}_P \omega \mathbf{s}_P \\
\dot{\mathbf{r}}_Q = \dot{\mathbf{r}}_j + \mathbf{A}_Q \omega \mathbf{s}_Q
\]  

(6)
where \( \mathbf{r}_i \) and \( \mathbf{r}_j \) are the velocities of the origin of the body fixed referential and \( \mathbf{\Omega}'_i \) and \( \mathbf{\Omega}'_j \) the skew-symmetric matrices associated to the angular velocities of body \( i \) and \( j \), respectively, expressed in the body fixed coordinate systems. The relative velocity between bodies \( i \) and \( j \) in the point of contact is obtained as, \( \mathbf{r}_{pq} = \mathbf{r}_p - \mathbf{r}_q \), and the sliding velocity in the contact point is just the projection of the same vector in the tangent plane to the contacting surfaces, as, \( \mathbf{u} = \mathbf{r}_{pq} - (\mathbf{r}_{pq} \cdot \mathbf{n}_Q) \mathbf{n}_Q \).

5.1 Normal Contact Forces

In the roller bearing contact problem, the dimension of the contact area is small when compared with the typical dimensions of the contacting bodies. Hence, the normal loads that develop in the contact patch can be replaced by normal forces and the contact is said to be non-conforming [37]. The contact patches can be divided in two categories, point or line, depending on the surface geometry. This results in two different equations used to calculate the normal force. The normal contact force, in the case of point contact is \( f_n = K_{pt} b_{pt}^{3/2} \). According to the Hertz theory [12], proposed here to study the contact problem, the stiffness \( K_{pt} \) depends on the dimensions of the contact area, the material properties and the surfaces curvature of the contacting bodies. For the line contact is given by Palmgren’s simplified equation, depending on the line length, \( L_{ef} \) and number of slices, \( N_{sl} \), and equivalent modulus \( E' \) as, \( f_{ns} = 0.356E'N_{sl}^{1/3}L_{of}^{8/9} \delta^{10/9} \).

5.2 Tangential Contact Forces

Besides the normal forces that develop during contact, also tangential forces due to friction or due to the lubrication fluid develop between the contacting bodies. The tangential forces are also applied in each pair of contacting points \( P \) and \( Q \) and their value is proportional to the normal force developed during contact. Regardless of the type of contact, the relation between the tangential forces and the normal contact forces is given by,

\[
f_t = \mu f_n \tag{7}
\]

where \( \mu \) is the equivalent friction coefficient. Note that the tangential force is applied in the opposite direction of the relative velocity between the contacting surfaces, \( \mathbf{u} \). The simple form of Eq.(7) hides the complexity of the calculation of \( \mu \) for many important tangential forces, as in the case of lubricated contact. In the case of lubrication between the contacting surfaces, the friction coefficient depends on the lubricant film thickness and on the roughness of the contacting surfaces, described as the lubricant film parameter, \( \Lambda \). The type of contact is different and the equivalent friction coefficient has to be evaluated differently. Assuming for each lubrication mode a different equivalent friction coefficient, the equivalent friction coefficient is written as [8,13]

\[
\mu = \begin{cases} 
\mu_{bd} & \Lambda < \Lambda_{bd} \\
\mu_{bd} - \mu_{fm} \left( \Lambda_{bd} - \Lambda_{fm} \right)^{6/5} + \mu_{fm} & \Lambda_{bd} \leq \Lambda \leq \Lambda_{fm} \\
\mu_{fm} & \Lambda > \Lambda_{fm}
\end{cases}
\tag{8}
\]

where \( \Lambda \) is the lubricant film parameter. The typical values for the lubricant film parameter used to define the transitions between the different lubrication modes are \( \Lambda_{bd} = 0.01 \) and \( \Lambda_{fm} = 1.5 \) [13]. The evaluation of the equivalent friction coefficient requires the prior evaluation of the lubricant film parameter, which in turn imply the calculation of the lubricant film thickness, and the equivalent friction coefficients for each mode of lubrication, \( \mu_{bd} \) and \( \mu_{fm} \), calculated theoretically. All this values are calculated using oil Total Carter EP220 properties.
6 COMPUTATIONAL IMPLEMENTATION

The function F>Contact is where all the contact detection and contact forces application occurs. The program, depending on the variable method_flag selected by the user, has three different methods to solve the contact detection: fsolve, Numerical Newton Raphson and Analytical Newton Raphson. Each timestep, F>Contact starts by solving each contact detection individually and if the contact exists (\( \delta < 0 \)), proceeds to calculate the normal and tangential contact forces, based on the indentation and lubrication conditions, and applying the forces and resultant torque to the force vector. The function then proceeds to the next contact points until all of them are evaluated for that timestep. For contact detection, the function CostFunction is where the system of equations is formulated. The function starts by calling Geometry two times, one for each geometry involved in the contact, where the vectors formulations for the different geometries are stored. With the obtained vectors, the systems of equations are formulated, according to the correspondent contact type (surface to surface, surface to line or line to line). Since there are no other available similar enough programs or studies which can be used to compare results and experimental procedures, the verification was made in two parts: first with MATLAB® a visualization tool represents the geometries in the roller bearing and contact points detected in the initial timestep, then with a dynamic visualization tool named SAGA the kinematic response is visualized. To report forces and contact points position happening during the simulation, it is important to identify the success timesteps from ODE45 and guarantee that the values obtain in the so called “prediction” steps are not considered.

6 RESULTS AND DISCUSSION

Different preliminary results and verification methods for BearDyn allow for a better understanding on how the implemented models work together and how to achieve the computational effectiveness of BearDyn on limited computational resources. Even though the program is not fully functional at this time, it is still possible to obtain results for different cases and with the results to illustrate problems which ultimately help understanding the difficulties. For this purpose, the contact detection is verified with the representation of the obtained contact points and geometries for the first timestep is represented in MATLAB®. Around 80% of the time spent in the simulation is evaluating the contact points. To compare the methods implemented for contact detection, BearDyn simulated, under the same conditions, the movement for t=0.03s using the different methods. The elapsed time is showed in Table 1.

<table>
<thead>
<tr>
<th>Method</th>
<th>Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>fsolve</td>
<td>24h</td>
</tr>
<tr>
<td>Computational Newton Raphson</td>
<td>8h</td>
</tr>
<tr>
<td>Analytical Newton Raphson</td>
<td>9h30</td>
</tr>
</tbody>
</table>

Table 1: Time elapsed for the same t=0.03s simulation using three different methods to solve contact

It is clear that both Newton Raphson methods are quicker than the optimization function. It is expected that the Newton Raphson with the analytical Jacobian matrix installed in the program, specific for each contact, is the fastest of the three methods, since the derivatives are more precise and lead to the solution in fewer steps. This is not verified and is explained by the extra computational effort involved in creating the analytical Jacobian.

7.1 Results of the dynamic analysis

As a first approach, a simulation of the tapered roller bearing with tangential forces but no load is addressed. This simulation allows to test the contact detection inside a dynamic analysis, as well as the models used for the calculation of the contact forces. The analysis is performed for a total integration time of 0.03 seconds, where the roller progressed about 1/4 of the complete pitch circle. Due to the initial velocity and gravity force, the rollers start to contact with the outer raceway and are pushed to one side, until the spherical surface from the large top of the roller start to contact the right flange. The
reported forces show that the tangential force is always 10 percent of the normal force, which means that the contact is dry and the lubricant is unable to decrease the friction force. For this magnitude of load applied a larger lubricant film thickness in the contact points is needed, meaning that the oil Total Carter EP220 is not suitable for high pressure roller bearings from axleboxes and instead, a lubricant grease must be selected. With SAGA is possible to visualize the points of more proximity detecting contact and the force being applied.

When introducing load to the inner raceway, it is clear that misalignment of the raceways occurs due to the applied force. This problem was corrected by introducing an axial force in the inner raceway to contradict this movement. To try to understand the behaviour of the roller bearing under load, the force magnitude was reduced to 5kN. This force allowed the simulation to run for \( t=0.2\) s. The performance time of the simulation is not ideal, since it is not enough for the rollers to complete one revolution around the bearing center. However, this simulation allows the broadest results possible to be obtained for the time. In Figure 6-(a) the forces in the inner raceway are represented. The system shows a transient behaviour in the beginning and converges after \( t=0.01\) s to the value 5kN, which is the value of the load force being applied.

![Figure 6](image)

Figure 6: (a) Forces in Inner Raceway; (b) Frequency response with FFT

To obtain the frequency response of the system, the total forces actuating on the inner raceway over time are converted into a function of amplitude and frequency, using a Fast-Fourier Transform algorithm, with the results relative to the first 0.01 seconds of the simulation ignored. The resulting graphic, presented in Figure 6, showed a frequency band standing out around 30 kHz. According to Li [14], a bearing with no defects can have a response where no frequency stands out. This does not correspond to the case presented, since it should be noted that only with the dynamic response obtained for several complete revolutions of the roller bearing are the FFT reliable for the frequency analysis.

8 CONCLUSIONS

This work focused on the remaining parts fundamental to the dynamic tool: first, being the contact detection, based on the kinematics and geometry of each body, while the second consists in the evaluation of the normal and tangential forces developed with lubrication models. The formulation to describe all the geometries needed in BearDyn is explained and validated with a developed MATLAB® visualization tool, with necessary modifications. Contact detection is implemented in BearDyn and the first timestep possible contact points are represented and validated. The system assembly for contact detection was implemented with a simple and organized method involving only two functions and one flag to create any system of an implemented contact, allowing for future contact detections in Spherical Roller Bearings to be introduced in the same functions. Three different methods to solve the systems were developed and installed, in order to find the fastest and most efficient,
reducing the simulation time to more than half. Normal and tangential forces, due to normal contact and to lubrication, are modelled. To allow more realistic simulation, the lubricant must be replaced. The correct grease lubricant must be selected from manufacturers and the lubricant properties changed.

When performing dynamic simulations of the models with BearDyn for a bigger timespan, it is identified the need to address the extremely high computational effort. This difficulty clearly identifies the need for the development of robust and efficient computational algorithms, eventually based on the use of parallel computation strategies. The code must be adapted to simulate a double row tapered bearing, since this is the roller bearing used in the axlebox and only with a double row it’s possible to maintain equilibrium. After working, the double row bearing must be compared with experimental results.

REFERENCES