Tribo-Dynamics of Bearings for Electric and Hybrid Electric Vehicles’ Powertrains

2nd Year Progress Report

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Abstract

Roller bearings are critical components in hybrid and electric vehicle powertrains. They are often performance limiting, and introduce NVH (Noise, Vibration and Harshness), tribological and wear challenges. The high-speed and varying load conditions of modern electric powertrains necessitates accurate modelling of the bearings to ensure satisfactory system performance and durability. Furthermore, with a push towards achieving zero-prototype development, the use of advanced simulation tools to accurately predict their behaviour at both component and system level is becoming more prevalent.

Roller bearings are critical components of these motors. Current dynamic analyses that assume dry contacts are not valid in high-speed applications due to the role of elastohydrodynamic (EHL) lubricant film. Under light load and high speed, the EHL contacts of these bearings change their tribological behaviour, affecting parameters such as thermal performance, wear, sub-surface stress and frictional power loss. Hence, a combination of dynamics and contact mechanics is required for accurate performance and durability modelling. Furthermore, dynamic analyses that assume rigid bearing races do not accurately capture contact conditions between the roller and race, limiting the accuracy of the models in predicting the above parameters. There is a paucity of experimental or numerical tribo-dynamic work for predicting high-speed bearing behaviour.

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**Nomenclature:**

|  |  |
| --- | --- |
|  | Acceleration (m/s2) |
|  | Half-width of the contact (mm) |
| *C* | Radial clearance (µm) |
|  | Diameter of roller (mm) |
|  | Pitch diameter (mm) |
|  | Equivalent (reduced) elastic modulus (GPa) |
|  | Radial load in x-direction (N) |
|  | Radial load in y-direction (N) |
|  | Ball Pass Frequency of Outer Race (Hz) |
|  | Ball Pass Frequency of Inner Race (Hz) |
|  | Shaft Rotational Frequency (Hz) |
|  | Dimensionless equivalent geometry (-) |
|  | Central film thickness (m) |
|  | Stiffness (N/m) |
|  | Roller length (mm) |
|  | Exponent of localized deflection (-) |
| *N* | Number of rolling elements (-) |
|  | Contact pressure (GPa) |
|  | Maximum Hertzian pressure (GPa) |
|  | Dimensionless contact pressure |
|  | Equivalent radius of contact (mm) |
|  | Speed of entraining motion (m/s) |
|  | Dimensionless speed parameter (-) |
|  | Velocity (m/s) |
|  | Contact load (N) |
|  | Dimensionless load parameter (-) |
|  | Conjunction x-coordinate (µm) |
|  | Dimensionless x-coordinate (-) |
|  | Conjunction y-coordinate (µm) |

**Greek Symbols:**

|  |  |
| --- | --- |
|  | Angular position (rad) |
|  | Pressure viscosity coefficient (m2N-1) |
|  | Contact Deflection (m) |
|  | Stribeck parameter (-) |
|  | Atmospheric lubricant dynamic viscosity (Pa.s) |
|  | Lubricant dynamic viscosity (Pa.s) |
|  | Dimensionless viscosity (-) |
|  | Lubricant density (kg/m3) |
|  | Atmospheric lubricant density (kg/m3) |
|  | Dimensionless density (-) |
|  | Composite surface roughness (m) |
|  | Boundary shear strength of asperities (Pa) |
|  | Angular velocity of cage (rad/s) |
|  | Angular velocity of inner race (rad/s) |
|  | Angular velocity of shaft (rad/s) |
|  | Relaxation factor (-) |

# Research Motivation

The automotive industry is currently transitioning into the next phase of powertrain technology. As automotive manufacturers are forced to meet tightening fleet-wide emissions regulations, the electrified vehicle market share will increase. Early adoption barriers will begin to reduce as non-ICE (internal combustion engine) vehicle sales are incentivised. Coupled with improvements in technology and a shift in customer attitudes, electric and hybrid electric vehicles are estimated to have a global market share of 48% by 2030 (Mosquet *et al.*, 2018).

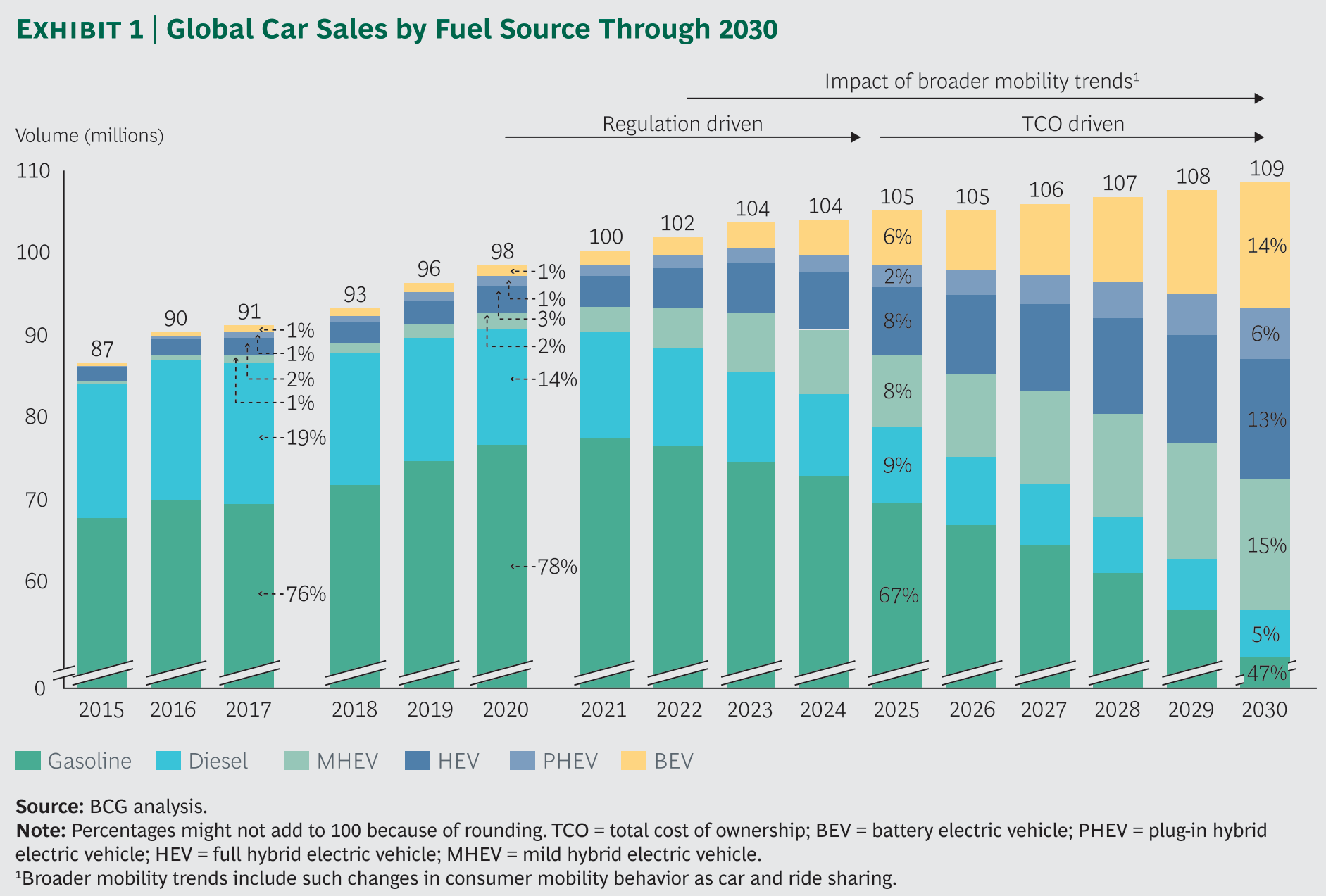


Figure 1 - Global Car Sales by Fuel Source Through 2030 (Mosquet et al., 2018)

The use of ultra-high speed and low load motors introduces new challenges regarding NVH (Noise, Vibration and Harshness) and the tribology of interacting conjunctions. The compact, lightweight and efficient motors operate under significantly different working conditions and are subject to different underlying physics; such as regime of lubrication, dynamic response and magneto-mechanical interactions. This style of powertrain architecture therefore involves high-speed bearing operation in both the motor and transmission.

With a trend towards cost saving zero-prototype development, the use of simulation tools in modern powertrain development is also growing. Significant cost reductions can be achieved using commercial flexible multi-body dynamic solvers to replicate system level operation of these vehicles. Multi-system vehicle powertrain concepts are pushing complexity of simulation models and this requires accurate and robust component level understanding. Associated performance characteristics of the bearings such as friction and wear, thermal stability and generated vibration and noise must be accurately modelled at the development stage to ensure full system success (Wensing, 1972).

# Aims and Objectives

The aim of this work is to create a dynamic bearing model that implicitly includes the effects of the elastohydrodynamic (EHL) lubricant film at the roller-race conjunction. The bearing races will be modelled as flexible bodies to represent the realistic compliance of modern transmission housings and account for this at conjunction level. The numerical models will be valid for high-speed operating conditions present in modern electric vehicle powertrains and able to replicate the transient operating nature of the bearings such as fluctuating input loads and speeds.

A mixture of numerical models and experimental validation of these models will be used to achieve the research aim.

The objectives are outlined below:

1. **1D EHL Model:**

* Develop a 1-dimensional EHL model that numerically solves the film thickness and pressure distribution at the roller-race contact, valid for high-speeds and loading conditions.
* Numerically validate this using case-studies in literature.
* Experimentally validate this using the mini-traction machine in the Tribology Lab.

1. **Dynamic model:**

* Develop a dynamic bearing model.
* Numerically validate the model using literature.
* Implement this model in AVL Excite (multi-body dynamic simulation software) for system level simulations.

1. **Flexible bearing race model:**

* Investigate the effect of modelling the bearing races as flexible rather than rigid bodies.
* Implement this work into the bearing dynamic model.
* Experimental validation using a high-speed bearing test rig. A soft bearing bore will be used to allow for raceway deformation and will be instrumented with accelerometers to measure acceleration and displacement in time and frequency domains at different locations.

1. **Full tribodynamic model:**

* Use the bearing dynamic model with flexible bearing races to find displacement of the bearing inner race and hence roller load based on lubricated contact mechanics. Use these as boundary conditions in the numerical EHL model to find film thickness and thus study the effects of inlet starvation, boundary and viscous friction, power loss and dynamic response for various loading conditions.

Experimental validation will be performed using a high-speed bearing test rig instrumented to measure dynamic response, as well as friction and power loss using a high-resolution torque transducer.

**5. Use Artificial Intelligence for enhancement of the EHL model**

* Perform a case study to understand how machine-learning can be used to speed up the computation process of the EHL solver using an already established AI platform at AVL.
* Include an investigation on how much training data is required for the desired accuracy and what the limitations with regard to loading conditions may be.

# Novelties

The following novelties have been identified in addressing the aims of this work. In subsequent sections, these will be discussed in more detail in the context of available literature.

**Full Tribodynamic Model:**

Previous high degree-of-freedom dynamic models have assumed a dry contact between the roller and race. To fully investigate NVH and durability of roller bearings in this work, the effect of the lubricant film will be coupled with a bearing dynamic model for a full solution. This study will also include much higher relative surface velocities between roller and race than in previous literature to replicate conditions in modern EVs.

**Flexible Bearing Race:**

A common assumption in dynamic bearing models is that the rollers and races can be modelled as rigid bodies due to the high stiffness of bearing mounts. Modern, high-performance transmission housings are becoming increasingly more compliant, thus rendering this assumption invalid. The deformation of the bearing race will not only affect the contact conditions between the roller and race, but also the dynamic stiffness and behaviour of the bearing. In these studies, the rollers and race will be modelled as flexible bodies.

**Artificial Intelligence:**

A full tribodynamic model is highly coupled and thus computationally intensive to solve. The use of machine learning to reduce computation time will be investigated in this work. The use of machine learning has not, to the authors knowledge, been reported in the field of tribology.

**Experimental Dynamics and Numerical Tribological Modelling:**

An experimental test rig was instrumented to obtain boundary conditions for use in tribological models. This captures the dynamic displacement of the bearing centre and circumvents the need for a dynamic model. To the authors knowledge, this methodology and the high-rotational speeds have not been employed in open literature.

# Numerical EHL Model and Solution

## Reynold’s Equation

Reynold’s equation (Reynolds, 1886) is the governing equation of fluid film lubrication theory. For Newtonian fluids it can be derived from the full Navier-Stokes equations making the following assumptions, primarily the neglection of inertial forces and only retaining viscous forces on the lubricant (Gohar, 1988):

1. Body forces are negligible (mass of film is negligible)
2. Pressure is constant through the lubricant film (z-direction) due to thin film (dimensions of the region of pressure are typically 100 times the central film thickness).
3. No slip at boundaries
4. Lubricant flow is laminar (low Reynolds number)
5. Inertia and surface tension forces are negligible compared with viscous forces (working fluid has low mass and low acceleration)
6. Shear stress and velocity gradients are only significant across the lubricant film (z-direction)
7. The lubricant behaves as a Newtonian fluid
8. Lubricant viscosity is constant across the film (z-direction)
9. The lubricant boundary surfaces are parallel or at a small angle with respect to each other

Reynolds equation is a second order, non-linear partial differential equation. It is made up of the pressure induced terms (Poiseuille flow) and the boundary velocity-induced term (Couette flow).

For the line contact problem, such as that at the conjunction between a cylindrical roller and race, dimensions in the side-leakage direction, , are much bigger than the direction of entraining motion, . Pressure in direction is assumed constant due to the negligible gradient, and the contact can be analysed in 1-dimension. The assumption is valid in the contact apart from small regions near the edge where the roller profile changes. A simplified 1-dimensional version of Reynolds equation can therefore be used:

|  |  |
| --- | --- |
|  | [6] |

To solve Reynolds equation numerically, it must first be discretized and then solved using the finite-difference method. The following procedure explains this discretization

Removing the transient squeeze term and replacing terms for simplification:

|  |  |
| --- | --- |
|  | [7] |

Due to the many orders of magnitude differences between lubricant film thickness (µm) and pressures (GPa), the numerical solution often becomes unstable. Dimensionless parameters are therefore defined to remove this instability. These are as follows:

|  |  |
| --- | --- |
|  |  |
|  |  |
|  |  |
|  |  |
|  |  |
|  |  |
|  |  |

Terms in the simplified Reynolds equation are replaced with dimensionless parameters. Similar terms are then grouped and rearranged to give the final form:

|  |  |
| --- | --- |
|  | [8] |

where

|  |  |
| --- | --- |
|  | [9] |

Grouping terms for simplicity:

|  |  |
| --- | --- |
|  | [10] |

|  |  |
| --- | --- |
|  | [11] |

Making substitutions

|  |  |
| --- | --- |
|  | [12] |

|  |  |
| --- | --- |
|  | [13] |

The final term is removed as velocity, , is independent of when no stretching of the surfaces occurs. This is then differentiated to give:

|  |  |
| --- | --- |
|  | [14] |

and

|  |  |
| --- | --- |
|  | [15] |

Substituting into equation 13:

|  |  |
| --- | --- |
|  | [16] |

|  |  |
| --- | --- |
|  | [17] |

The final form is therefore:

|  |  |
| --- | --- |
|  | [18] |

## Finite Difference Formulation

For finite difference formulation, the central difference formula based on Taylor series expansion (Hoffmann and Chiang, 2000) is used. The second derivative of pressure using second order central discretization for the spatial domain is therefore:

|  |  |
| --- | --- |
|  | [19] |

and the first derivative is given by:

|  |  |
| --- | --- |
|  | [20] |

Replacing terms in the final form of discretized Reynold equation:

|  |  |
| --- | --- |
|  | [21] |

|  |  |
| --- | --- |
|  | [22] |

Pressure at each node point can then be represented by:

|  |  |
| --- | --- |
|  | [23] |

Simplified to

|  |  |
| --- | --- |
|  | [24] |

where,

|  |  |
| --- | --- |
|  | [26] |

|  |  |
| --- | --- |
|  | [27] |

|  |  |
| --- | --- |
|  | [28] |

## Effect of pressure on lubricant viscosity

EHL temperatures are typically in the region of 0.5-4 GPa. The resultant behaviour of the viscosity at these pressures is instrumental in forming the EHL film and must be accounted for. The Barus law (Barus, 1893) determines viscosity increase with pressure assuming constant ambient temperature:

|  |  |
| --- | --- |
|  | [29] |

where is the lubricant viscosity at gauge pressure, **,**  is the viscosity at = 0, and is the pressure-viscosity coefficient (m2/N) and is specific to the lubricant. This relationship does not account for the change in with temperature and pressure (Gohar and Rahnejat, 2019), becoming inaccurate above 0.5 GPa.

A more comprehensive relationship which simultaneously includes the effects of temperature and pressure was one proposed by Roelands (Roelands, 1966) and developed by Houpert (Houpert, 1984). Roeland’s law is therefore accurate at higher contact pressures:

|  |  |
| --- | --- |
|  | [30] |

The Roelands pressure-viscosity coefficient, , is a function of both and , with being the reference or ambient temperature, for example at the inlet:

|  |  |
| --- | --- |
|  | [31] |

where

|  |  |
| --- | --- |
|  | [32] |

and

|  |  |
| --- | --- |
|  | [33] |

The oil constants and are independent of both pressure and temperature, and can be typically taken as 0.68 for computational purposes.

## Effect of pressure on lubricant density

For accurate film EHL film shape calculations, the effect that pressure has on the lubricant density must be considered. The most common equation for this is the widely used Dowson and Higginson model (Dowson and Higginson, 1977):

|  |  |
| --- | --- |
|  | [34] |

where is the lubricant atmospheric pressure. This has also been modified to account for temperature effects:

|  |  |
| --- | --- |
|  | [35] |

## Effect of temperature on viscosity

Most EHL work assumes constant temperature of the contact and that viscosity and density are dependent on pressure only. Standard experiments have been performed to assess effect of temperature on viscosity. Results have previously been curve fit by Crouch and Cameron (Crouch and Cameron, 1961), with the most simple fit due to Reynolds:

|  |  |
| --- | --- |
|  | [36] |

where is the viscosity of the lubricant at temperature , is the viscosity at representative temperature , and represents the temperature difference between the two. is the thermoviscous constant and is lubricant specific. This relationship is only valid for small temperature rises of the lubricant. A more accurate and widely used equation is the expression from Vogel:

|  |  |
| --- | --- |
|  | [37] |

with the three constants dependant on the lubricant, obtained from knowing three pairs of values for and .

## 1D EHL Solution Methodology

Reynolds equation is used to calculate contact pressures. Assuming a thin film of Newtonian lubricant in a line contact, following form is used and discretized in the manner shown previously:

|  |  |
| --- | --- |
|  | [38] |

where is the direction of entraining motion into the contact. Squeeze film motion is neglected for this analysis. For the pressure-density relationship in the compressible model, Dowson-Higginson model (Dowson and Higginson, 1977) is used

|  |  |
| --- | --- |
|  | [39] |

The increase in lubricant viscosity with pressure is modelled using Roeland’s law (Roelands, 1966) due to its accuracy at higher contact EHL pressures

|  |  |
| --- | --- |
|  | [40] |

where

|  |  |
| --- | --- |
|  | [41] |

Pressure distribution is obtained from the variations in film thickness at the contact, which is defined as below:

|  |  |
| --- | --- |
|  | [42] |

where is the central film thickness, the second term represents an idealised film thickness parabola, with the ultimate term representing the localised contact deflection. Central film thickness is first estimated using:

|  |  |
| --- | --- |
|  | [43] |

The dimensionless materials, speed and load parameters used are:

|  |  |
| --- | --- |
|  | [44] |

Below, the method of solution for the numerical EHL model is provided:

1. The load value at the roller-race contact is input, this is often obtained from a dynamic model.
2. An initial estimation of lubricant film thickness, is made.
3. Inlet and outlet distances are set to -4.5 to 1.5 based on the contact half width calculation. This sets up the computational domain.
4. Pressure distribution and film thickness are obtained through simultaneous solutions of equations 38-42. Newton-Raphson iterative scheme is used for speed and robustness of convergence (Okamura, 1993). Pressure convergence criterion is required for the iterative solution:

|  |  |
| --- | --- |
|  | [45] |

where

Under-relaxation is applied between successive iterations where the criterion is not met

|  |  |
| --- | --- |
|  | [46] |

where the under-relaxation factor is typically .

1. Hydrodynamic reaction load is calculated using the integration of pressure over the computational domain

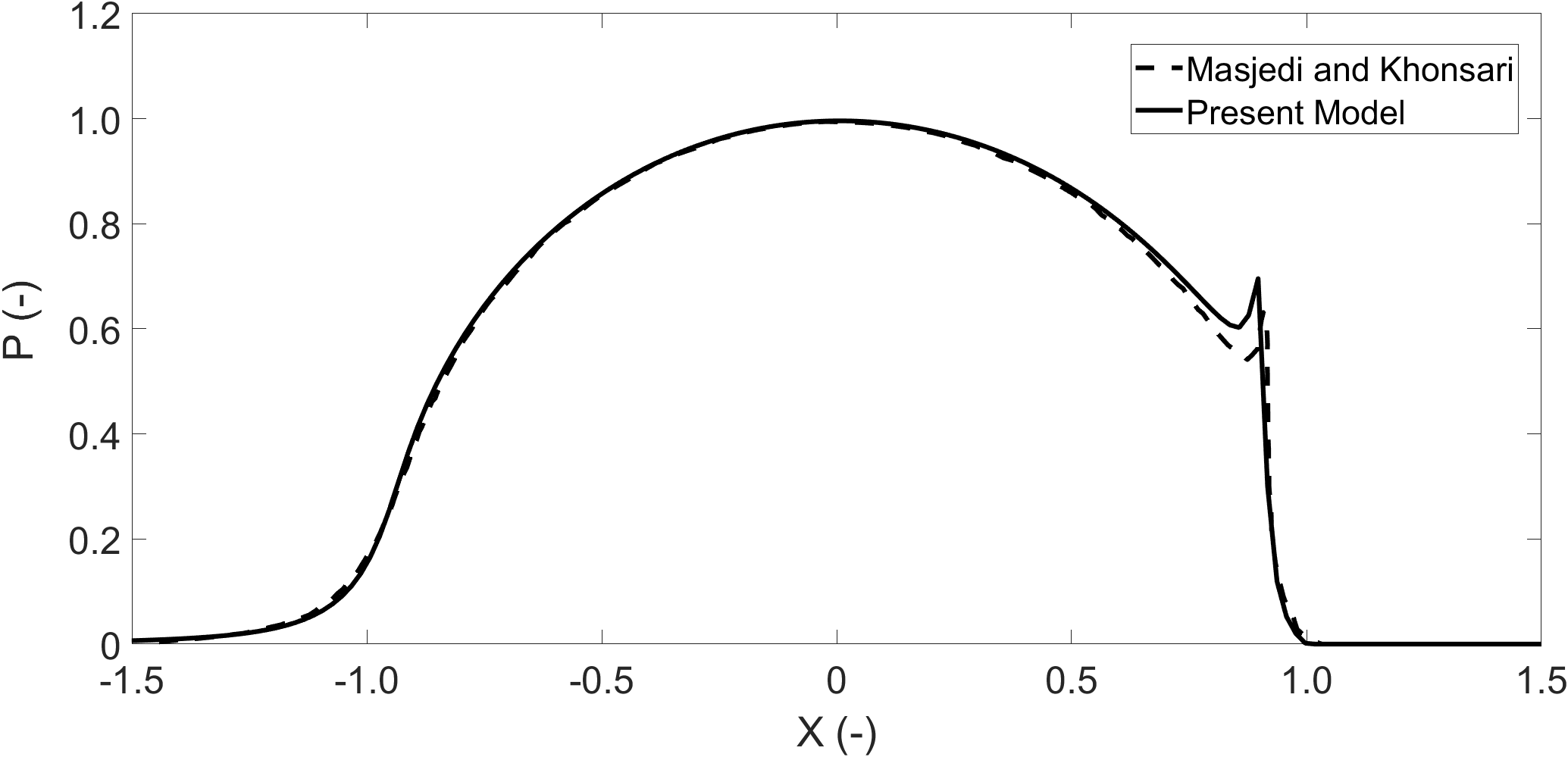
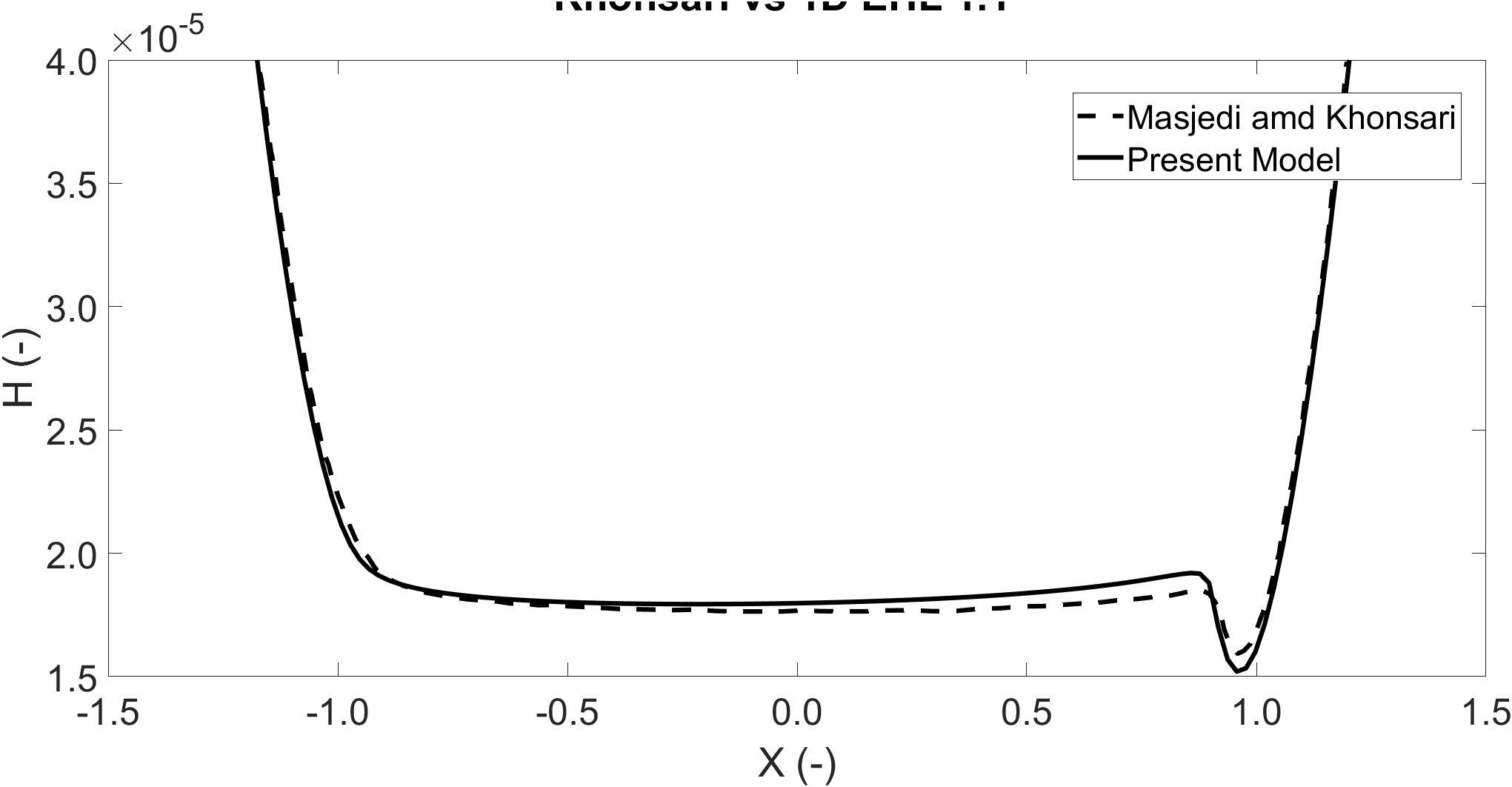
|  |  |
| --- | --- |
|  | [47] |

The total reaction from the hydrodynamic load should equal the total load share on the roller, , obtained from the explicit tribological model. Convergence criterion is applied

|  |  |
| --- | --- |
|  | [48] |

where

## Numerical EHL Validation

1. Conjunction level validation of the numerical method for solving the EHL film thickness and pressure distributions was performed using the work of Masjedi and Khonsari (Masjedi and Khonsari, 2012). These were validated against their smooth surface plots with no asperity pressure contribution. The dimensionless input parameters were = 1 x 10-4, = 1 x 10-11 and = 4500. The results shown in Figure 9 and Figure 10 show a good agreement between the model used in the study and the work of Masjedi and Khonsari for the dimensionless pressure, , and dimensionless film thickness, .
2. 
3. Figure 9 - Validation of dimensionless pressure distribution, present model (solid), Masjedi and Khonsari (dashed)
4. 
5. Figure 10 - Validation of dimensionless film thickness distribution, present model (solid), Masjedi and Khonsari (dashed)

# Initial Tribo-Dynamic Investigations Using Experimental Dynamics Measurement and Numerical Tribological Model

Prior to the development of the 6 DOF dynamic model, a high-speed experimental test rig was instrumented to obtain boundary conditions for subsequent tribological models. This work was performed to ascertain what type of models need to be developed in the tribology domain without the need for a dynamic model. The requirement for lubricated rather than dry contact models, regimes of lubrication and workflows such as implicit or explicit modelling have been investigated.

## Methodology

To obtain the film thickness at the roller-race contact, the load on each roller must be calculated implicitly based on the contact deflection. This is obtained conventionally from a dynamic model by solving the equations of motion, but these are currently lacking the required in-depth physics such as system flexibility, flexibility of races and thermal effects (Meyer, Ahlgren and Weichbrodt, 1980) (Matsubara, Rahnejat and Gohar, 1988) (Wang *et al.*, 2015) (Liu, Shi and Shao, 2017). To circumvent the need for a complex flexible dynamic model and to capture real behaviour of a bearing under test, displacement of the bearing centre and hence contact deflection is captured from experimental test results.

An experimental test rig is used to obtain the relative displacement between the inner and outer bearing races. Then, using the Hertzian load-deflection relationship and accounting for the lubricant film thickness, load on an individual roller is found at each instantaneous position around the bearing centre through a speed sweep of 0 – 15 000 rpm. An implicit analytical tribological model is used which calculates the film thickness at the contact. This loop is iterated to account for the tribo-dynamic coupling. After the experimentally informed implicit tribo-dynamic model is solved, load and speed values at specific rotational velocities are used within an explicit numerical EHL model to calculate film thickness and pressure distribution across the contact. The flow diagram in Figure 1 illustrates the methodology used. Interactions between each stage will be explained in subsequent sections.



Figure 1 - Methodology overview

The rollers within a bearing carry an instantaneous share of the overall load applied to the bearing (Guo and Keller, 2020). Deviation of the supported shaft centre from its nominal geometric centre results in a loaded region of the bearing. In the conjunction between the bearing roller and race, the non-conformal nature of contact generates very high pressures when under load. This causes local surface deformation and an increase in lubricant viscosity, resulting in EHL film formation (Gohar, 1988) (Grubin, 1949). Emerging clearances in unloaded regions of the bearing results, which can cause the roller-to-race contact to deviate from the EHL regime towards the hydrodynamic regime, resulting in sliding and roller-cage collisions (Mohammadpour, Johns-Rahnejat and Rahnejat, 2015). Hence, the contact may go through different regimes of lubrication throughout its rotation (Denni *et al.*, 2019).

Figure 2 shows the cylindrical roller bearing (CRB) in equilibrium position with zero preload or design clearance. Under zero applied radial load, , the initial deformation, , and radial clearance, , between rollers and races are both zero. Due to external force application and system dynamics, an instantaneous radial load, , will displace the inner bearing race from its equilibrium state ( and ). By analysing an individual roller at its instantaneous angular position, , the resultant displacement of the bearing centre can be used to find deflection at the roller-race contact, . These contact deformations will result in contact forces, , which act to keep the rollers and races in dynamic equilibrium. The above interpretation is valid considering rigid inner and outer races.

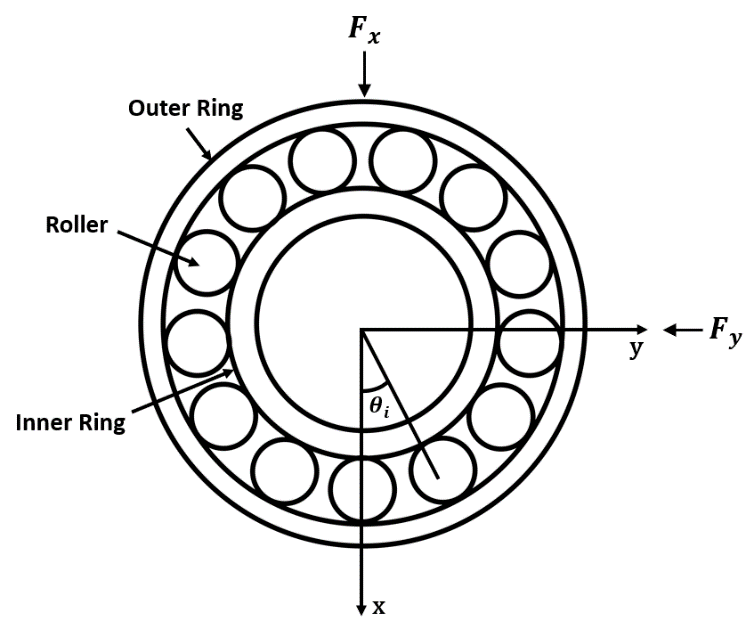


Figure 2 - CRB in equilibrium position

Figure 3 presents the case whereby an instantaneous force is applied to the inner race. This causes deflection of the rollers in the loaded region and clearance or less deflection around the rollers in the unloaded region. The value of the total deflection, at the inner and outer race corresponds to the component of displacement of the bearing centre at the instantaneous position of that particular roller. In the current study, this value was experimentally measured from the test rig and is used quasi-statically as the boundary condition for the tribological model. It should be noted that in-plane two degrees of freedom motion are analysed in this study which is a valid assumption under dominant radial loading with secondary horizontal motion from the full system dynamics. Misalignment along the length of the rollers is not considered due to the high stiffness of the shaft and bracket, hence a 1-dimensional analysis for EHL is sufficient (Gupta, 1979).

In the loaded region, the Hertzian load-deflection relationship is used to obtain the resulting instantaneous load on the roller. This value is then used implicitly within an analytical tribological model to calculate contact film thickness for an individual roller as it passes through different angular positions during a speed sweep. An extrapolated film thickness formula is used for this purpose. The calculated film thickness imposes additional deformation at the contact points which itself changes the calculated load (as shown in equation 5). Hence, an iterative approach is required between the force calculation and implicit tribological model. In this part of the workflow, the stiffness and damping of the EHL film is neglected due to its rigid-like stiffness, which is several orders of magnitude higher than the Hertzian contact (Dareing and Johnson, 1975) (Mehdigoli, Rahnejat and Gohar, 1990).

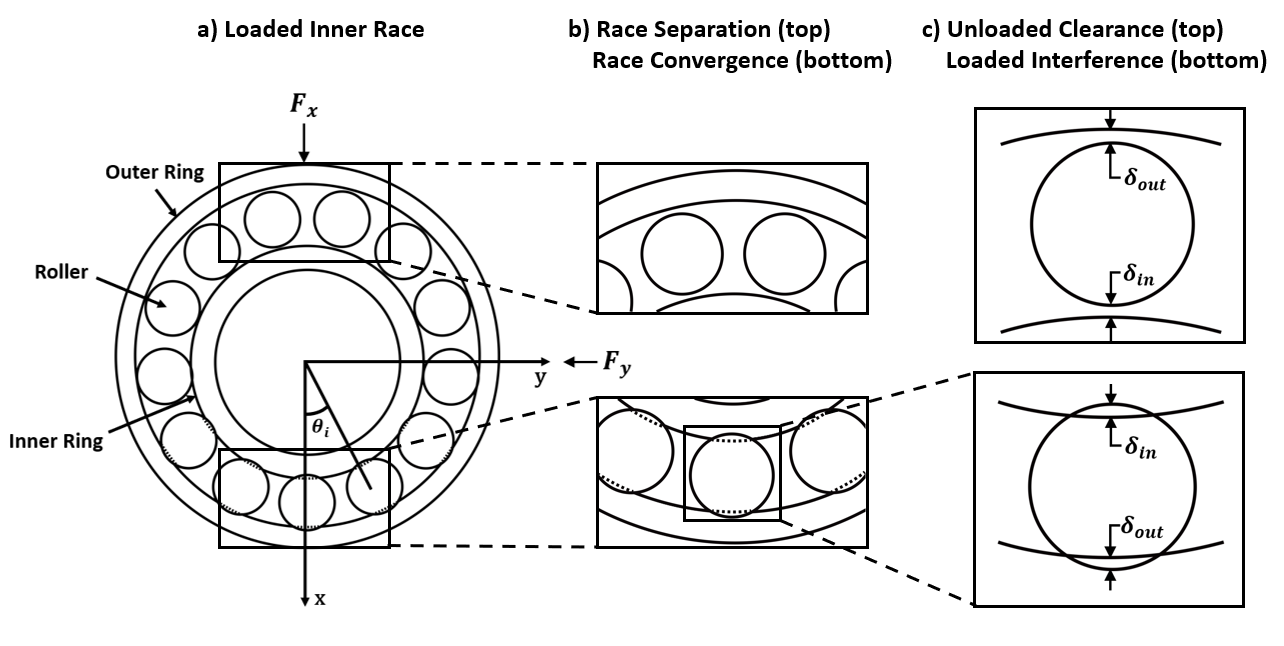


Figure 3 – Mutual separation and convergence of inner and outer races: a) Loaded inner race, b) Race separation and convergence, c) Resultant interference and clearance

For in-depth tribological investigations, the numerical solution of the fluid film is essential. This provides the pressure, film thickness and shear distributions in the contact to study the durability and frictional efficiency of the system. In the current study, the load on the roller and contact kinematics obtained from the implicit bearing model is used explicitly in a 1-dimensional elastohydrodynamic model to obtain film thickness and pressure distribution at the contact for specific loading periods through the speed sweep, as is shown in Figure 4. This explicit approach significantly improves the computational efficiency of the model and in principle, maintains the accuracy since only the central value of the film is required in the load calculation.

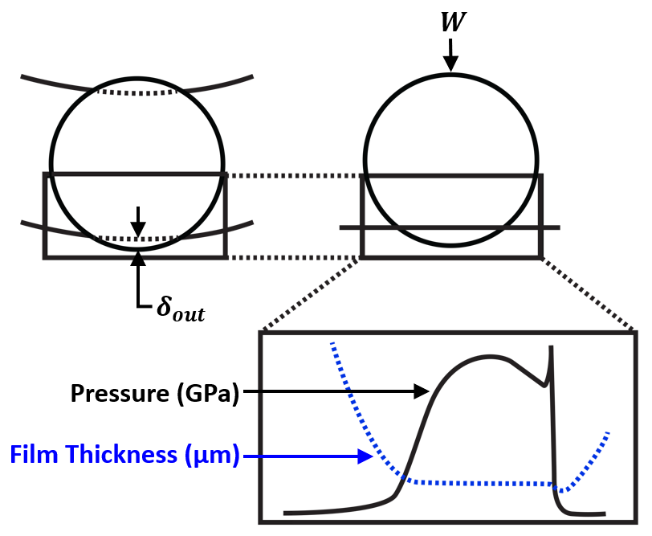


Figure 4 - EHL film thickness and pressure distribution at contact

## Experimental Test Rig

## The displacement of the bearing centre governs the conditions at the contact and was found experimentally using a high-speed bearing test rig, originally reported by Walker *et al.* (Walker *et al.*, 2018). A 5 kW AC synchronous motor, capable of reaching up to 32 000 rpm was coupled to a steel shaft that is supported by two bearing brackets. Radial force was transferred to inner bearing race via the shaft using a hinge/arm mechanism and load application device on the shaft. Displacement data were obtained from an instrumented bearing bracket. Figure 5 shows a schematic of the rig.

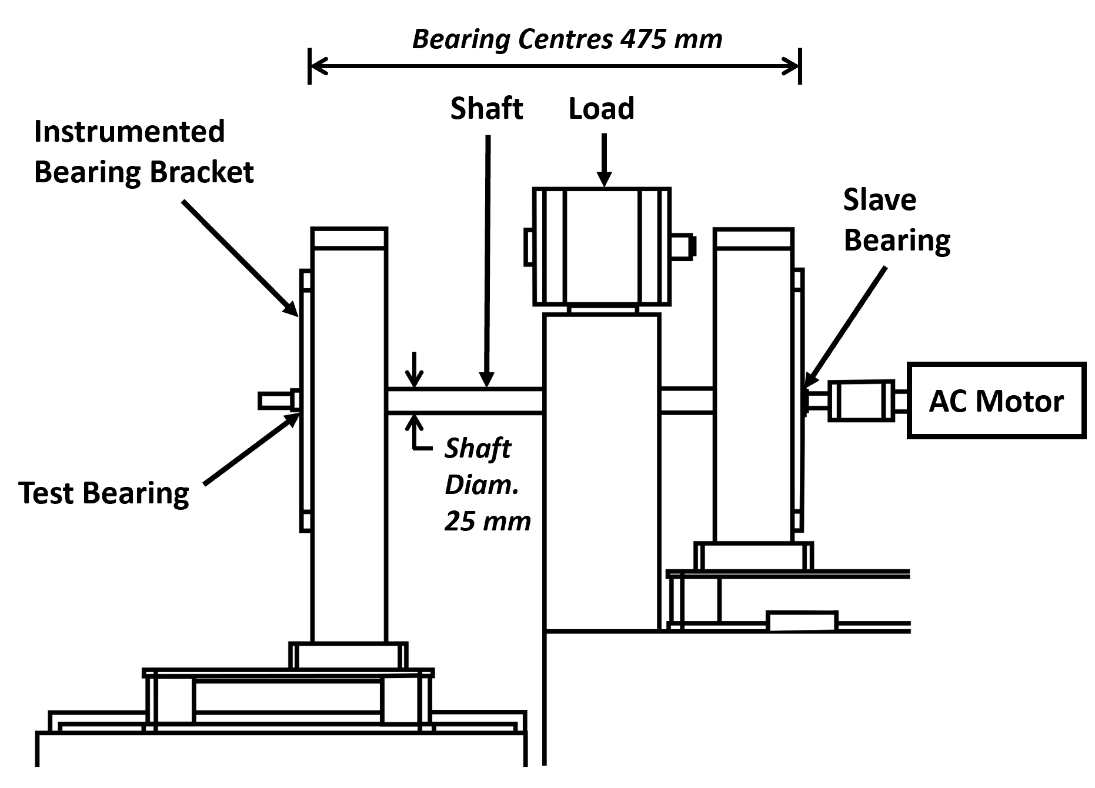


Figure 5 – Experimental rig schematic

The motor control unit was connected to a voltage input, transmitted via an NI cDAQ-9178 USB chassis for optimum resolution of the input voltage. A MATLAB script controlled the voltage ramp over a specified time-period and thus the spindle acceleration. In this study, a transient speed sweep was performed from 0 – 15 000 rpm in a 4 s period, with 750 N of static radial load applied to the shaft. The bearing under test was a single row cylindrical roller bearing, NU 205 ECP, located in an aluminium test bracket that had an extruded bore for instrumentation.

* + 1. Instrumentation

The outer surface of the bearing bore was instrumented with two Type 4383 single axis piezo-electric charge accelerometers, with a frequency range of 0.1 – 8.5 kHz and sensitivity of 3.16 pC/ms2. These measured acceleration of the bracket’s bore, corresponding to the outer race of the bearing (Figure 6(a)). Two single beam laser vibrometers measured the displacement of the shaft at the edge of the bearing which corresponds to the displacement of the inner race of the bearing (Figure 6(b)). A dual-beam vibrometer was used to measure the rotational speed of the shaft. All laser vibrometers had a frequency range of 0 – 10 kHz and maximum speed of 20 000 rpm.

A program in MATLAB controlled the speed of the shaft through a ramped voltage input. Data was simultaneously acquired from the accelerometers and laser vibrometers through synchronised input channels at a sampling rate of 100kHz. Simultaneous control and data acquisition ensure accurate results between different instrumentation locations.

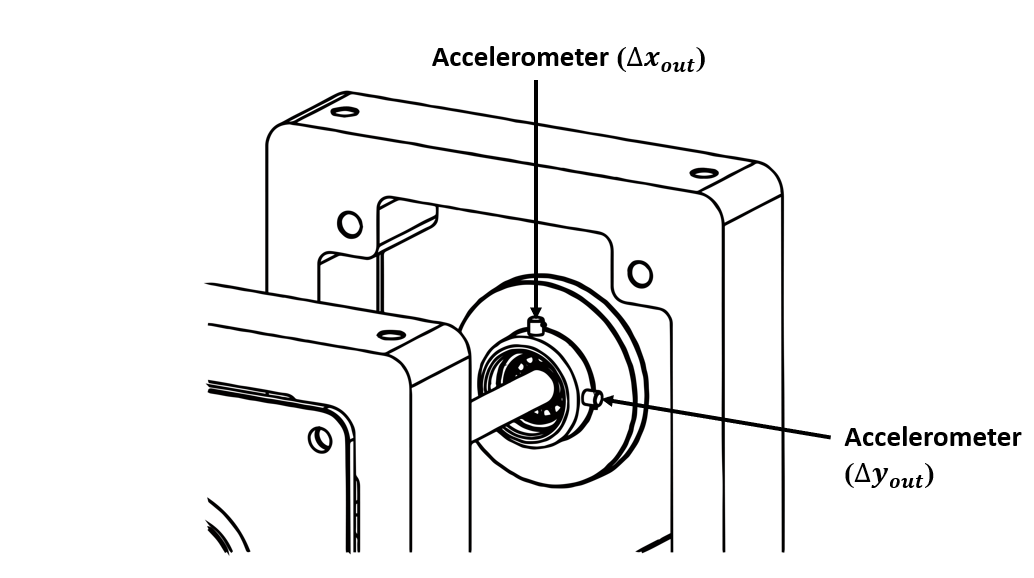


Figure 6(a) - Accelerometer Locations on Test Bracket

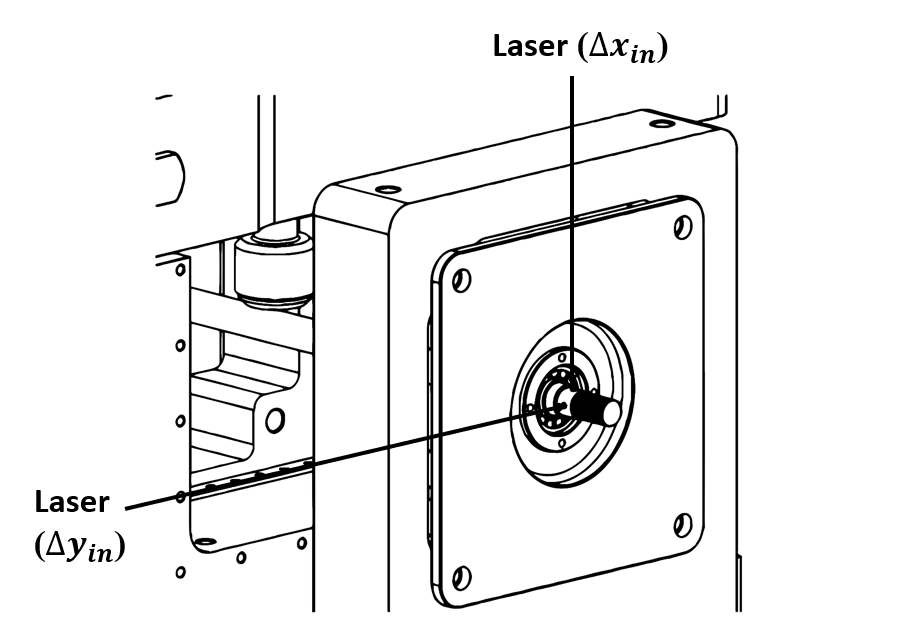


Figure 6(b) - Laser Vibrometer Locations on Shaft

* + 1. Signal Processing

1. The bearing race displacements were obtained from the instrumented bearing using time-domain data from the accelerometers and laser vibrometers. The laser vibrometer directly provided displacement data, whereas the accelerometer data required post-processing of acceleration data. To ascertain the displacement of the bearing bore and thus outer bearing race, the accelerometer results were integrated twice:

|  |  |
| --- | --- |
|  | [1] |
|  | [2] |

1. As this double integration amplifies non-linearities in the original signal, it requires pre-processing (Takeda *et al.*, 2014). Low frequency drift is removed from the unfiltered time-domain data by applying a linear fit to the data and then removing the trend from it.
2. To extract the frequency content of the speed ramp, waterfall plots were generated using the accelerometer and laser vibrometer data. Short-time fast-Fourier transform (STFFT) was used, with data windows of six shaft rotations at each 100 rpm increment. The resulting signal was then filtered using a Butterworth bandpass filter for a frequency range of 70 - 10 000 Hz to remove the high amplitude low frequency noise that was inherent to the equipment setup. The filter used 3 dB of passband ripple and 7 dB attenuation in the stopbands, which were set at 35 - 12 500 Hz.
3. The relative displacement of the bearing centre could then be obtained from the relative displacement between the shaft (measured using the single beam laser vibrometer) and bearing bore:

|  |  |
| --- | --- |
|  |  |
|  | [3] |

1. where and correspond to the displacements of the shaft and bearing bore respectively.

## Contact Mechanics of the Roller-Race Conjunction

The contact between roller and race is modelled as an elastic deformation between an equivalent finite length elastic cylinder and rigid plate. This assumption is realistic under the conditions of an EHL contact. The contact force on an individual roller at each instantaneous position, , is obtained from the Hertzian load-deflection relationship:

|  |  |
| --- | --- |
|  | [4] |

where is the Hertzian contact stiffness non-linearity between a rolling element and the inner or outer raceway groove. For the case of rolling element bearings, the exponent of localised deflection, , is equal to 10/9 (Harris, 1984). The contact deflection of a roller relative to the race, , is due to the normal dynamic motion (i.e. the local mutual convergence) of the inner and outer bearing races, contribution of lubricant film thickness and any additional clearance or interference fit (Mohammadpour, Johns-Rahnejat and Rahnejat, 2015). This is expressed as:

|  |  |
| --- | --- |
|  | [5] |

where is the local radial clearance, and and are the displacement components of the inner bearing race from its geometric centre. This normal approach between both races is the sum of the total deformation of the rollers and both races (Hamrock and Dowson, 1981), hence:

|  |  |
| --- | --- |
|  | [6] |

The equilibrium of forces in a system of stiffnesses in series is therefore:

|  |  |
| --- | --- |
|  | [7] |

To find the individual deflection of each contact due to differences in contact stiffness, the following relationship is used:

|  |  |
| --- | --- |
|  | [8] |

The deflection value at each contact based on the total deflection and relationship between contact stiffness is then found from Equation (6) and (8) to give:

|  |  |
| --- | --- |
|  | [9] |

The normal stiffness of the inner and outer races differs due to their geometry. To calculate the stiffness at each contact, the following equation is used:

|  |  |
| --- | --- |
|  | [10] |

The deflection for a range of loads is calculated based on the geometry and material properties at the inner and outer race contacts. This non-linear relationship is numerically obtained and then curve fitted and represented by a power function in the form , with and representing the contact stiffness. Individual contact stiffness and deflection at the inner and outer race contacts is then found. The overall contact stiffness, , is given by:

|  |  |
| --- | --- |
|  | [11] |

where and are the stiffnesses of the inner and outer race contacts respectively.

## Implicit Tribological Model

A stepwise solution was performed on an individual roller as it passes through each angular position. The roller bearing and bracket tolerances are such that the internal clearance is 0 µm between roller and race. This means that in an unloaded state, there is no deflection of elements or raceway. It also means that displacement in positive x-y corresponds to the same total magnitude of deflection of roller and race contact. For each time step, the bearing is first assumed to be in equilibrium position, and film thickness is assumed to be 0 µm. The deflection of the bearing is calculated under these conditions and is therefore a function of the relative displacement between inner and outer bearing races. With deflection at the time step calculated, the resultant lubricant regime and subsequent analytical solutions can be performed based on the following three conditions:

indicates a film of 0 µm and no load.

indicates complete separation of the roller and race. In this instance, the lubricant is assumed to fill the separation gap, with the film thickness value equalling the magnitude of the separation:

|  |  |
| --- | --- |
|  | [12] |

Under this condition, the lubrication is in the hydrodynamic regime. The hydrodynamic lubricant reaction load was derived by Rahnejat (Rahnejat, 1984), and is given by:

|  |  |
| --- | --- |
|  | [13] |

where is the half length of the contact, is the speed of lubricant entrainment into the contact, is the lubricant viscosity, is the reduced radius of the roller and race and is lubricant film thickness.

indicates deflection at the roller-race contact. This means that contact pressure is sufficiently high for the lubrication regime to be elastohydrodynamic. For the elastohydrodynamic regime, an iterative process is performed to solve film thickness. This is due to the contribution of EHL film towards deformation and consequently the load in the contact.

The cylindrical roller and race contacts are modelled by an equivalent rigid roller against a semi-infinite elastic half space of equivalent elastic modulus, . The extrapolated central film thickness for a line contact is therefore obtained (Dowson and Toyoda, 1979) from:

|  |  |
| --- | --- |
|  | [14] |

where the following dimensionless parameters are used:

|  |  |
| --- | --- |
|  | [15] |

where is the reduced radius of the contact, is the length of the roller and is the speed of entraining motion into the contact and is the contact load. Assuming pure rolling, the speed of entraining motion is given by:

|  |  |
| --- | --- |
|  | [16] |

An iterative process is used to calculate load on the roller based on total deflection including lubricant film (back to Eq.5). At each time step where an EHL film is present, the following convergence criteria must be met before the next time step is calculated:

|  |  |
| --- | --- |
|  | [17] |

## Friction Calculations

Frictional power is a crucial factor for characterising bearing performance. Under modern high-speed conditions, it is important to also understand the frictional behaviour of the system. Roller bearings often operate within the mixed elastohydrodynamic regime of lubrication. Friction is generated by a combination of viscous shear of the lubricant and asperity interactions. Total friction at the mixed-EHL contact is a combination of the boundary friction, , and hydrodynamic or viscous friction, :

|  |  |
| --- | --- |
|  | [18] |

Boundary friction is a function of asperity contact pressure, calculated using the Greenwood and Tripp model in this paper (Greenwood and Tripp, 1970):

|  |  |
| --- | --- |
|  | [19] |

where is the Eyring shear stress of lubricant and is the pressure coefficient for shear strength of asperities, obtained from asperity level friction measurement. Asperity load, , and area occupied by asperities within the apparent contact are obtained as below assuming a Guassian distribution of asperity peak counts:

|  |  |
| --- | --- |
|  | [20] |

and

|  |  |
| --- | --- |
|  | [21] |

where is the Stribeck parameter and and are statistical functions obtained from numerical integration of the Guassian distribution of asperities.

Viscous Friction due to shearing of the lubricant film in the EHL contact is found using the below experimentally validated formulae (Evans and Johnson, 1986):

|  |  |
| --- | --- |
|  | [22] |

where

|  |  |
| --- | --- |
|  | [23] |

where is the average pressure at the apparent contact, is the lubricant thermal conductivity and is given by:

|  |  |
| --- | --- |
|  | [24] |

where 𝐾′, 𝜌′, and 𝑐′ are respectively the thermal conductivity, density, and specific heat capacity of the contacting solid.

To inform the boundary friction model, surface topography data are required. An Alicona InfiniteFocus Variation Microscope with a x10 objective was used for topography measurements to calculate the roughness parameter, . This had a vertical resolution of 30 nm and sampling point separation of 176.9 nm in the *y-z* plane of the roller and 1 nm in *x*. An area of 530 by 588 µm was captured. Data was processed using Vision65 Map Premium, where the profile of the radius of the roller was removed. The measured parameters are presented in Table 1.

**Table 1** - Surface Topography Data

|  |  |
| --- | --- |
| Root-mean-square height | 0.197 µm |
| Density of peaks | 0.00116 1/(µm)2 |
| Arithmetic mean peak curvature | 0.180 1/µm |

## **Results**

* + 1. Experimental Result Validation

The experimental results obtained from the presented rig were verified against analytically calculated frequency contents. This was to ensure the correct post processing of data for the input to numerical models. The bearing and lubricant data properties are given in Tables 2 and 3 respectively which were also used in simulations.

Table 2 - Bearing Specification

|  |  |
| --- | --- |
| Inner Race Bore | 25 mm |
| Inner Race Diameter | 31.5 mm |
| Outer Race Diameter | 46.5 mm |
| Roller Diameter | 7.5 mm |
| Roller Length | 9 mm |
| Number of Rollers | 12 |
| External Load | 740 N |
| Radial Interference | 0 µm |

Table 3 – Lubricant and Material Properties

|  |  |
| --- | --- |
| Pressure Viscosity Coefficient () | 2.1 10-8 Pa-1 |
| Atmospheric lubricant dynamic viscosity () | 0.08 Pa.s |
| Lubricant inlet density () | 833.8 kg.m-3 |
| Eyring stress ( | 2 MPa |
| Shear strength of asperities ( | 0.3 |
| Thermal conductivity of fluid | 1600 W.m-1.K-1 |
| Modulus of elasticity of contacting solids | 210 GPa |
| Poisson’s ratio of contacting solids | 0.3 |
| Density of contacting solids | 7850 kg.m-3 |
| Thermal conductivity of contacting solids | 46 W.m-1.K-1 |
| Specific heat capacity of contacting solids | 470 J.kg-1.K-1 |

Primary frequencies in the system due to the interaction of the rolling elements, races and the shaft could then be verified. These frequencies were calculated analytically, with and representing the ball pass frequencies of the inner and outer race respectively and being the rotational frequency of the shaft:

|  |  |
| --- | --- |
|  | [35] |
|  | [34] |
|  | [36] |

At 14 000 rpm, the theoretical inner and outer race frequencies are calculated to be 1669 Hz and 1131 Hz respectively, with the experimental results being 1611 Hz and 1131 Hz. The first order shaft rotational frequency from the experiment was 232 Hz compared to theoretical calculation of 233 Hz. The above frequencies can be seen clearly in Figure 7, which shows the bearing bore displacement spectra, and Figure 8 which represents the shaft displacement spectra. The verification frequencies are identical in both, confirming that the bearing motion is accurately measured by the experimental methodology. It is observed that at certain speeds, ball pass frequency of outer race has larger contribution than the inner race. This is particularly highlighted at 12000-14000 rpm. These regional effects are contributed by modal behaviour of the bracket and the bed. The critical speed of the unloaded shaft, where lateral bending frequency of the shaft is equal to the rotation frequency (Budynas and Nisbett, 2011), occurs at 12 660 rpm and 211 Hz.

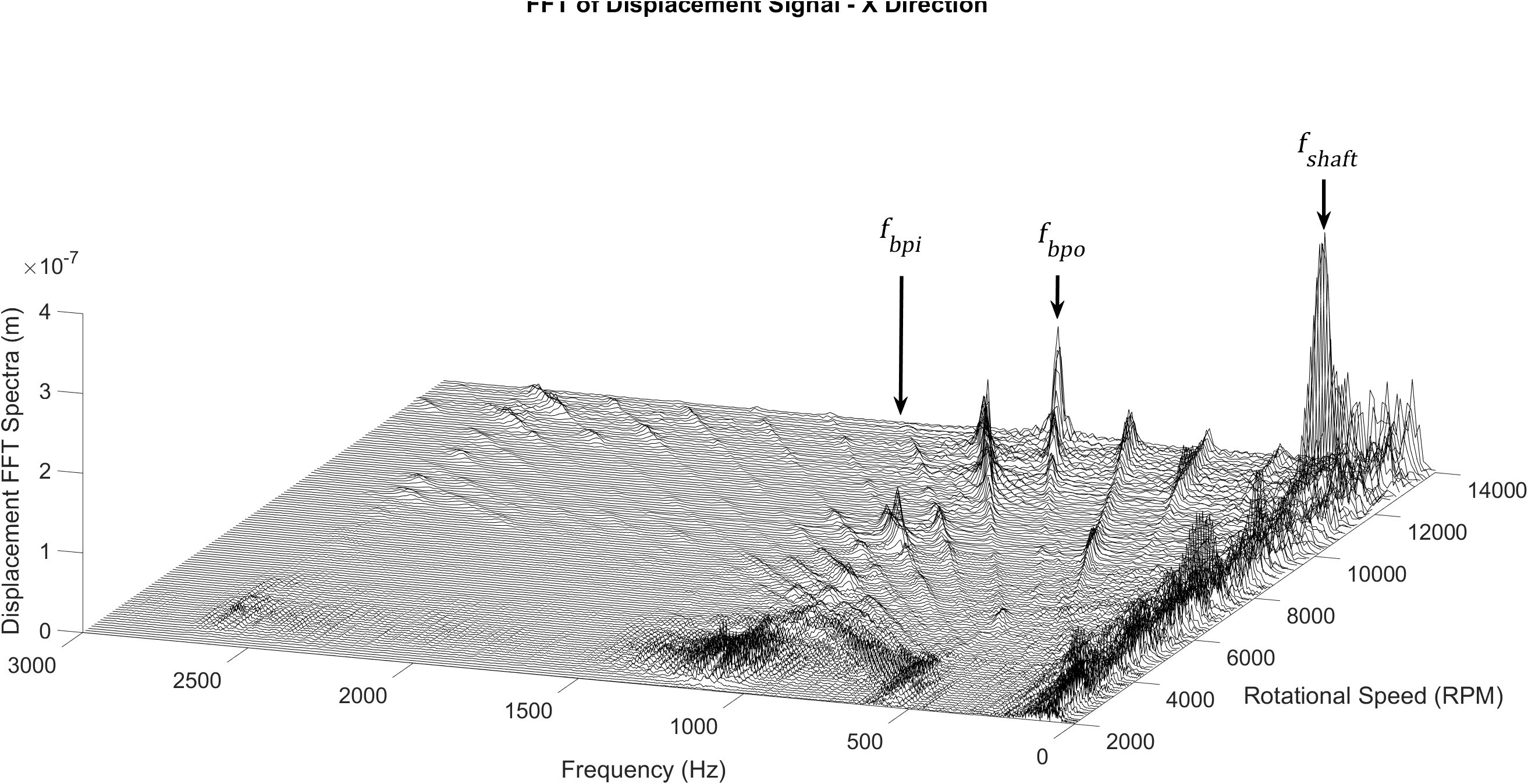


Figure 7 - Bearing Bore Displacement Frequency Spectra

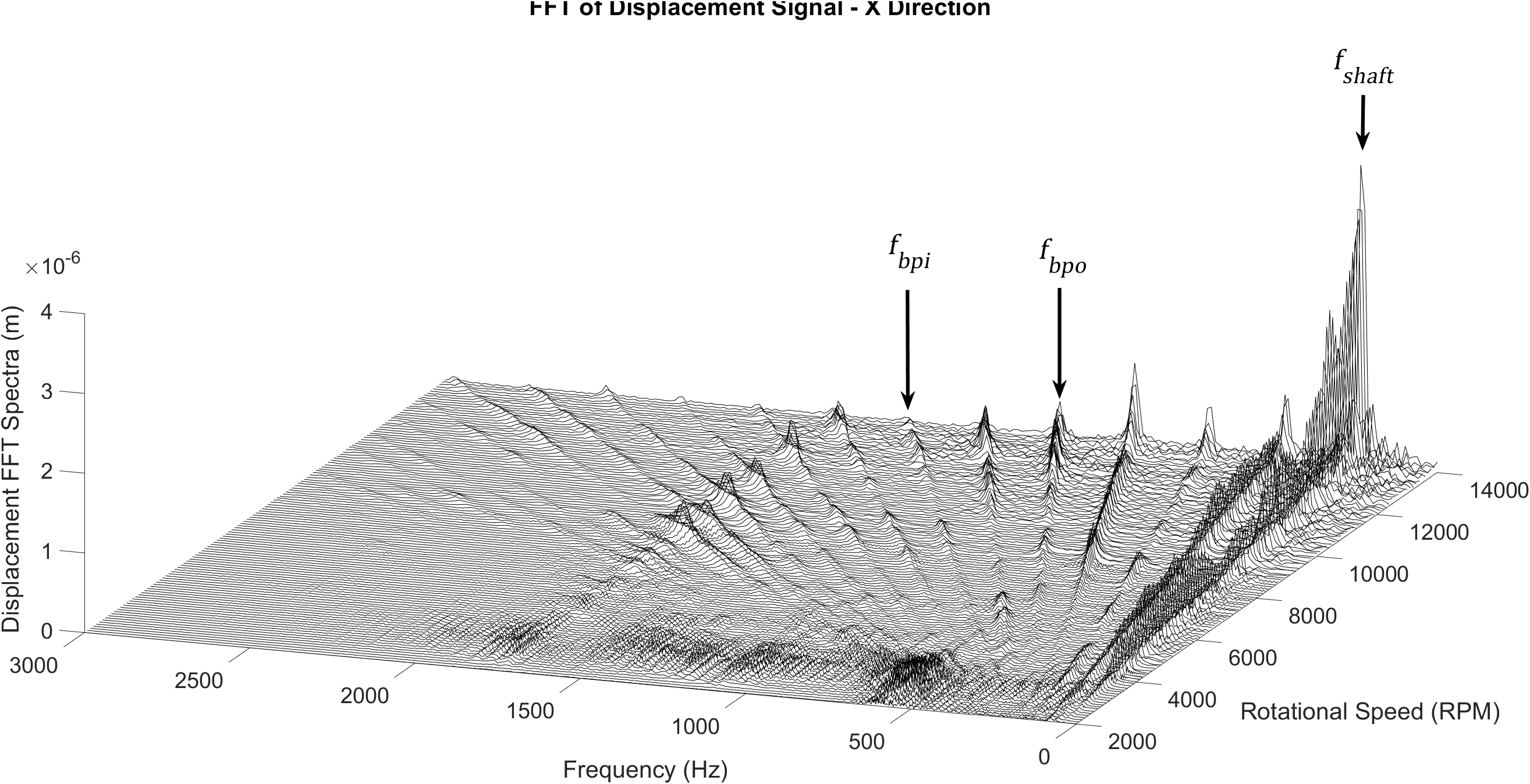


Figure 8 - Shaft Displacement Frequency Spectra

* + 1. Dry vs Lubricated Contact Model

The contact mechanics model was run both dry and lubricated. At high speeds during periods of resonance, the deflection due to the large magnitudes of bearing displacement dominated and the percentage difference in load between dry and lubricated models is 1.9%. However, after a period of resonance at 14,900rpm, the film dominates deflection in the contact region and percentage differences as high as 16% are observed.

* + 1. Film Thickness and Load Across Speed Sweep

The variation in EHL film thickness and roller contact load across the speed sweep from 0 - 15 000 rpm at the outer race contact are shown in Figure 11. Only the EHL regions are shown, where loads are significant enough to cause contact deformation. The upper limit of the film is where the roller and races diverge and approach the hydrodynamic regime where the film is hence governed by the entraining motion of lubricant into the contact. The lubricant film, as seen from the film thickness equation, is more affected by the speed of entraining motion rather than the load. This explains the increasing film thickness values in Figure 11 despite increasing load. The film thickness is increased from 0.1 to 1.9 µm across the speed sweep, revealing a significant increase that can affect the tribo-dynamic behaviour of the bearing, as explained in following sections. From the tribological model, it is possible to observe the transition between mixed-EHL to the purely hydrodynamic regime as the roller passes in and out of the loaded region of the bearing. Full system and rotor dynamics also contribute to the total load on the roller, with periods of resonance at 3 000 rpm and 14 000 rpm marked as A and B respectively in Figure 11.

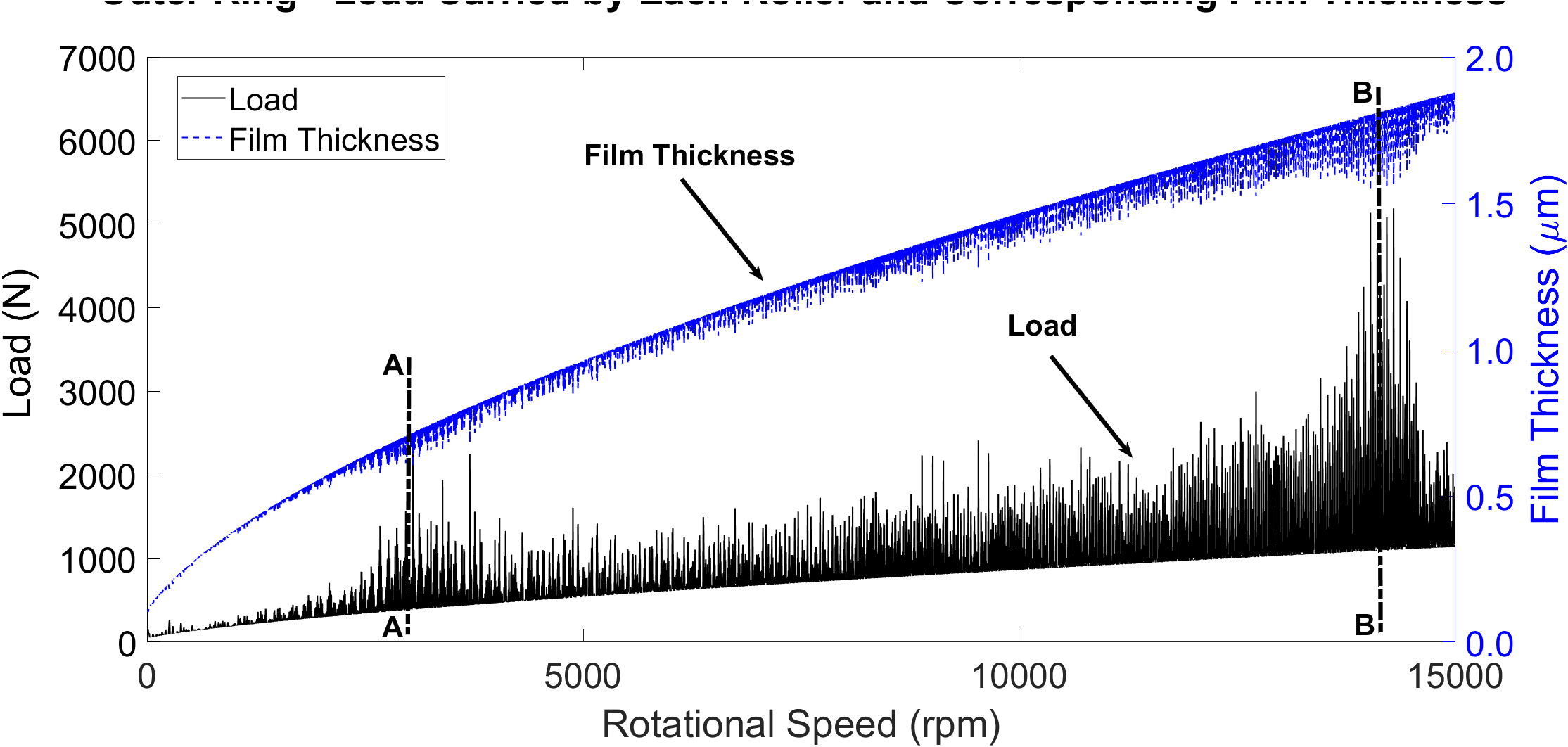


Figure 11 - Contact Load and Film Thickness - Outer Race

Figure 12 presents an interval of the speed sweep where the effect of the EHL load on reducing the film thickness under oscillating conditions can be observed and the hydrodynamic film growth as the roller is unloaded. It is possible to see the effect of the resonant frequency at 1765 Hz superimposed on the lower fundamental train frequency within the loaded region as the inner and outer rings converge and diverge. The results in Figure 12 confirm the significant effect of dynamic behaviour as well as multi-regime nature of the lubrication due to dynamic effects.

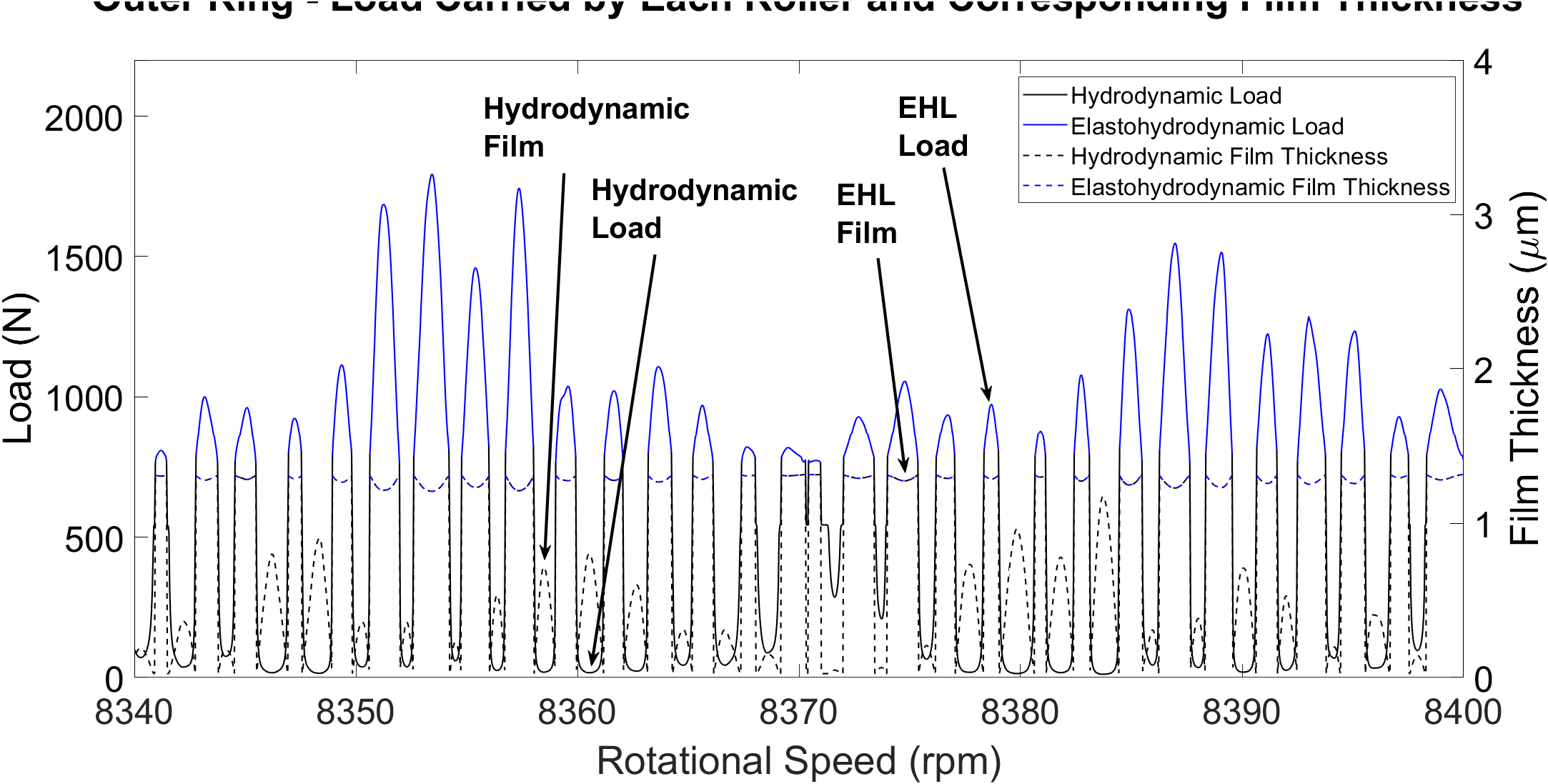


Figure 12 - Film and Load - EHL to Hydrodynamic Regime

* + 1. Friction

Focussing on data from the EHL regime, boundary and viscous friction through the speed sweep can be analysed. Figure 13 shows the boundary and viscous frictions across the speed sweep. As the film thickness increases with speed, the separation of the contacting surfaces increases, reducing the boundary interaction of asperities. The resonant period at 14 000 rpm reduces the film height, increasing the likelihood of asperity interaction in that region and hence boundary friction and potential for wear. In general, the boundary friction reduces towards higher rotational and entrainment velocities. This does not, however, account for lubricant inlet starvation at high speeds or roller sliding. Viscous friction increases with higher speed and hence the shear as expected. Although the film thickness also increases at higher speeds which may reduce the amount of shear, the effect of increasing speed is more dominant. The peak values occurring again at a resonance speed of 14 000 rpm where the highest loads and smallest film occurs.

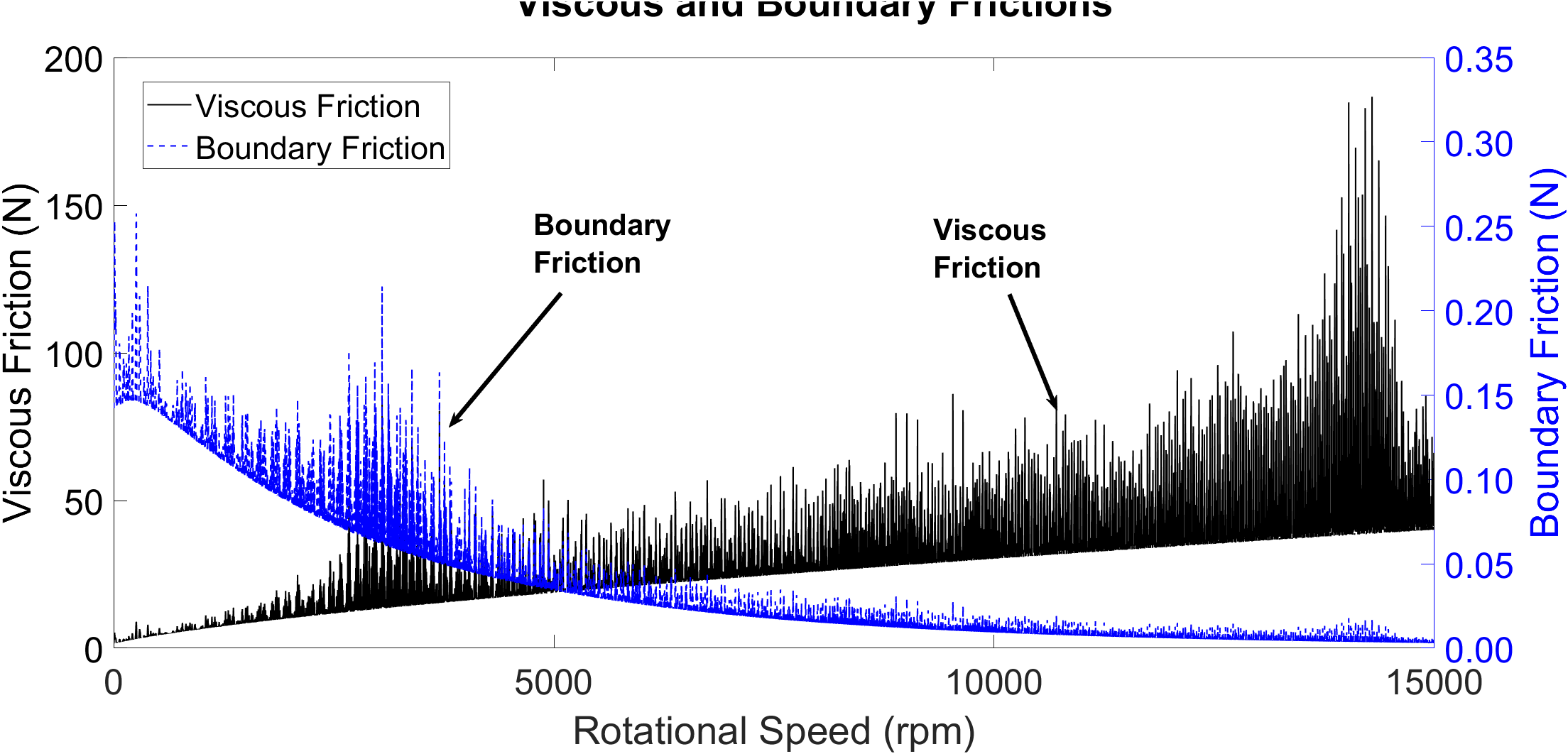


Figure 13 – Viscous and Boundary Friction at Outer Race Contact

* + 1. EHL Regimes

As has been demonstrated in the results analysis, the contact conditions deviate from the elastohydrodynamic regime of lubrication into the hydrodynamic regime throughout the roller orbital motion. These conditions can be verified by presenting the results on the Greenwood chart for fluid regimes of lubrication. The charts display the physical effects instrumental to EHL formation under isothermal conditions: viscosity rise due to pressure and elastic deformation of the surface. These two effects are quantified by two dimensionless parameters, and , as defined below (Gohar, 1988):

|  |  |
| --- | --- |
|  | [37] |

|  |  |
| --- | --- |
|  | [38] |

Four regions exist and the boundaries of these regions are defined by the geometry of contacting bodies, material, and lubricant properties. As is shown in Figure 14, the outer roller-race contact conditions move between the viscous elastic and iso-viscous rigid regimes, representing EHL and hydrodynamic respectively. The viscous elastic regime signifies the EHL regime of lubrication where contact pressures are such that elastic deformation of the surfaces and viscosity rise due to pressure increase is significant. The iso-viscous rigid regime occurs when the magnitude of elastic deformation is insignificant, and the contact pressures are low enough that viscosity rise is negligible, i.e. hydrodynamic lubrication (Hamrock, 1980). The boundary between the two also corresponds well with the distinction being made between hydrodynamic and EHL in this methodology, presented by the black and blue regions of the plot.

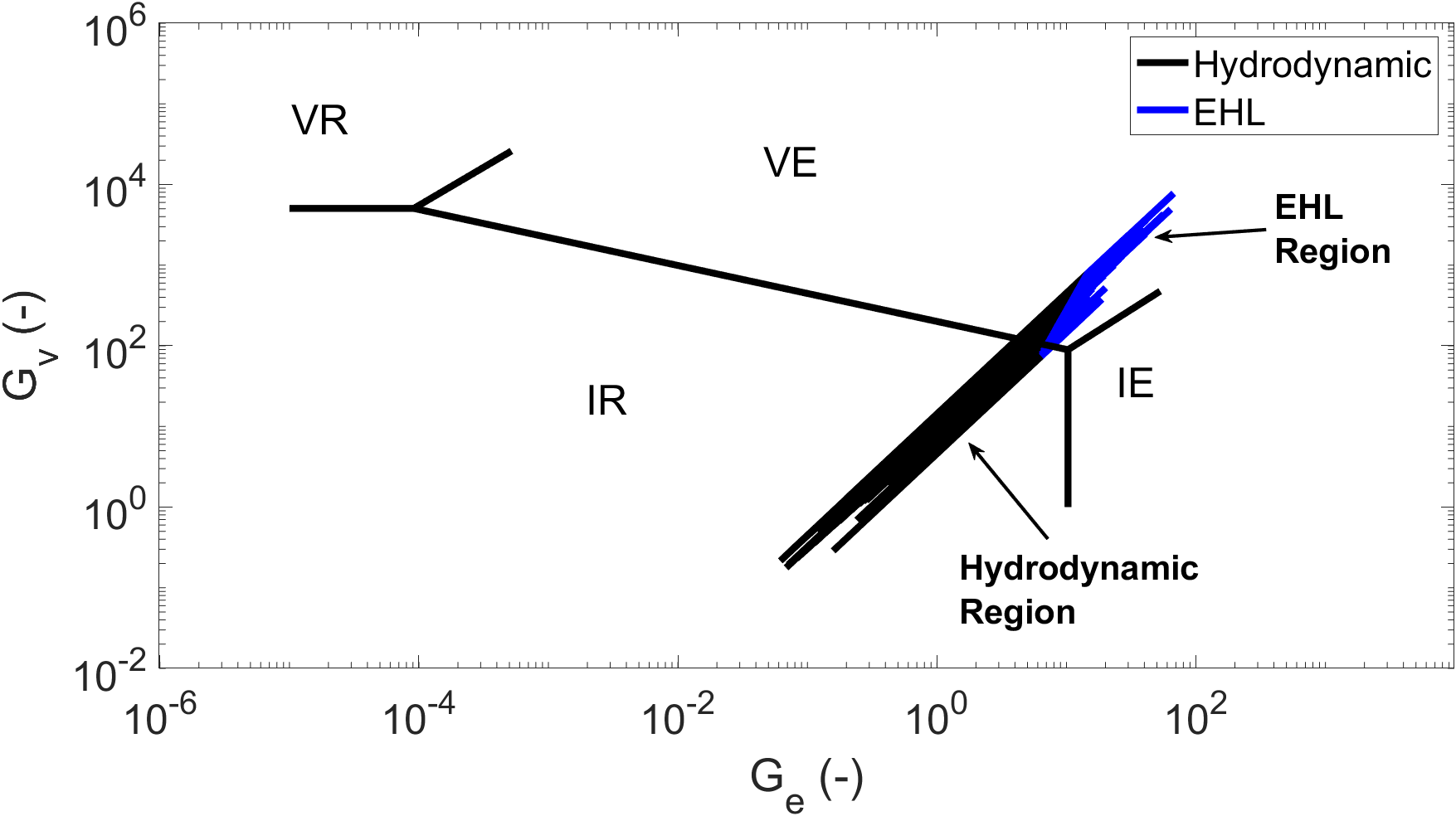


Figure 14 – EHL and hydrodynamic conditions during roller operation. Key: IR = Iso-viscous Rigid, VR = Viscous Rigid, VE = Viscous Elastic, IE = Iso-viscous Elastic

* + 1. Dry vs Lubricated Tribo-Dynamic Model

Previously presented results confirmed the significance of considering tribo-dynamic coupling on the tribological predictions. The aim of this section is to assess the significance of this coupling on dynamics via affecting contact load and stiffness values. The surface deformation at the EHL contact is further exacerbated by the presence of the lubricant film. Since the contact load and contact stiffness are governed by this deformation, neglecting the film leads to an underestimation of the total load at the roller-race contact. At higher speeds, such as those present in modern electrified powertrains, the growth of the lubricant film due to the increased entraining motion into the conjunction cannot be neglected – as is shown in Figure 11 with a film growth from 0.1 to 1.9 µm. The implicit tribological model was run for two cases, including and negating lubricant film thickness in the deformation obtained from equation 5. The difference in magnitude at each time step is computed, and the increase in load magnitude between a dry and lubricated model is calculated. For EHL loads, the magnitude of the load difference through the speed sweep is plotted in Figure 15.

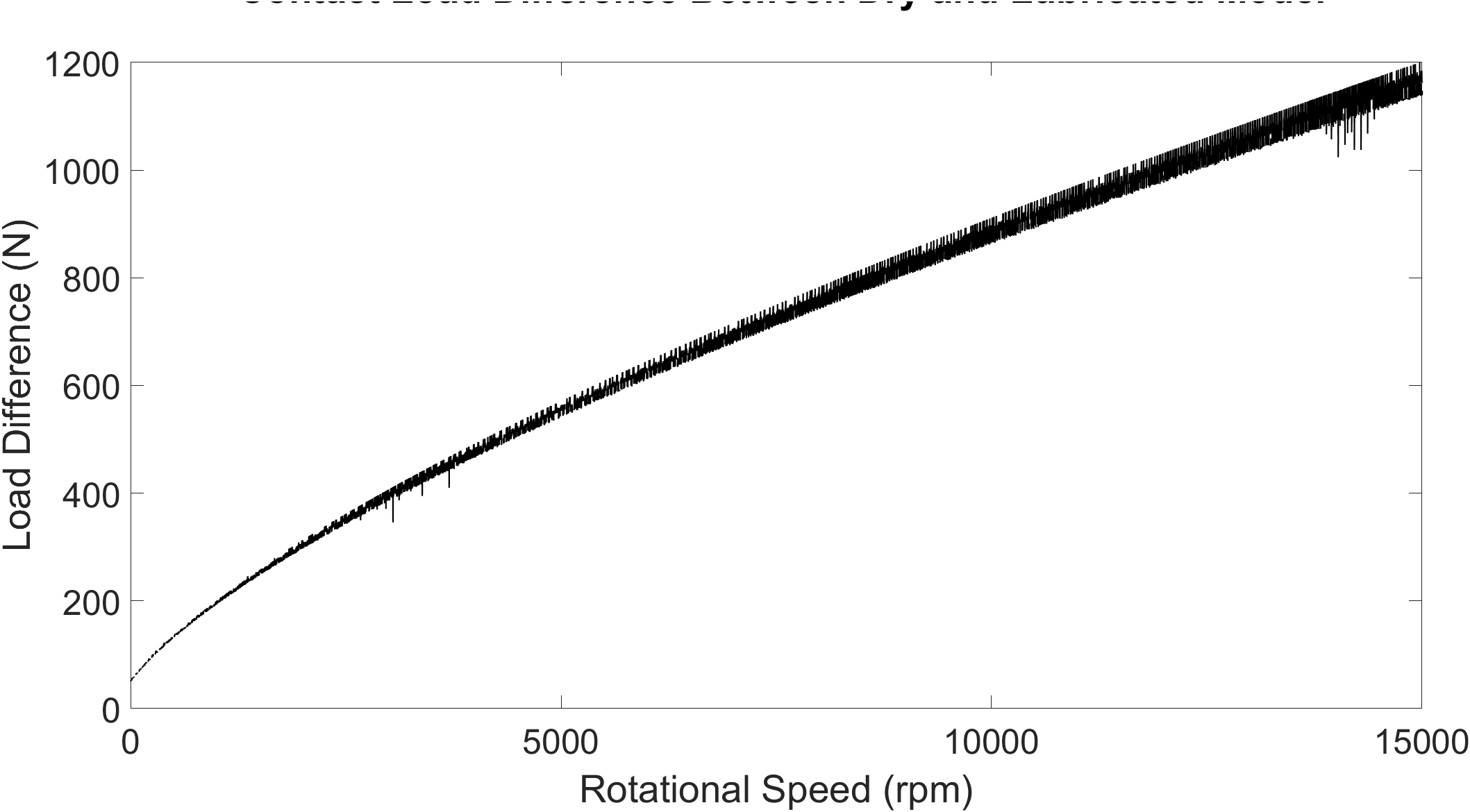
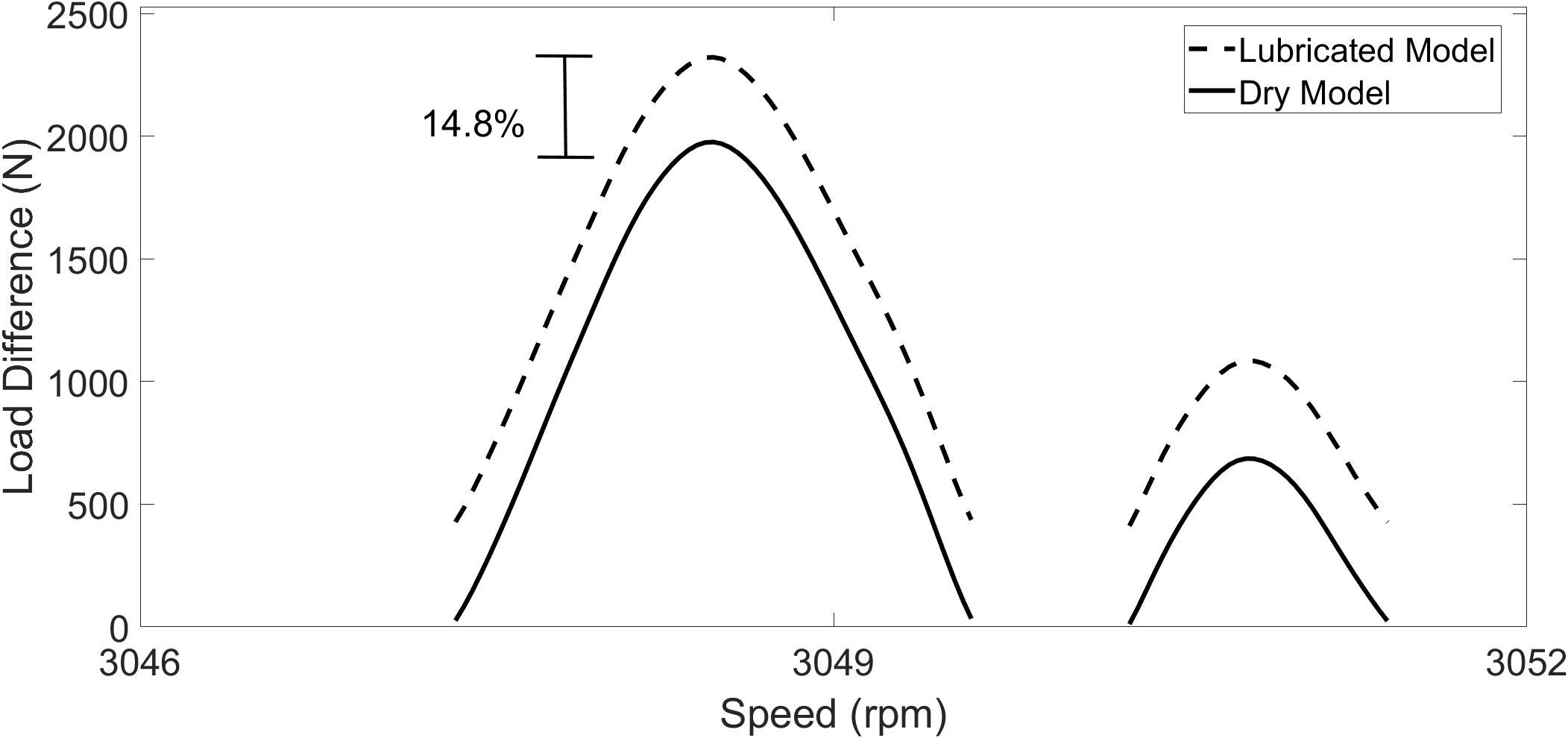
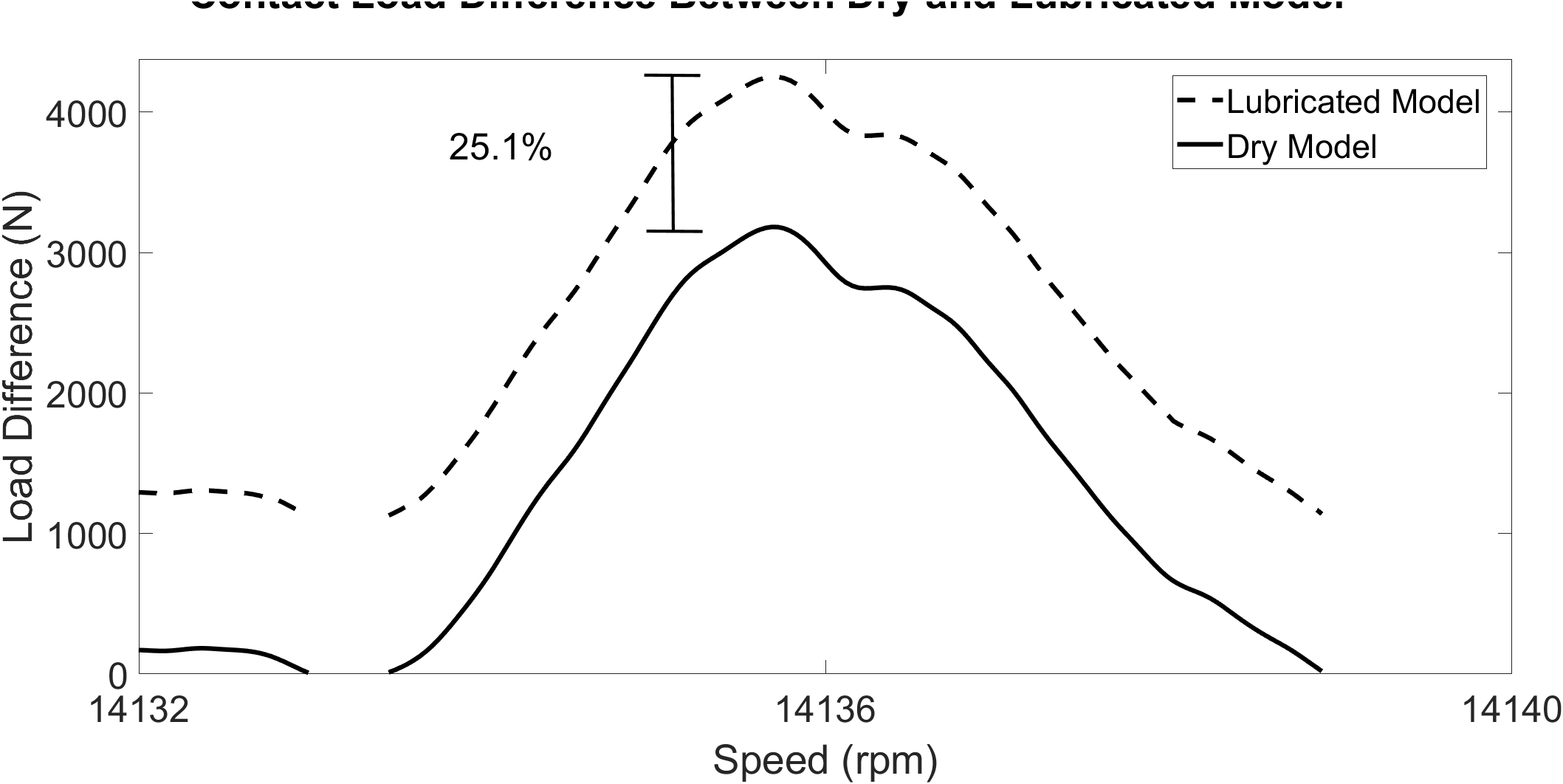


Figure 15 - Contact Load Difference Between Dry and Lubricated Model

There is an increasing load difference across the speed sweep, with fluctuations arising from the dynamics of the system. It is clear that neglecting lubricant film thickness leads to significant underestimation of the contact load, hence inaccurate dynamics as well as tribological calculations. This effect gains more significance at higher speeds which highlights the necessity of considering tribo-dynamic coupling for high-speed conditions in electrified powertrains. To fully understand the requirement for a lubricated bearing model, the percentage difference between both cases is presented at three different rotational speed snapshots of 3 050 rpm, 14 135 rpm, and 14 855 rpm in Figure 16. At low speeds and relatively low dynamic load, the addition of the film contributes to a 14.8% greater load prediction than a dry model, showing that the film inclusion has a significant contribution even at low rotational speeds. At shaft speeds of 14 135 rpm, the first order resonance in the system, as shown in the frequency plots, creates high dynamic loading. At peak load, the growth of the film is still present, however, the high contact deformation is close to the magnitude of the film growth, hence the difference between dry and lubricated model reaches 25.1%. As the system passes through this resonant region and the overall dynamic load is reduced, the percentage load difference reaches values as high as 149% as the effect of the film growth at high speeds exceeds that of the surface deformation.

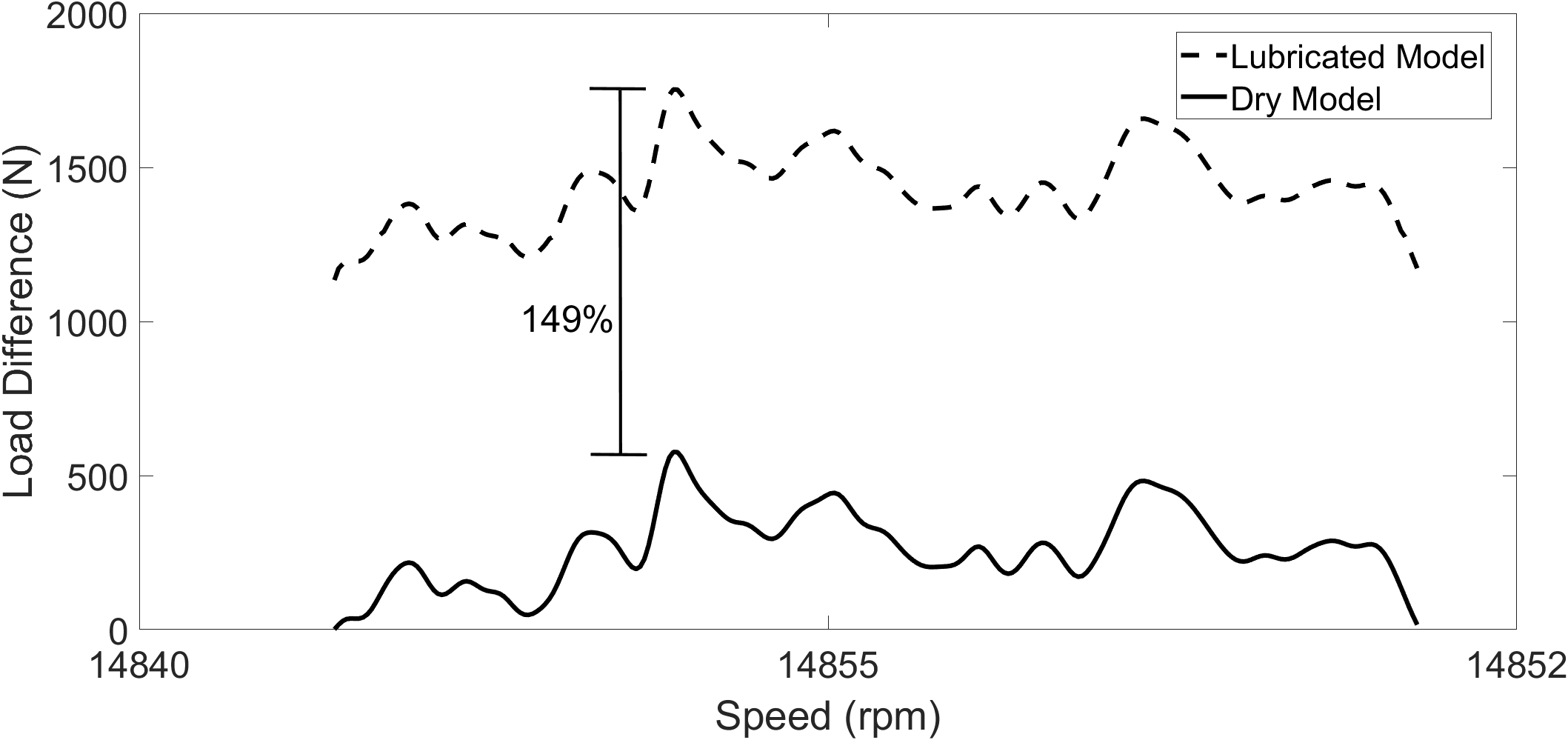


Figure 16 - Dry and lubricated model load difference: a) 3 050 rpm, b) 14 135 rpm, c) 14 855 rpm

At high speeds in periods of resonance, the magnitude of the bearing load dominates, and the effect of the increasing film thickness with speed diminishes in regions of resonance. The percentage difference between the dry and lubricated model is lower as the external force and corresponding surface deformation prevails the effect of the film. However, at high speeds outside of this period of resonance, the film thickness is of the same order as the deformation and the percentage difference between the two models is much greater.

* + 1. Numerical EHL Results

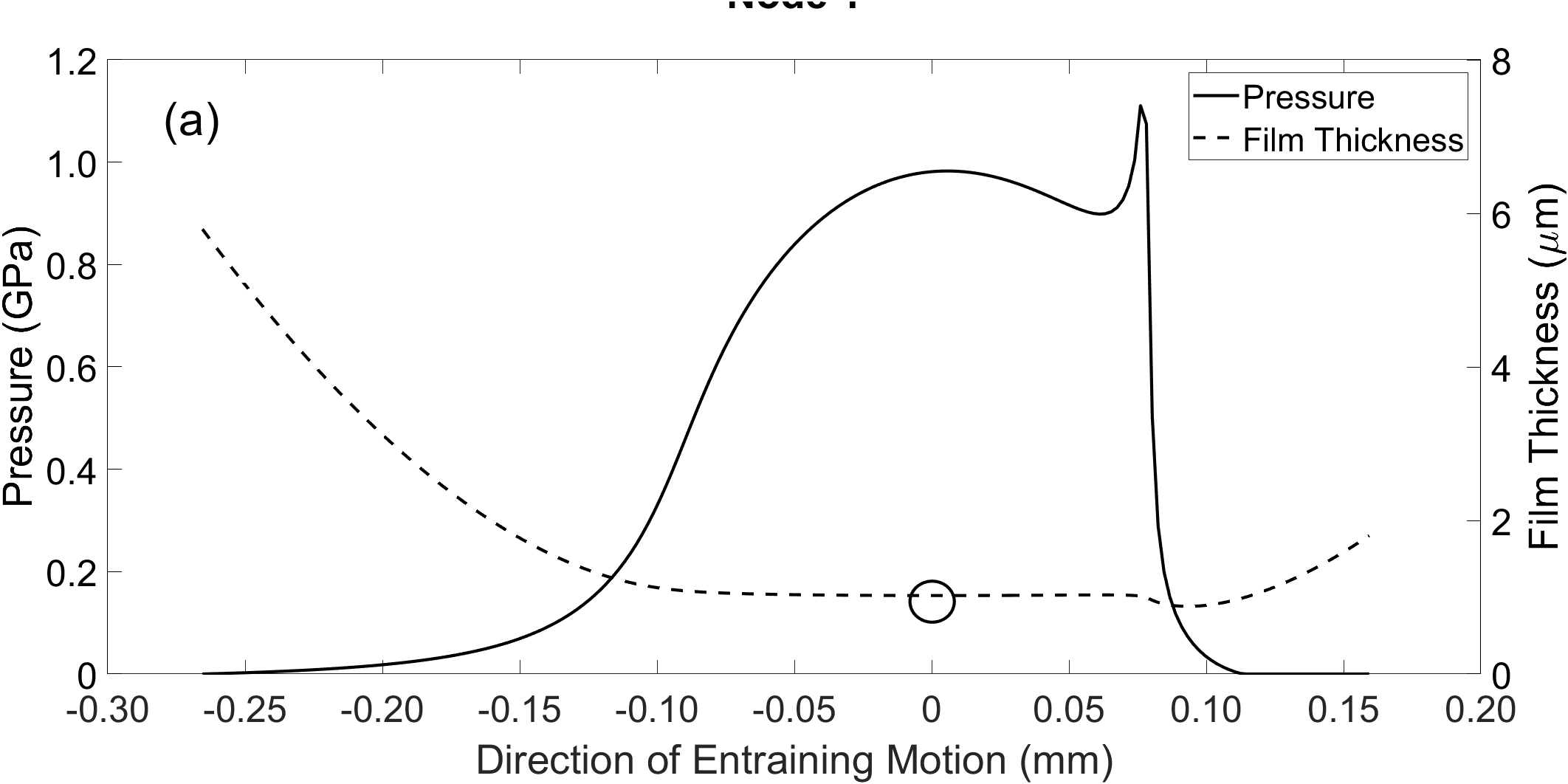
Full numerical simulation is required to obtain detailed pressure and film thickness distributions. These distributions reveal the realistic pressure and film values at the contact for in depth durability, efficiency and thermal analysis. At 8 350 rpm, focussing on one roller orbit, the selected points for EHL numerical analysis are shown in Figure 17. These load values are found from the implicit tribological model when the roller enters the loaded region of the bearing. The corresponding points on the bearing circumference are also shown.

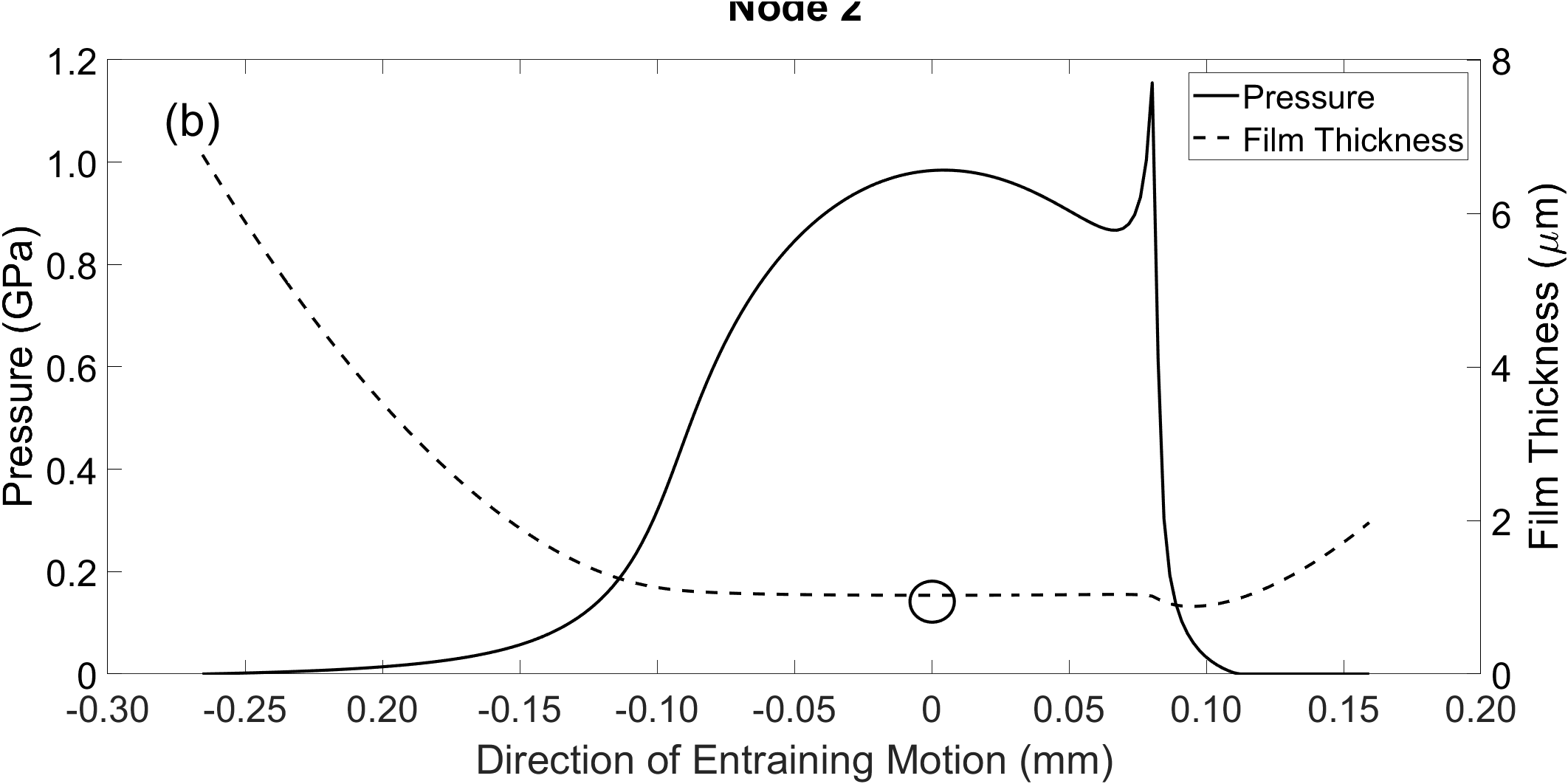
A screenshot of a cell phone

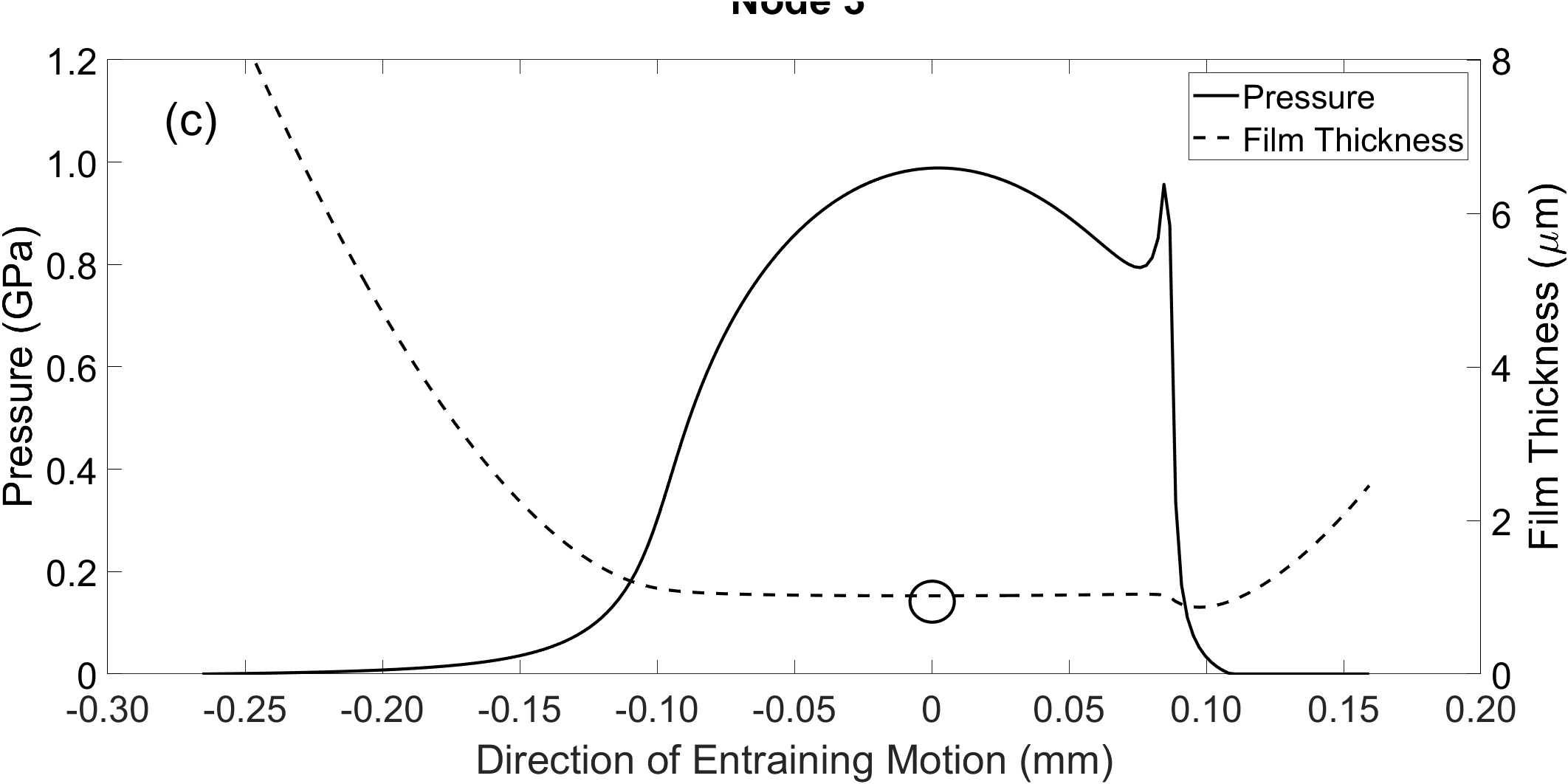
Description automatically generated

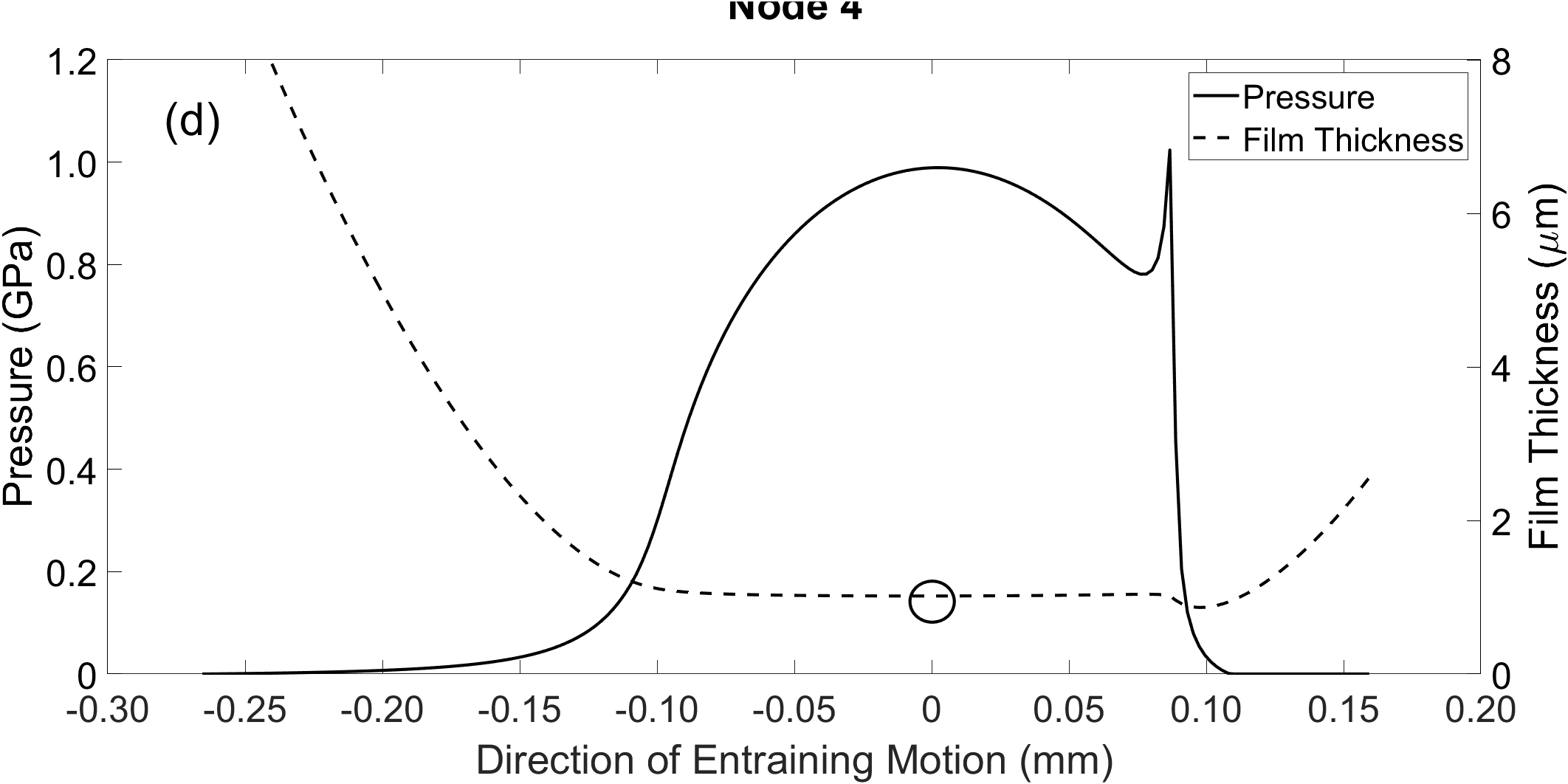
Figure 17 – Selected Points for Numerical EHL Model

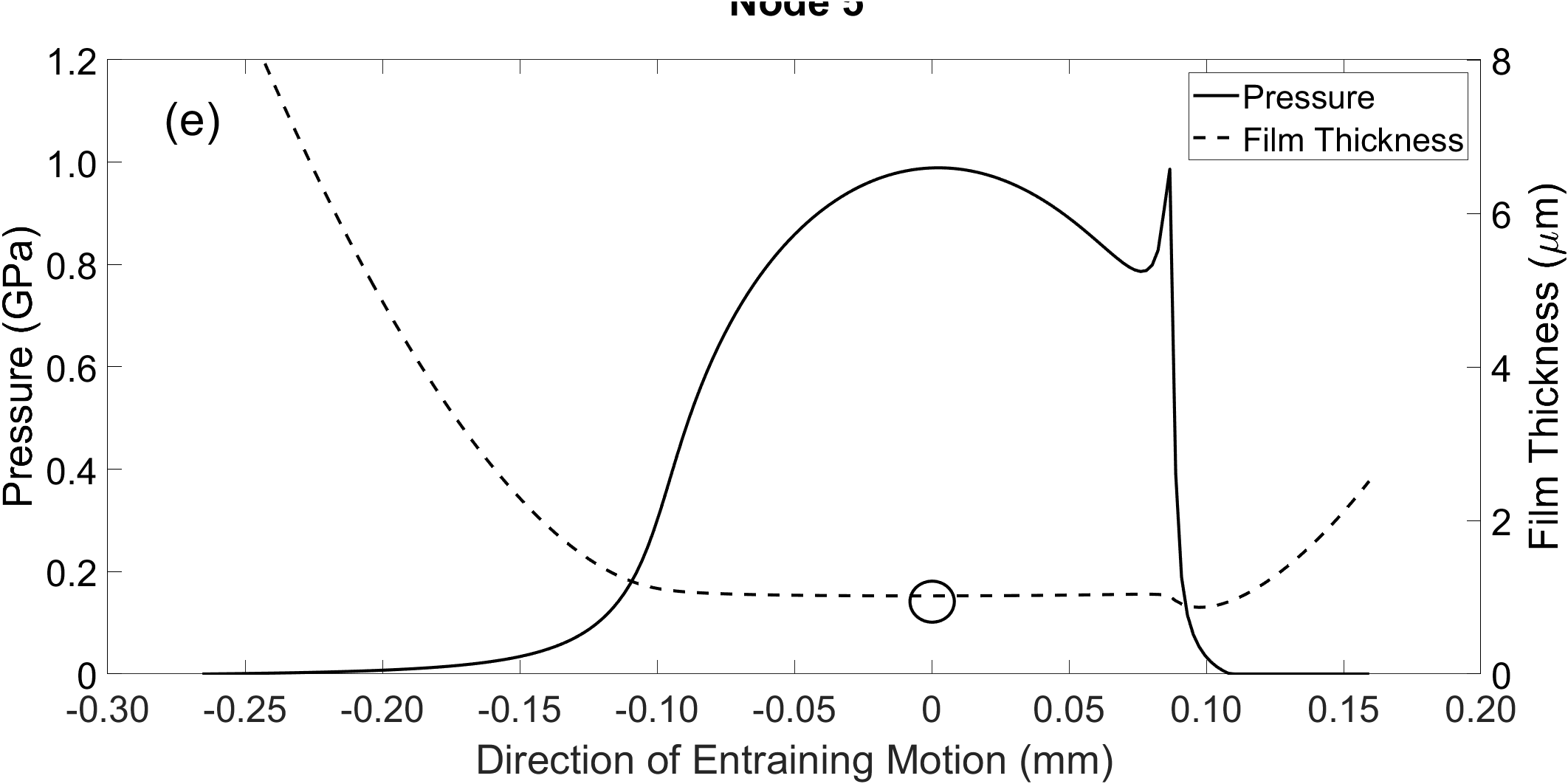
The load values are passed explicitly to the numerical EHL model along with entrainment velocity, lubricant and solid properties. From the nodes presented, the pressure distribution and film thickness across the contact are obtained. These are presented in Figure 18.











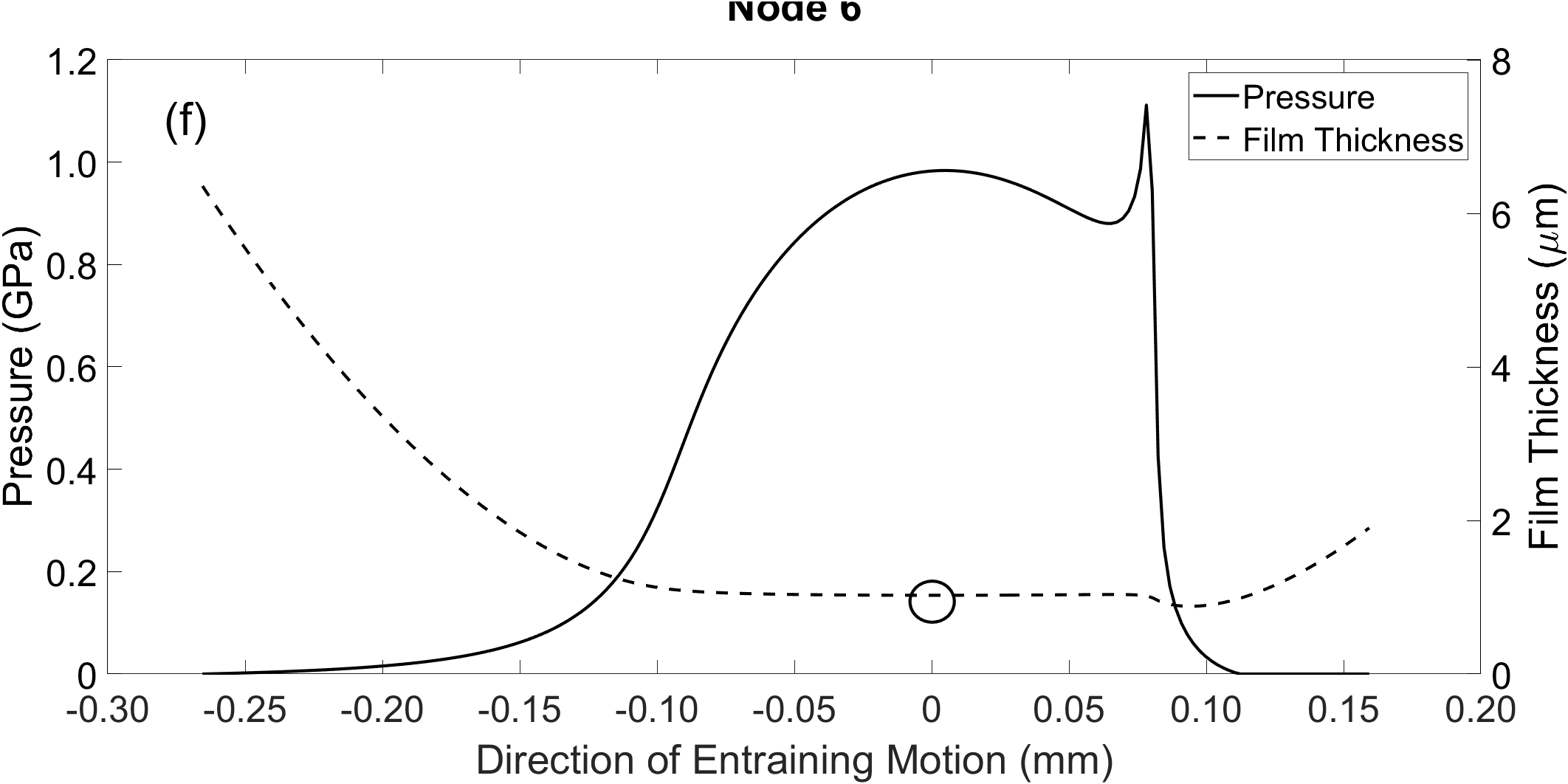


Figure 18 - EHL Pressure and Film Thickness Distributions: a) Node 1, b) Node 2, c) Node 3, d) Node 4, e) Node 5, f) Node 6

In each of these plots, it is possible again to see the central film thickness drop as the roller enters, then exits the loaded region. These central film values are represented by circles in Figure 18. Agreement between the extrapolated film formulae and the numerical model for calculating central film values are presented in Table 4. This confirms the validity of using extrapolated film thickness equation in the implicit tribological model. It is shown that the central nodes which correspond to the higher loads in the loaded region have the lowest percentage difference in comparison to the lightly loaded nodes at the outer edges.

*Table 4 – Extrapolated film formulae and numerical model central film thickness comparison*

|  |  |  |  |
| --- | --- | --- | --- |
| **Node** | **Extrapolated Formulae Central Film Thickness (µm)** | **Numerical Model Central Film Thickness (µm)** | **Percentage Difference (%)** |
| 1 | 1.142 | 1.026 | 11.3 |
| 2 | 1.122 | 1.025 | 9.46 |
| 3 | 1.075 | 1.019 | 5.49 |
| 4 | 1.068 | 1.016 | 5.11 |
| 5 | 1.072 | 1.018 | 5.30 |
| 6 | 1.122 | 1.025 | 9.46 |

## **Conclusions**

A new methodology comprising experiments and numerical modelling has been developed to allow component and conjunction level tribo-dynamic analysis of a roller bearing under speeds and loading conditions previously not reported. The experimental data contain the physics of the dynamics and tribology within the bearing, negating the need for a simplified and computationally intensive dynamic bearing model. The tribological conditions at the contact between an individual roller and raceways are numerically analysed. All possible lubrication regimes, including mixed-EHL and hydrodynamic, are considered as the roller passes through loaded and unloaded regions.

The presented research investigates a new range of working conditions under high speeds, representative of modern electrified powertrains. This study helps to understand the prevailing regimes of lubrication as well as range of tribological quantities. Additionally, the interaction of tribological behaviour with dynamics of the system is investigated. The deeper and comprehensive understanding of these matters will support objective development of electrified powertrains in the future towards higher efficiency, durability and NVH refinement. The acquired knowledge and understanding will also support future developments of predictive tools by understanding the interaction of tribology and dynamics and the significance of considering this multi-physics interaction. The following conclusions can be made based on presented results:

* The contact experiences an order of magnitude increase in film thickness across the speed sweep. This highlights the necessity of implicitly including the lubricant film in any predictive tribo-dynamic model; an approach that is not reported on in open literature for high-speed dynamic roller bearing analysis.
* A comparison between the dry and lubricated contact model further reinforces the need to include the lubricant film in high-speed roller bearing analysis, with percentage differences in load between both models up to 149% at 15 000 rpm. Neglecting this film by using the common dry Hertzian approach would lead to an underestimation of the total load at the roller-race contact.
* At higher speeds, such as those present in modern electric powertrains, it is shown that the growth of the lubricant film must be included implicitly within the dynamic bearing analysis.
* The load values obtained from the lubricated tribological model have been used explicitly within a 1-dimensional EHL model to calculate the pressure distribution and film thickness across the contact. The workflow of using an explicit EHL model based on the analytical tribological contact mechanics is valid, with good agreement between central film thickness values for both.
* The explicit EHL approach significantly improves the computational efficiency of the model whilst maintaining accuracy, since only the central value of the film is required in the load calculation. This can be implicitly coupled to broaden the scope of the analysis that is being performed.

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