Multi-scale Finite Elements Modeling of Rubber Friction Toward Prediction of Hydroplaning Potential

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Hydroplaning is a phenomenon that occurs when a layer of water between the tire and pavement pushes the tire upward. The tire detaches from the pavement, preventing it from providing sufficient forces and moments for the vehicle to respond to driver control inputs such as breaking, accelerating and steering. This work is mainly focused on the tire and its interaction with the pavement to address hydroplaning. Before using a full-scale tire model, interactions of the tread block with a specific surface is studied. To do so, several mechanical tests such as uniaxial, biaxial, planar (shear), and DMA are conducted to predict the hyper-viscoelastic properties of the rubber. Using multi-scale modeling techniques, the friction coefficient between the tire and pavement, for wet conditions, is characterized via developing 2D and 3D model representing the rubber tread interacting with the rough surface.

Using a tire model that is validated based on results found in the literature as well as in-house experimental data, fluid-structure interaction (FSI) between the tire-water-road surfaces are investigated through two approaches. In the first approach, the coupled Eulerian-Lagrangian (CEL) formulation was used. The drawback associated with the CEL method is the laminar assumption that the behavior of the fluid at length scales smaller than the smallest element size is not captured. To improve the simulation results, in the second approach, an FSI model
incorporating finite-element methods and the Navier-Stokes equations for a two-phase flow of water and air, and the shear stress transport \( k-\omega \) turbulence model, was developed and validated, improving the prediction of real hydroplaning scenarios. The improved FSI model was applied to hydroplaning speed and cornering force scenarios. In addition, tire contact patch length was calculated using the developed FSI model and was compared to the results obtained from the intelligent tire.
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GENERAL AUDIENCE ABSTRACT

Hydroplaning is a phenomenon that occurs when a layer of water between the tire and pavement pushes the tire upward. The tire detaches from the pavement, preventing it from providing sufficient forces and moments for the vehicle to respond to driver control inputs such as breaking, accelerating and steering. Hydroplaning as well as low skid resistance are considered as the main factors leading to traffic accidents. This work is mainly focused on the tire and its interaction with the pavement to address hydroplaning. Different factors involve in the hydroplaning phenomenon such as water film thickness, tire pressure, tire tread pattern, tire tread depth, vehicle speed and pavement texture. Before using a full-scale tire model, interactions of the tire tread with a specific surface is studied. To do so, several mechanical tests are conducted to predict the hyper-viscoelastic properties of the rubber. Using a single scale methodology is not capable to obtain the sufficient information regarding the effect of roughness on the friction. As a result, using multi-scale modeling techniques, the friction coefficient between the tire and pavement, for wet conditions, is characterized via developing 2D and 3D model representing the rubber tread interacting with the rough surface.

Since in the hydroplaning problem, a solid structure and a fluid domain are in interaction, such a problem considered as a fluid-structure interaction (FSI) problem. In this work, the FSI between
the tire-water-road surfaces are investigated through two approaches. To improve the simulation results, an FSI model incorporating finite-element methods and the Navier-Stokes equations for a two-phase flow of water and air, and the shear stress transport k-ω turbulence model, was developed and validated, improving the prediction of real hydroplaning scenarios. In addition, tire contact patch length was calculated using the developed FSI model and was compared to the results obtained from the intelligent tire.
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$b$</td>
<td>Body force</td>
</tr>
<tr>
<td>$c_{10}$</td>
<td>Constant describing the shear modulus $G = G/2$</td>
</tr>
<tr>
<td>$d$</td>
<td>Displacement</td>
</tr>
<tr>
<td>$D_1$</td>
<td>Constant describing the bulk modulus $K = 2/k$</td>
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<tr>
<td>$e$</td>
<td>Energy density</td>
</tr>
<tr>
<td>$E_H$</td>
<td>Specific energy per unit mass</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravity</td>
</tr>
<tr>
<td>$K$</td>
<td>Bulk modules</td>
</tr>
<tr>
<td>$\bar{I}_1$</td>
<td>The first invariant of the deviatoric strain tensor</td>
</tr>
<tr>
<td>$\bar{J}_{e1}$</td>
<td>The elastic volume ratio</td>
</tr>
<tr>
<td>$W$</td>
<td>Strain energy potential (density) or stored energy</td>
</tr>
<tr>
<td>$\bar{W}$</td>
<td>function defined per unit volume.</td>
</tr>
<tr>
<td>$K$</td>
<td>Bulk modulus</td>
</tr>
<tr>
<td>$AR$</td>
<td>Footprint aspect ratio</td>
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<td>$p_i$</td>
<td>Inflation pressure</td>
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<tr>
<td>$v_{cr}$</td>
<td>Hydroplaning critical speed</td>
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<tr>
<td>$F_y$</td>
<td>Normal load on the wheel</td>
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<td>$WFT$</td>
<td>Water film thickness</td>
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<tr>
<td>$p$</td>
<td>Fluid pressure</td>
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<td>$p_H$</td>
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<tr>
<td>$\bar{\tau}$</td>
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<td>Effective diffusive rate</td>
</tr>
<tr>
<td>$Y_k$</td>
<td>Dissipation of turbulence kinetic energy</td>
</tr>
<tr>
<td>$Y_\omega$</td>
<td>Dissipation of Specific rate</td>
</tr>
<tr>
<td>$\mu_t$</td>
<td>Eddy-viscosity</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Volume fraction</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Mass density</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Cauchy stress</td>
</tr>
<tr>
<td>$h$</td>
<td>Grid size</td>
</tr>
<tr>
<td>$N$</td>
<td>Total number of grid cells</td>
</tr>
<tr>
<td>$r$</td>
<td>Grid refinement factor</td>
</tr>
<tr>
<td>GCCI</td>
<td>Grid convergence index</td>
</tr>
</tbody>
</table>
$p$  Apparent order
1 INTRODUCTION

Friction is defined as the resisting force when two surfaces are sliding relative to each other. Several frictions forces are introduced such as dry friction, lubricated friction, fluid friction, and internal friction. The friction can desirable in some applications such traction which make the motion possible. However, friction can cause some undesirable consequences such as wear that can reduce the performance of each mechanical system. It is believed that for the first time, the classic laws of the friction force are introduced by Leonardo da Vinci in 1493 [1-4]. However, archaeological evidences show that throughout the history, human being has been familiar with the friction concepts. For instance, remains of Persepolis (a world heritage site constructed date back to 515 BC) shows that the principles of the friction are considered in this magnificent site. The 20-meter-tall columns used in the building of Apadana (a palace made of stone columns in Persepolis shown in Figure 1-1) enjoy several parts that are fixed to each other using a pin mechanism, demonstrating the application of friction.
The understanding of friction was further developed more extensively by Guillaume Amontons (1699), Bernard Forest de Bélidor (1750), Leonhard Euler (1750), John Theophilus Desaguliers (1734), Charles-Augustin de Coulomb (1785), Pieter van Musschenbroek (1762), John Leslie (1766), Osborne Reynolds (1866), Fleeming Jenkin (1877) and J. A. Ewing (1877).

In 20th century most of the studies conducted by the scientist have been focused on to developing the theories to explain the physical mechanisms behind the friction. Arthur Morin for the first time presented the coefficient of friction term shown in Eq. 1-1 and Frank Philip Bowden & David Tabor (1950) classified the friction into static and dynamic frictions.

$$\mu = \frac{F_{\text{friction}}}{F_{\text{normal}}}$$

1-1
When two surfaces that are in contact has no relative motion, the friction between the surfaces is considered static friction. While the friction between the surfaces that have relative motion is considered dynamic friction. Since behavior of the dynamic friction is highly complex, a universal model to characterize the dynamic friction coefficient does not exist. Accordingly, the friction coefficient characterization has been always a hot topic in tribology. One of most influential application of the friction is the vehicle traction that caused by the interaction of the tire and pavement. The interaction of the tire and road surface (pavement) even become much more complex when the surface is wet. Depends on the water film thickness (WFT), this problem can be studied either as wet traction or hydroplaning phenomenon. The problem considered as a hydroplaning phenomenon when a layer of water between the tire and pavement pushes the tire upward. The tire detaches from the pavement, preventing it from providing sufficient forces and moments for the vehicle to respond to driver control inputs such as breaking, accelerating and steering.

1.1 Motivation and contributions

The potential for hydroplaning is a major safety risk facing the traveling public. Wet pavements contribute to about 16% of vehicle crashes in the United States, injuring over 350,000 people and killing over 4,400 each year [6]. It is believed that a part of these crashes is caused by hydroplaning. Previously, hydroplaning phenomenon has been considered as a road geometry design problem, and previous protocols for reducing the hydroplaning potential during road and highway geometric design were presented with regard to the limited number of experimental studies and definitions of hydroplaning that stress a single hydroplaning velocity [7]. Still, there exist a large number of the parameters involving in the hydroplaning phenomenon which a comprehensive study over the
influential factors assist to obtain more accurate interpretation of hydroplaning potential without relying on a single critical hydroplaning velocity.

During the last 10 years, using finite element modeling (FEM), a large number of studies have been conducted to characterize the risk of hydroplaning for different road and vehicle conditions. Most of the studies were focused on a full tire model with a constant friction coefficient as an input defining the interaction of the tire with pavement. The primary causes of the hydroplaning phenomenon are the water film on the pavement and the vehicle velocity. In addition, the friction between the tire and the pavement for wet condition plays a crucial role on the critical hydroplaning speed, longitudinal and cornering forces. The conventional hydroplaning modeling approaches mostly rely on solving a single-phase fluid as an Eulerian fluid elements which are assumed inviscid and laminar flow. Such an assumption places some limitations on the simplified hydroplaning model such as ignoring the effect of shear stress at the interface. Comparing the experimental results with simulation shows that the conventional simulation approaches can be improved towards more accurate predictions for the hydroplaning phenomenon. Accordingly, one of the key contributions of the current research is the development of an advance hydroplaning model using the incorporation of CFD turbulence and multiphase flow models with FEA for a more complete description of FSI coupling. In the new approach, the shear-stress Transport (SST) k-ω model has been coupled in the CFD model for its ability to predict the onset and flow separation from surfaces. The SST k-ω model combines the k-ω turbulence model to predict the boundary layer and the k-ε to predict the free stream flow.

In addition, the conventional FE modeling of a full-scale tire model cannot provide sufficient information to study the effect of the pavement roughness and aggregate size on the tire-pavement interaction. To address this issue, the roughness of the pavement is modeled using multi-scale
modeling techniques to characterize the friction coefficient between the tire tread and the rough surface. The hysteresis friction coefficient obtained from the developed multi-scale FE model, can be utilized in the hydroplaning model to incorporate the effect of the pavement roughness in hydroplaning analysis. Accordingly, one of the main contributions of the current work is developing a 2D and 3D multi-scale FE model to predict the friction and provide more comprehensive information from the interface of the rubber and rough substrate. one of the main motivations of extending a 2D FE model to a 3D multi-scale FE model for the rubber block sliding on the rough surface is enhancing our capability to obtain more information such as strain/stress contours within the contact surface.

The simulation results compared with the experimental data obtained from the tests conducted in the lab as a part of the CenTiRe project MODL-2017-B6-11. The presented methodology is able to consider the effect the roughness at different scales without the limitation associated with the physics-based model such as self-affine fractal assumption.

In this work, it is shown that the proposed hydroplaning modeling approaches are able to capture the effect of water on the net force as well as tire concerning force. This work also shows the differences between the two FSI approaches using Abaqus-CEL and Abaqus-Star-CCM+ coupling, due to different assumptions of viscous flow and free water surface tracking through tire splash interaction. In the new hydroplaning modeling approach, free surface flow with moving boundaries to build and track the amount of water as well as the volume fraction of other fluids can be captured, which can be used to study the tire tread design and tire aerodynamics. Development of two-phase flow model for hydroplaning analysis in the current research can also enhance the capability of the R&D engineers to study the air flow around the moving tire and
develop advance technologies to control the air flow near the tire surface towards developing tires with better splash, spray and aerodynamics performances.

In summary, the scientific contributions in this research are as follows:

I. Developed a multi-length scale 2D and 3D FE model to predict the hysteresis friction of sliding rubber tread

II. Considered the asperities of the specific rough surface to predict the friction

III. Improved FE model prediction via calibrating visco-hyperelaric material model using MATLAB genetic algorithm and 3 different loading modes

IV. Developed a fully coupled FE-CFD tire models to predict the hydroplaning potential and cornering forces with different tire tread depth, normal loads and slip angles [8-10]

V. Improved the real hydroplaning scenarios prediction using an FSI model incorporating finite-element method and Navier-Stokes equations for a two-phase flow of water and air

1.2 Organization of the document

The organization of this document is as follows:

- **Chapter 1: Introduction**

In this chapter hydroplaning concept as the key problem in this research is explained and the role of the friction in this problem and its importance throughout the history is elaborated. the motivation and the contribution are explained in this chapter.

- **Chapter 2: Literature Review and Background**

This chapter reviews the studies related to hydroplaning that have been conducted by other researchers. It also provides the background knowledge required to understand the current work. First, a literature review regarding the analytical friction models characterizing the rubber-surface
interaction is conducted. Then, different FE models and their assumptions are briefly reviewed, and different approaches to model hydroplaning is introduced.

- **Chapter 3: Rheological Properties of Rubber Materials for Tire FE Modeling**
  In this chapter, the experimental data used to characterize the hyperplastic material model are explained. In this work, since the sliding rubber experiences dramatic shear stress from the interaction with rough surface, apart from uniaxial and biaxial, the planer shear test data is also involved in the characterizing the material model to obtain more robustness. In addition, the viscoelastic properties of the rubber are characterized to be implemented in FE code using Prony series. To do so, a novel nonlinear programming and Genetic Algorithms (GAs) are implemented in MTLAB. The results compared with the experimental data to verify the developed methodology.

- **Chapter 4: Multiscale Modeling of Hysteresis Rubber Friction and Homogenization**
  In this chapter, the procedure used to obtain the surface profile of the rough surface is explained. Furthermore, homogenization technique is used to transfer the results from upper scale to lower scale. Since the interaction of the rubber with the rough substrate is influenced by several scales, a multi-length scale frictional constitutive model which is able to solve the contact problem between the rubber and rough substrate in lower scale and then transfer the result to upper scaling is the motivation of this chapter. The 2D FE model is also extended to the 3D model and the obtained results compared with the experimental data.

**Chapter 5: Hydroplaning Model Development**

This chapter explains the tire modeling procedures in ABAQUS. Since most the tire modeling steps in ABAQUS are done within the input file environment, it is useful for the researcher to follow this chapter which describes preprocessing in ABAQUS CAE, preparing input files, running ABAQUS Implicit, transferring the result to ABAQUS Explicit and postprocessing of the
output (odb) files. In addition, preparation of the FSI in Star CCM+ and running the co-simulation of the CFD and FE software packages will be discussed.

In this chapter, numerical methodology for the fluid flow model and the coupling between the two the FE and CFD codes are described. The shear-stress Transport (SST) k-ω model has been used the fluid flow and the volume of fraction (VOF) is used to model water and air, and their flow surrounding the tire, such as splash and spray.

- **Chapter 6: Conclusion and Future Work**

The summary of the obtained achievements in this work are presented in this chapter. In addition, this chapter explains the potential future work of this project. Multi-Scale modeling and testing of wet traction is a new project that has been approved by CenTiRe’s Industry Advisory Board (IAB). The 3D multi-scale finite element modeling (FEM) procedures developed here are going to be utilized in that project for further investigation.
Hydroplaning is a key issue for safe driving on wet roadways, and depends on vehicle velocity, water film thickness (WFT), tire construction, and tread pattern, and etc. Hydroplaning is more probable when the volume of the water in contact with the tire is more than the ability of the tire to expel the water. When the tire velocity reaches the hydroplaning speed, the force caused by the fluid pressure at the water-tire interface lifts the tire from the pavement. The risk of a vehicle accident when the pavement is wet is much more considerable than for dry conditions. Statistics from different regions of the world show that nearly twenty percent of traffic accidents happen on wet roads [11, 12]. Detailed information regarding the exact reasons for accidents during wet road conditions is not available. However, hydroplaning as well as low skid resistance are considered as the main factors leading to traffic accidents [13]. Accordingly, hydroplaning has become an important field of study in the automotive and tire industries as they try to address the factors affecting vehicle hydroplaning. Since it is expensive and difficult to create conditions in a laboratory environment that induce hydroplaning, multi-physics modeling and computer simulations are used to study the onset of hydroplaning.

Albert [14] discussed the effects of tire design parameters on hydroplaning and showed that the most important factor is geometric design of the tread pattern. Using a three-regions concept, Albert concluded that as the vehicle speed increased, the dynamic water pressure increased at the front edge of the tire and completely lifted the tire, leading to hydroplaning. Initial studies on hydroplaning have been mostly focused on physical experiments and empirical methods. Recently, improvements with computer resources have provided opportunities to solve complex problems and there have been efforts devoted to developing computer codes to study hydroplaning.
Hydroplaning involves the reactions when a layer of water impedes perfect contact between the tire and the road. Fluid-structure interaction (FSI) is the interaction of a moving deformed structure with fluid flow and can be used to model hydroplaning. FSI is an important and interesting phenomenon, but because it poses challenges for numerical modeling, several methods are utilized to solve FSI problems, such as the finite-element method (FEM) and the explicit finite-volume method (FVM). In fact, coupling FEM and FVM can improve numerical convergence in a stable configuration, while other methods can generate oscillations and can even produce unstable behaviors. Matthies et al. [15] presented a theoretical study of FSI in which the strengths and weaknesses of coupling FSI solvers were discussed. The displacements responses of weak and strong couplings were compared, and based on a numerical evaluation of the Jacobian for the complete system of equations obtained using separate solvers, a strong coupling approach was proposed. In the work by Walhorn et al. [16], a strong coupling method was presented for an FEM solver developed by the authors. In their approach, weak coupling problems due to time-marching schemes and exchanging information were formulated and solved simultaneously at each time step. The approach considered a nonlinear Lagrangian motion for the structure and a mixed Lagrangian-Eulerian formulation for the fluid. Kim et al. [17] applied a combination of finite-differencing and an FE tire model to iteratively predict the onset of hydroplaning. They compared approaches for a full-iteration method, a one-iteration method and no-iteration method and determined that the hydroplaning speeds of a straight-grooved tire predicted using the one-iteration method was almost the same as that obtained from the full-iteration method.

In a study that incorporated fluid motion, Vincent et al. [18] used a generalized formulation of the Navier–Stokes equations for a two-phase flow to simulate the interaction between the air–water flow and the tire. However, it is known that the flow experienced during hydroplaning is largely
turbulent in nature and should be considered. Ong and Fwa [19] indicated that the k-ε turbulence model can be used for flows at high or near hydroplaning speeds. The k-ε turbulence model is the simplest turbulence model but may not properly predict complex flows involving severe pressure gradients, separation, and strong streamline curvature [20, 21]. Guo et al. [22] selected the renormalization group (RNG) k-ε model since it can successfully predict high strain-rate flows and large bending fluid streamlines. Zhou et al. [23] chose the shear-stress transport k-ω model because it has advantages of high and low Reynolds number (Re) flows. The shear stress transport k-ω model utilizes mixed functions to achieve gradual transition from the standard k-ω model within the boundary layer to a high Reynolds number model outside the boundary layer and improves the transition from the near-wall region to the full development region with higher accuracy. The hydroplaning flow also involves two or more immiscible fluids, such as air, water and water vapor. Most researchers [13, 23-27] presented computational fluid dynamics (CFD) results using the volume of fraction (VOF) multiphase flow model by solving a single set of momentum equations and tracking the volume fraction of each fluid throughout the domain.

Several studies used commercial finite-element codes to predict hydroplaning for different conditions, employing approaches such as the arbitrary Lagrangian-Eulerian (ALE) and coupled Eulerian-Lagrangian (CEL) methods [12, 28-34]. However, these studies only solved the single-phase fluid as an Eulerian fluid element assuming inviscid, laminar flow. The assumption of inviscid flow ignores the effect of shear stress and the laminar assumption simplifies hydroplaning behavior thus neglecting the presence of turbulence. In addition, modeling fluids as an Eulerian element is not computationally practical since the fluid solution is dependent on the mesh size and requires the element size to be extremely small to resolve the fluid length scale. However, using
CFD can better capture the free surface flow with moving boundaries to build and track the amount of water as well as the volume fraction of other fluids with the VOF model.

In this work, hydroplaning is investigated using two FSI methods. The first method solves both structure and fluid using the structural solver within ABAQUS with the CEL formulation. The second method uses Star-CCM+ to model the two-phase turbulent flow of water and air, and the solutions are coupled with ABAQUS to model the tire deformation. First, the ABAQUS-CEL hydroplaning model is verified using empirical relations developed by NASA [35]. Second, the CFD approach is validated and numerical accuracy is determined to ensure grid-independent solutions. Next, the effect of hydrodynamic forces at different speeds lower than the critical hydroplaning speed, and the corresponding cornering forces at different slip angles are presented for both FSI approaches and compared with experimental data in the literature. It is shown that the proposed hydroplaning modeling approaches are able to capture the effect of water on the net force as well as tire concerning force. Besides the agreement, this work also shows the differences between the two FSI approaches using Abaqus-CEL and Abaqus-Star-CCM+ coupling, due to different assumptions of viscous flow and free water surface tracking through tire splash interaction.

2.1 Fluid-structure Interactions

Since utilizing the computational methods to consider the pavement asperities and tire parameters is not timely effective for many years, empirical methods have been always a solution for hydroplaning prediction. Apart from the empirical methods, advent of new computers provides the researcher with opportunities to study the effect of tire design such as tread patterns on
hydroplaning velocity. In fact, researchers are trying to design the tread grooves in a way that be able to guide the water flow efficiently through the channels created by the groove patterns.

An optimum designed tire pattern with regard to the vehicle and pavement condition can facilitate the process of discharging the water remained between the pavement and contact patch and as a result, the hydroplaning velocity increases.

For many years, the researchers and engineering have conducted a large number of simulations using the commercial finite element codes [17, 18, 36-40].

Coupled Euler Lagrangian (CEL) or other FSI formulation such as Arbitrary Euler Lagrangian (ALE) can be used to simulate FSI in hydroplaning simulation. In fact, we have to utilize the strength of both Eulerian and Lagrangian method to solve the FSI problem practically.

In the hydroplaning simulation process, we need to consider an Eulerian domain to the water and a Lagrangian domain for the solid structure (tire). This method is highly efficient in terms of the computational time and the mesh refinement and can reduces the computational time considerably.

Water layers around as well as the fluid interfere are divided into small size meshes. To have an optimum discretization to have an efficient code, the region that have more distance from the interfacial region obtain more coarse mesh sizes. If the splashing simulation would of the interest, a void layer over the Eulerian domain can capture the water drops when the tire moves through a pavement with water film thickness.

In [41] the effect of the pavement groove design on the hydroplaning velocity is studied. The simulation results are compared with the work done in [42]. Using the commercial packages, the finite volume method is implemented to study the effect of tread pattern on the hydroplaning velocity is shown in the following table.
Table 2-1. The dimensions of the groove pattern used in [41]

<table>
<thead>
<tr>
<th>Groove dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grooving design</td>
</tr>
<tr>
<td>A1</td>
</tr>
<tr>
<td>A2</td>
</tr>
<tr>
<td>A3</td>
</tr>
</tbody>
</table>

With regard to the result shown in Table 2-1, it is obvious that the design A1, with a tire inflation pressure of 186.2 kPa has more hydroplaning velocity compared to the design A2 and A3. The dimensions of the depth and width of the grooves are changed for designs A2 and A3, however, the hydroplaning velocity obtained for design A1 is much more than A2 and A3. It is shown that the groove parameters such as width and depth have more contribution on hydroplaning velocity than the other parameters such as spacing. In addition, the effect of inflation pressure on the hydroplaning speed is discussed in [43]. In this work, the effect of tread groove design on the hydroplaning velocity is explained. With regard to the simulation results shown, the increase in the groove depth is accompanied with increase in the hydroplaning velocity as long as the range of velocity would be less than 70 (km/h). Also, the effect of the groove width on the hydroplaning velocity is studied. It is shown that as long as the vehicle speed would be less than 105 (km/h) with the more groove width the more hydroplaning velocity will be obtained. Also, it is observed that if the vehicle speed would be less that 35 (km/h) the increase in hydroplaning speed can be obtained by reducing the space between the groves. In [44] the same procedure has been conducted to show
the effect of the groove dimension of hydroplaning velocity that has a very good agreement with the previous study. In both works, it has shown that the groove width has more contribution on hydroplaning velocity compared to the groove depth and space between the groves.

The direction of the pavement groves also affects the hydroplaning velocity. The summery of the results from simulation of smooth tire on the grooved pavement showing the effect of pavement grooving on the hydroplaning velocity is depicted in Table 2-2.

Table 2-2. The effect of pavement grooving shape on the hydroplaning speed for smooth tire

<table>
<thead>
<tr>
<th>Pavement grooving shape</th>
<th>Varied Parameter</th>
<th>Increase in hydroplaning velocity per unit increase in groove parameter (km/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal Depth</td>
<td></td>
<td>0-9</td>
</tr>
<tr>
<td>Longitudinal Width</td>
<td></td>
<td>0-16</td>
</tr>
<tr>
<td>Longitudinal Spacing</td>
<td></td>
<td>0-5.25</td>
</tr>
<tr>
<td>Transverse Depth</td>
<td></td>
<td>0-10</td>
</tr>
<tr>
<td>Transverse Width</td>
<td></td>
<td>15-35</td>
</tr>
<tr>
<td>Transverse Spacing</td>
<td></td>
<td>0-12</td>
</tr>
</tbody>
</table>

In general, the studies show that pavement grooving has more contribution than the tire tread groove dimension on the hydroplaning velocity. A comparison between grooved tire and grooved pavement is presented in Table 2-3.
Table 2-3. A comparison between grooved tire and grooved pavement

<table>
<thead>
<tr>
<th>Hydroplaning Speed Increase per unit increase in Pavement Groove Depth</th>
<th>Hydroplaning Speed Increase per unit Increase in Tire Groove Depth</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulation performed</td>
<td>Km/h per mm depth</td>
</tr>
<tr>
<td>Longitudinally grooved pavement &amp; smooth tire</td>
<td>1.93</td>
</tr>
<tr>
<td>Longitudinally grooved pavement &amp; longitudinally grooved tire</td>
<td>1.26</td>
</tr>
<tr>
<td>Longitudinally grooved pavement &amp; transversely grooved tire</td>
<td>2.41</td>
</tr>
<tr>
<td>Transversely grooved pavement &amp; smooth tire</td>
<td>12.86</td>
</tr>
<tr>
<td>Transversely grooved pavement &amp; longitudinally grooved tire</td>
<td>12.80</td>
</tr>
<tr>
<td>Transversely grooved pavement &amp; transversely grooved tire</td>
<td>10.91</td>
</tr>
</tbody>
</table>
2.2 Three Zones in Tire Hydroplaning Phenomenon

In different studies it is shown than when the tire goes through the pavement with existence of water film thickness three zone shape beneath the tire.

In region I, the water film collides with the tire leading edge at given vehicle speed so that the kinetic energy of the water on the pavement changes to the hydrodynamic pressure, leading to the tire deformation and floating on the pavement once the hydrodynamic force would be more than the normal load from tire to the pavement. In region II a transition happens from full contact of tire and pavement to the floating. Accordingly, the tire starts to slip on the water film as a result of hydrodynamic pressure from the water on the tire to lift it up. It is noteworthy to mention that in the region I the contribution of the hydrodynamic pressure is more influential, however, in region II the viscous effect of the water film is more dominant. Region III is the region where the tire totally adheres to the pavement. As the vehicle speed reaches to the hydroplaning speed, the leading edge of the tire totally lifts the whole tire and as a result region III reduces and region I becomes completely dominant. The summery of the aforementioned regions are depicted in Figure 2-1.

![Figure 2-1. Three zones in tire hydroplaning phenomenon (Regenerated from [45])]
2.3 Empirical Model for Hydroplaning Prediction

A large number of studies from 1960 have been conducted to considered different parameters toward prediction of the hydroplaning velocity. A summary of the equation in this regard is presented in bellow.

Table 2-4. Summary of the empirical model to predict hydroplaning velocity [46]

<table>
<thead>
<tr>
<th>Equation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_h = 6.35 \sqrt{p}$</td>
<td>tire hydroplaning speed (km/h) with regard to inflation pressure in kPa [47].</td>
</tr>
<tr>
<td>$V_p = 10.35 \sqrt{p}$</td>
<td>hydroplaning speed in mph with regard to inflation pressure in psi. [47].</td>
</tr>
<tr>
<td>$v_p = 51.80 - 17.15(FAR) + 0.72p$</td>
<td>The modified NASA hydroplaning equation was originally developed by Horne [48] was modified to its most recent form [49]</td>
</tr>
<tr>
<td>$V_p = 7.95 \sqrt{p(FAR)^{-1}}$</td>
<td>Critical Hydroplaning Speed based on Load and Contact Patch [49]</td>
</tr>
<tr>
<td>$V = 508 \sqrt{\frac{Q}{B \ast t \ast c_i}}$</td>
<td>Critical Hydroplaning Speed based on Load, Contact Patch, and Water Thickness [50]</td>
</tr>
<tr>
<td>$Vp = (SD)^{0.04}(p)^{0.3}(TD + 1)^{0.06}A$</td>
<td>Critical Hydroplaning Speed based on Inflation Pressure, Pavement Texture, and Water Thickness [51]</td>
</tr>
<tr>
<td>$A = \max \left{ \begin{array}{l} 10.409 \left[ \frac{V_D^{0.06}}{10} + 3.507 \right] \frac{28.952}{WD^{0.06}} \ - 7.817 \left[ TXD^{0.14} \right] \end{array} \right}$</td>
<td></td>
</tr>
</tbody>
</table>

18
\[
v_p = (p_t^{0.5})(WL)^{0.2}\left[\frac{0.82}{t_w^{0.06}} + 0.49\right]
\]
Modified to Horne’s equations based on Fwa’s work [52]

\[
v_p = a(p_t)^{0.21}\left(\frac{1.4}{\text{FAR}}\right)^{0.5}\left(\frac{0.268}{t_w^{0.651}} + 1\right)
\]
Modified to Horne’s equations based on Fwa’s work for truck tire [52]

\[
v_p = 5.43\sqrt{p_t}
\]
hydroplaning speed for a completely locked wheel [53]

\[
v_p = 7.2\sqrt{P_t}
\]
hydroplaning speed in cases where shallow water prevails [54]

As it is shown in the above table, most of the convectional hydroplaning model based on the speed are empirical. Although these models are simple to use, they are only valid in the range of the input variables that they are parametrized for. If they would be used for the range of the input beyond the values that they are verified for, considerable risk will be associated with the accuracy of these empirical models.

### 2.4 Hysteresis Friction

Rubber friction is a heated debate among the scientist studying the interaction of rubber-pavement. A large number of studies have been conducted to characterize the rubber friction at different condition. One of the most recent and most advanced theories is presented by B. N. J. Persson [55, 56]. He has tried to relate the hysteresis component of the friction to the roughness of the surface. To do so, he first tried to characterize the roughness of a surface using power-spectral density. For
sliding at a constant velocity \( v \), the friction coefficient caused rubber energy dissipation is

\[
\mu \approx \frac{1}{2} \int_{q_0}^{q_1} dq \ q^3 C(q) S(q) P(q) \times \int_0^{2\pi} d\phi \ \cos \phi \ \text{Im} \ \frac{E(q \cos \phi, T_0)}{(1 - v^2)\sigma_0} \quad 2-1
\]

Where \( \sigma_0 \) is the nominal contact pressure on the rubber block sliding on the rough surface. \( C(q) \) is the power spectral density of the surface. \( E(q \cos \phi, T_0) \) is the viscoelastic modulus of the rubber compound which is a function of loading frequency and reference temperature. \( P(q) \) is a function related to real contact area. In fact, when the interface of rubber and hard substrate are magnified at the magnification value of \( \zeta = \frac{q}{q_0} \). Here the \( q_0 \) is the shortest wavevector.

Accordingly, \( P(q) \) can be defined as follows:

\[
P(q) = \frac{2}{\pi} \int_0^{\infty} dx \ \frac{\sin x}{x} \ \exp[-x^2 G(q)] = \text{erf} \ \frac{1}{2\sqrt{G}} \quad 2-2
\]

\[
G(q) = \frac{1}{8} \int_{q_0}^{q_1} dq \ q^3 C(q) \int_0^{2\pi} d\phi \ \left| \frac{E(q \cos \phi, T_0)}{(1 - v^2)\sigma_0} \right|^2 \quad 2-3
\]

\[
S(q) = \gamma + (1 - \gamma)P^2(q) \quad 2-4
\]

In order to involve the effect of the roughness asperity which deforms the rubber, the \( P^2(q) \) is used. However, using a correction factor \( S(q) \) the effect of non-complete contact will be considered. For rubber sliding \( \gamma \approx \frac{1}{2} \) and \( S \rightarrow 1 \) as \( P \rightarrow 1 \) which is complete contact. The aforementioned theory to characterize the friction is also completed by bringing the role of the real
contact area into characterizing adhesive friction and the contribution of hysteresis compared to adhesive is discussed in [57].

In 2004 a paper published by B. N. J. Persson et. al stating that sealing is at the origin of rubber slipping on wet roads [58]. In fact, basic assumption is therefore that when rubber slides on the wet rough surface, the friction force will be determined by the modified power spectrum $C'(q) < C(q)$, explaining friction drops by as much as 20–30% relative to the dry case at low sliding velocities cannot be explained only through the small contact area. This cannot be the result of a water-induced change of adhesion. On the other hand, the friction decrease cannot be blamed on a purely hydrodynamic effect either that leaves the possibility that water might change precisely the bulk, hysteretic friction [58].

It is explained that water pools that form in the wet rough substrate are sealed off by the rubber, as sketched in Figure 2-2, and that will effectively smoothen the substrate surface. Smoothening reduces the viscoelastic deformation from the surface asperities, and thus reduces rubber friction.

![Figure 2-2](image)

Figure 2-2. A rubber block sliding on a rough hard substrate. On a wet substrate the valley turns into a water pool. Sealing of the pool prevents the rubber from entering the valley. By
removing the valley contribution to the frictional force, the sealing effect reduces the overall sliding friction.

Assuming isotropic and translationally invariant statistical properties for the substrate, \( C(q) \) will only depend on the magnitude \( q = |q| \) of the wavevector \( q \). The upper curve in Figure 2-3 shows the power spectrum extracted by Eq. 2-2 from the measured height profile \( h(x) \). The log–log scale shows that for \( q > 1,600 \text{ m}^{-1} \), \( C(q) \) drops as a power law, as expected for a self-affine fractal surface. The fractal dimension of this surface is determined by the slope of the curve in Figure 2-3 (which equals \( 2D_f - 8 \), where \( D_f \) is the fractal dimension) and is about \( D_f = 2.2 \); the r.m.s. roughness can be obtained directly from the height profile, \( h_{(r.m.s.)} \approx 0.3 \text{ mm} \).

![Figure 2-3. Surface roughness power spectra C(q). Top curve, extracted from the measured height profile for a dry asphalt road surface, and bottom curve, calculated assuming sealing of all pools in the same surface when wet, as in Figure 2-2 [58].](image)

At low velocities \( (v < 60 \text{ km/h}) \) there will be negligible hydrodynamic water build-up between the tire and the road surface. In essence, if \( v < (\sigma/\rho)^{1/2} \), where \( \sigma \) is the perpendicular stress in the tire–pavement contact area and \( \rho \) the water mass density, there is sufficient time for the water
to be squeezed out of the contact regions between the tire and the road surface, except for water trapped in road cavities. From the water-smoothened height profile $h'(x)$ a modified power spectrum $C'(q)$ shown by the lower curve in Figure 2-3 is obtained.

2.5 Effect of Lubrication on Adhesion Friction

Beside the hysteresis component, adhesion is the most significant contribution to rubber friction on rough surfaces as long as wear and hydrodynamic effects are negligible during sliding process. Basically, adhesion can even take place during rubber sliding friction on wet rough surfaces. The presence of lubricant is a necessary but not sufficient condition for the suppression of adhesion [59]. In the last two decades, much progress has been obtained in modeling the friction behavior of elastomer materials on rough substrates, which commonly can be described by fractal or self-affine roughness spectra. A large number of studies are conducted regarding the sliding friction on dry and wet condition. Here we have tried to bring the summery of the recent studies related to sliding friction at wet condition.

During the last 20 years a question of high interest is the microscopic mechanism governing the well-known improvement of wet traction of tire treads, when carbon black is replaced by silica. Also, it has been argued that the improvement is due to differences in the morphology of the filler network, as quantified by the lower activation energy in the case of silica. This was related to “dynamically softer” filler-filler bonds of the silica network, which can be assumed to be responsible for a deeper penetration of the rubber into the track roughness.

Le Gal et al. combined experimental and theoretical approach to understand the hysteresis and adhesion contributions to rubber friction on dry and lubricated rough surfaces [59]. By the widely
accepted decomposition of the friction coefficient $\mu$ into hysteresis and adhesion components, the
adhesion friction coefficient is given as follows:

$$\mu_{Adh} = \frac{\tau_s}{\sigma_0} \times \frac{A_c}{A_0} \quad 2-5$$

Following previous investigations carried out on the formation and breakage of contact patches
between rubber-like materials and hard substrates, a semi-empirical formulation for the velocity
dependence of the true interfacial shear strength $\tau_s$ materials and hard substrates, a semi-empirical
formulation for the velocity dependence of the true interfacial shear strength $\tau_s$ was derived:

$$\tau_s = \tau_{s,0} \left(1 + \frac{E_{\infty}/E_0}{\left(1 + (v_c/v)\right)^n}\right) \quad 2-6$$

where $n$ is a material dependent exponent, $\tau_{s,0}$ is the interfacial shear strength in the limit of very
low velocities, $v_c$ is the critical velocity above which the true interfacial shear strength $\tau_s$ reaches
a plateau value, $v$ accounts for the sliding velocity and $E_{\infty}/E_0$ is the step height of the dynamic
modulus between rubbery and glassy state.

Eq. 2-7 relates the pronounced velocity dependence of the adhesion force to interfacial peeling
effects between the rubber and the substrate asperities. The rate dependence of peeling-off
experiments between rubber-like materials and blunt substrates has been
considered via the introduction of an effective surface energy $\gamma_{eff}$ describing the viscoelastic
nature of such processes. The effective surface energy $\gamma_{eff}$ can be assumed to be related to the
true interfacial shear strength $\tau_s$ via a characteristic length scale $l_s$ independent of peeling rate that
can be identified with the size of the peeling process zone ($l_s \approx \Delta \gamma_{eff}/\tau_s \approx \Delta \gamma_0/\tau_{s,0}$). Then
the rate dependence of the effective surface energy $\Delta \gamma_{eff}$ follows the same semi-empirical formulation as following equation:

$$\Delta \gamma_{eff} = \Delta \gamma_0 \left( 1 + \frac{E_{ao}/E_0}{\left(1 + (v_c/v)\right)^n} \right)$$

At low rates of separation, the effective surface energy exhibits a constant value corresponding to the static surface energy $\Delta \gamma_0$ which can be estimated e.g. by contact angle measurements. By increasing the rate of separation, $\Delta \gamma_{eff}$ goes through a modulus-like transition and follows a power law until a critical velocity $v_c$ is reached. The step height of the transition is given by the modulus ratio observed during mechanical spectroscopy, $E_{ao}/E_0$, i.e. the ratio of the dynamic moduli in the glassy and rubbery state.

In another work by the same authors [57], effective work of adhesion $\Delta \gamma_{eff}$ considering the viscoelastic energy dissipation in the crack opening area at the edge of the contact patches, has been included. They have shown if the surface is lubricated with a stabilized water-detergent film, the adhesion effects are totally suppressed, and friction is due to pure hysteresis. Indeed, the observed continuous increase of the experimental results with respect to the sliding velocity in Figure 2-4 can be well described by the simulated hysteresis friction curve. In addition, they have explained that the results concerning the role of the glass transition temperature $T_g$ and the modulus ratio $E_{ao}/E_0$ on the effective work of adhesion $\Delta \gamma_{eff}$ are also relevant for tire traction.
Figure 2-4 Stationary friction curves, (symbols) for the carbon-black (left) and silica (right) filled S-SBR samples on a granite surface at load 12.3 kPa under dry and wet conditions, as indicated [57].

Lorenz et al [55] also conducted extensive experiment and developed a semi-empirical model to estimate the adhesion friction. In their work, the adhesion is defined as the function of length scale and the sliding velocity. They have obtained the shear stress shown as follows:

\[ \tau_s = \tau_{s0} \exp\left(-c \left[ \log_{10} \left( \frac{v}{v_0} \right) \right]^2 \right) \]  

From combining Eq. 2-5 and Eq. 2-8, Eq. 2-9 will be obtained and shown as follows:

\[ \mu_{adh} = \frac{\tau_{s0}}{\sigma_0} \exp\left(-c \left[ \log_{10} \left( \frac{v}{v_0} \right) \right]^2 \right) P(q) \]  

Since in this dissertation, the hysteresis part of the friction is developed from finite element (FE) analysis, in order to compared the result with the experimental data, we will be using Eq. 2-9 to obtain total friction coefficient (summation of adhesion and hysteresis) toward verifying the multi-length scale FE hysteresis friction model.
2.6 Conclusion

In this chapter, with regard to the scope of the work, a literature review is conducted. First, the importance of the hydroplaning is elaborated and the important factors on the hydroplaning phenomenon such as vehicle velocity, water film thickness (WFT), tire construction, and tread pattern are discussed and the summery of the related works in the literature is presented. In addition, the major studies that characterized the hydroplaning from both empirical and computation approaches (FEM, FDM, FVM) are summarized. Furthermore, different friction coefficient mechanism specifically the hysteresis friction and the existing theory to characterize the friction is presented and the lubrication as one of the major factors affecting the friction coefficient is briefly explained. Apart from the papers that have been reviewed in this chapter, a few papers are also reviewed in the following chapters and the limitations associated with those works are presented in the corresponding chapters.
3 Rheological Properties of Rubber Materials for Tire FE Modeling

In this chapter, the experimental data used to characterize the hyper-elastic material model. First, the existing material models representing the hyper-elasticity properties of the rubber are reviewed and the pros and cons associated with each of them are summarized in the remarks. In this work, since the sliding rubber experiences dramatic shear stress from the interaction with rough surface, apart from uniaxial and biaxial, the planer shear test data is involved in the characterizing the material model to obtain more robustness. In addition, the viscoelastic properties of the rubber are characterized to be implemented in the FE code using Prony Series. To do so, a novel nonlinear programming and Genetic Algorithm (GA) are implemented in MTLAB. The results compared with the experimental data to verify the developed methodology.

3.1 Introduction

Viscous properties of the rubber made it highly appropriate in a wide range of applications such as tires, seals, wiper blades, shoes, etc. Rubber enjoys both elastic and viscous properties. Most of the material in the nature possess only the elastic properties which associated with a linear stress-strain curve before reaching to plastic region. There exists a type of elastic materials which the stress-strain curve is not linear. Rubber is also categorized as a material that linear elastic models can not represent its behaviors. To obtain this non-linear behaver, a strain energy density function shown in Eq. 3-1 relates the strain energy density of the material to the gradient of deformation. One of the non-linear materials derived from the strain energy density function is hyper-elastic material that represent the rubber elasticity behavior sufficiently.
$$U = f(I_1, I_2, I_3) = f(\lambda_1, \lambda_2, \lambda_3)$$

The strain energy density can be written as a function of principal stretch ratios \((\lambda_1, \lambda_2, \lambda_3)\) or the stress invariants \((I_1, I_2, I_3)\). There exist several hyper-elastic material models that represent the non-linearity of the rubber behaviors and mainly classified in 3 groups. The first group are the empirical or phenomenological models that mostly rely on experimental data. These models are not derived directly from the theory. The numerical methods are mainly used to parametrize these types of models toward finding the model constants. The hyper-elastic material models that can be classified in the first group are Mooney-Rivlin, Ogden, Polynomial, Yeoh, Marlow and Saint Venant–Kirchhoff.

The second group of the hyper-elastic material models are the mechanistic models that are based on statistical thermodynamics of cross-linked polymer chain network. Neo-Hookean and Arruda–Boyce model are categorized in the 2nd group.

The 3rd group of the material models are the hybrid models that obtain the combination of the 1st and 2nd groups’ characteristics. Gent and Van der Waals models are categorized in the 3rd group.

The summery of the aforementioned materials models are presented in the following table.

<table>
<thead>
<tr>
<th>Category</th>
<th>Model</th>
<th>Strain Energy Function</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empirical</td>
<td>Mooney-Rivlin</td>
<td>( W = C_1(I_1 - 3) + C_2(I_2 - 3) )</td>
<td>Proposed by Melvin Mooney for uncompressible material and expressed in terms of invariants by Ronald Rivlin [60].</td>
</tr>
</tbody>
</table>
Empirical Ogden

\[ W = \sum_{i=1}^{N} \frac{\mu_i}{\alpha_i} (\lambda_1^{\alpha_i} + \lambda_2^{\alpha_i} + \lambda_3^{\alpha_i} - 3) \]

The model was developed by Raymond Ogden and needs 3 deformation modes [61].

Empirical Polynomial

\[ W = \sum_{i,j=0}^{n} C_{ij}(I_1 - 3)^i(I_2 - 3)^j + \sum_{k=1}^{m} D_k(J - 1)^{2k} \]

The more general form of the Mooney Rivlin model with consideration of compressibility and needs data from at least 2 deformations [62].

Empirical Yeoh

\[ W = \sum_{i=1}^{3} C_i(I_1 - 3)^i \]

Introduced by Yeoh and needs data from at least 1 deformation mode [63].

Mechanistic Neo-Hookean

\[ W = C_1(I_1 - 3 - 2\ln J) + D_1(J - 1)^2 \]

Needs data from at least 1 deformation mode.

Mechanistic Arruda–Boyce

\[ W = C_1 \sum_{i=1}^{5} \alpha_i \beta^{2i-2}(I_1^i - 3^i) \]

Based on the statistical mechanics of a material [64].

Hybrid Gent

\[ W = -\frac{\mu J_m}{2} \ln \left(1 - \frac{I_1 - 3}{J_m}\right) \]

Based on the concept of limiting chain extensibility [65].

\[ I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \]

1\textsuperscript{st} strain invariants

\[ I_2 = \lambda_1^2 \lambda_2^2 + \lambda_2^2 \lambda_3^2 + \lambda_3^2 \lambda_1^2 \]

2\textsuperscript{nd} strain invariants

\[ I_3 = \lambda_1^2 \lambda_2^2 \lambda_3^2 \]

3\textsuperscript{rd} strain invariants
In the strain energy density functions presented in Table 3-1, \( I_1, I_2, \) and \( I_3 \) represent the stain invariants. Also \( \lambda_1, \lambda_2, \) and \( \lambda_3 \) are the principal stretch ratios. \( \alpha, \beta, \mu \) are the coefficients obtained from the curve fitting and \( J_m \) represents the elastic volume ratio. \( D \) also represents the compressibility of the material.

## 3.2 Some general remarks from literature

In order to choose the best material model representing the hyper-elasticity of the rubber, the following remarks from the literature are considered:

- Simplicity and the fewer fitting parameters can help in both computational time for numerical analysis and help to avoid the overfitting.
- The stress-strain curve obtained from the predictive model should have a good agreement with the one obtained from the experimental data.
- Depends on the rubber deformation in the real scenario, the material model representing all the deformation modes should be utilized.
- For a unique set of parameters, just one experiment would be sufficient to obtain parameters for Neo-Hookean, at least two experiments for Mooney-Rivlin; three for Signorini/Arruda-Boyce; and three or more for Ogden material models.
- The above-mentioned material models are reliable for specific range of magnitude of deformation. For instance, Neo-Hookean up to 100\%, Mooney-Rivlin up to 150-200\%; Full Ogden 0-Failure and Yeoh or Arruda-Boyce are reliable in a particular range deformation.
- If the fitting is conducted using a particular test type (s), then it would be useful to try other tests to check the stability of the model parameter(s).
3.3 Hyper-elastic Properties Based on the Mooney-Rivlin Model

First, Mooney-Rivlin material model is proposed to predict the hyper-elasticity properties for two rubber samples to see the accuracy of the prediction.

After the calibration is done, the constants of the model are obtained as follows:

Table 3-2. The obtained parameters after the calibration of the Mooney-Rivlin model

<table>
<thead>
<tr>
<th>Sample</th>
<th>C10</th>
<th>C01</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.009</td>
<td>0.064</td>
<td>1.178e-6</td>
</tr>
<tr>
<td>B</td>
<td>0.007</td>
<td>0.116</td>
<td>4.342e-5</td>
</tr>
</tbody>
</table>

Samples of SBR compounds and the testing data are provided by Sumitomo Rubber Industries (SRI) and the model prediction shows a good agreement with experimental data. Although the Moony-Rivlin material model can be calibrated with only two testing modes, it is observed that when the rubber experiences considerable shear stress, Moony-Rivlin model cannot represent the real scenario. In fact, the Moony Rivlin model is reliable up to 150-200% magnitude of deformation. Since the rubber model shows the deformation more than 200% at the leading edge, the full Ogden model can be considered a more reliable candidate for a highly wide range of deformation from 0% to failure. Consequently, in the next step, Ogden model is also calibrated and has been used as the main material model of FE analysis in this work. This material model has been calibrated with both uniaxial and planar test data towards providing more accurate performance.
3.4 Hyper-elastic Properties Based on the Ogden model

Ogden model has been used for predicting the nonlinear stress-strain behavior of materials such as rubber or polymer. It was introduced by Ogden in 1972, and the strain energy density function for an Ogden material is as follows

\[ W = \sum_{i=1}^{N} \frac{\mu_i}{\alpha_i} (\lambda_1^{\alpha_i} + \lambda_2^{\alpha_i} + \lambda_3^{\alpha_i} - 3) \] 3-2

where \( \lambda \ (j=1,2,3) \) is the principal stretch ratio and \( \mu \ (j=1,2,3) \) and \( \alpha \ (j=1,2,3) \) are empirically determined material constants.

3.5 Test Setup to Obtain the Experiment Data for Calibrating Ogden Model

As mentioned before, several tests data need to be used for 3 different deformation mode. Biaxial testing is often conducted on flat, cruciform shaped specimen, where the four arms of the specimen are pulled by the four perpendicular forces. This provides a homogenous distribution of strain in the thickness direction yielding in the center of the specimen.

The main reason for running biaxial tests, apart from the uniaxial test is to determine a specimen’s mechanical properties in its different points and analyze the stress and strain distribution. The raw test data for uniaxial, biaxial and shear tests shown in this part for the SBR compounds, are provided by Sumitomo Rubber Industries (SRI). To obtain the test data, planar biaxial testing machine with several actuators that can also be configured with different strokes and speeds are employed. The matrix of the equibixial tests at different conditions are shown in Table 3-3.
Table 3-3. The matrix of the equibixial tests at different loading rates and directions

<table>
<thead>
<tr>
<th>Test Index</th>
<th>Loading rate at X direction (mm/min)</th>
<th>Loading rate at Y direction (mm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
<td>1.3</td>
</tr>
<tr>
<td>2</td>
<td>5</td>
<td>2.5</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>3.8</td>
</tr>
</tbody>
</table>

Each actuator represents the different axis of the planar biaxial test and moves in equal or opposite directions so that the sample center point remains stationary. First, the biaxial specimen is mounted on the tester and the arms are clamped.

3.6 Ogden Model Calibration

In this part, the results obtained from the code development toward calibrating Ogden model is presented.

The model has several constants that they need to be indicated with regard to the calibration results. First, the uniaxial test data is compared with the result obtained from uniaxial Ogden model. This is the most convenient way the calibrate the model as single load is applied into the rubber sample.
As it is obvious, the prediction results have a very good agreement with the experimental test data shown in Figure 3-1.

Figure 3-2. Parameters obtained for calibration of uniaxial Ogden model

<table>
<thead>
<tr>
<th>alpha1</th>
<th>alpha2</th>
<th>alpha3</th>
<th>mu1</th>
<th>mu2</th>
<th>mu3</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.737141</td>
<td>2.177274</td>
<td>0.000001</td>
<td>0.049811</td>
<td>0.739676</td>
<td>0.000001</td>
</tr>
</tbody>
</table>

3.7 Model Calibration for Uniaxial and Shear Loads for Ogden Model

In many studies, just uniaxial tests are enough for FE modeling, however, since the rubber sample in our case studies experience shear loads, we need to find confidence over the planar test data. Accordingly, planar test data at x and y directions are compared with the planar Ogden models. It is observed that calibrating the Ogden model for the planar test causes deviation of the uniaxial Ogden model from the uniaxial experimental test data shown in the following figures.
Finally, it is concluded that just calibrating a single load, cannot be extended to the other modes of loading. Accordingly, all the uniaxial, planar and equibiaxial Ogden model should be calibrated with the corresponding test data at the same time. It demands a huge number of try and error to find a good agreement for all three modes of the loadings. The following figure shows that model predications have a very good agreement with the experimental data.

Table 3-4. Parameters obtained for calibration of biaxial Ogden model

<table>
<thead>
<tr>
<th></th>
<th>alpha1</th>
<th>alpha2</th>
<th>alpha3</th>
<th>mu1</th>
<th>mu2</th>
<th>mu3</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7.876291</td>
<td>1.481787</td>
<td>0.000001</td>
<td>0.00359</td>
<td>1.527536</td>
<td>0.000001</td>
</tr>
</tbody>
</table>

Figure 3-3. Biaxial results obtained from experiment and Ogden model calibration
Figure 3-4. Results obtained from experiment and Ogden Model Calibration for all the loading modes

Table 3-5. Parameters obtained for Ogden Model Calibration for all the loading modes

<table>
<thead>
<tr>
<th>alpha1</th>
<th>alpha2</th>
<th>alpha3</th>
<th>mu1</th>
<th>mu2</th>
<th>mu3</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.58754</td>
<td>4.021783</td>
<td>-3.37996</td>
<td>2.073074</td>
<td>0.174235</td>
<td>-0.01441</td>
</tr>
</tbody>
</table>

3.8 Viscoelastic Behavior

Apart from the elasticity, rubber shows different behavior at different loading frequencies. As a result, to characterize the time dependent behaviors of the viscous materials such as rubber, a complex module of elasticity is represented including storage modulus $E'$ and loss modulus $E''$ which is shown in the following equation.

$$ E^* = E' + iE'' $$  

3-3

3.9 Optimization Procedure to Obtain Viscoelastic Material Model

In order to obtain the frequency dependent behavior for the rubber viscoelastic model, first the DMA test data is obtained to shape the master curves for compounds A and B. The DMA test data is provided by SRI. The test data is obtained at different temperature between $-20^\circ$C to $80^\circ$C. Also
the test is conducted at the frequency range of 1Hz to 20 Hz. The DMA test data obtained from the conventional DMA test machine that has the limitation of the frequency range. In order to obtain the wider range of the frequencies toward obtaining the master curve, the time-temperature superposition is performed. In order to obtain the material properties at different frequency, WLF equation shown in Eq. 3-4 is used to obtain the temperature shift factor.

\[
\log \alpha_T = -\frac{C_1^T (T - T_0)}{C_2^T + (T - T_0)}
\]

where \( C_1^T \) and \( C_2^T \) are the constants that should be obtained through shifting the data at different temperature to match the data at the reference temperature 20°C. As it is shown in Figure 3-5, the shift factor at the reference temperature 20°C is equal to 1. The related shift factor for two rubber compounds at different temperature are shown in the following figure.

Figure 3-5. Obtained shift factor for two studied rubber compounds
The storage modulus and loss modulus from the experiment data for the two compounds are shown below. These 2 compounds show different dissipative behavior and used throughout this research.

![Storage and Loss Modulus Graphs](image)

Figure 3-6. The storage modulus and loss modulus from the experiment for two compounds

For FE modeling of the rubber sample slides on the different surfaces (rough and flat) we need to consider the viscoelastic material properties. To do so, physical model called Maxwell–Wiechert model shown below is used.

![Maxwell–Wiechert Model Diagram](image)

Figure 3-7. Maxwell–Wiechert model used as a physical model for presenting the viscoelastic properties
In this model, time and temperature are the input parameters which shape the relaxation modulus as a function of time and temperature. The relaxation modulus function can be elaborated in the shape of a series of exponentials called Prony Series as follow:

\[ E(t, T) = E_\infty + \sum_{i=1}^{n} E_i \cdot \exp \left( \frac{-t}{\alpha T \tau_i} \right) \]  

where \( n \) is the number of the terms used in the series. \( E_\infty \) is a constant showing the completely elastic behavior of the rubber and \( E_i \) and \( \tau_i \) are the constants showing the elastic and the relaxation time of the compounds. In this equation, \( \alpha T \) is the shift factor.

In the frequency domain the Prony series can be written in the following form.

\[ G'(\omega) = G_0 \left[ 1 - \sum_{i=1}^{N} g_i \right] + G_0 \sum_{i=1}^{N} \frac{g_i \tau_i^2 \omega^2}{1 + \tau_i^2 \omega^2} \]  

\[ G''(\omega) = G_0 \sum_{i=1}^{N} \frac{g_i \tau_i \omega}{1 + \tau_i^2 \omega^2} \]

The following flowchart shows the procedure employed in the MATLAB software to obtain the coefficient of the Prony series for the best prediction. In this process, MATLAB Genetic Algorithm (GA) tool is used.
Figure 3-8. The flowchart showing the procedure used in MATLAB environment to obtain the model parameters

Figure 3-9 shows the fitted Prony series for compound A. Less than 5% root mean square (RMS) error is obtained showing the Prony series has a good capability to predict the viscoelastic properties of the studied rubber compounds. For both compound A and compound B, the Prony series are obtained and results shown in Table 3-6.
Figure 3-9. The Fitted Prony series obtained to predict the viscoelastic properties of the studied rubber.

Table 3-6. The parameters obtained from GA to fit Prony series into the experimental data.

<table>
<thead>
<tr>
<th>Prony Series Constants</th>
<th>Material A</th>
<th>Material B</th>
</tr>
</thead>
<tbody>
<tr>
<td>g1 (MPa)</td>
<td>2.22E-05</td>
<td>0.000243</td>
</tr>
<tr>
<td>g2 (MPa)</td>
<td>1.79E-08</td>
<td>0.001169</td>
</tr>
<tr>
<td>g3 (MPa)</td>
<td>0.000112</td>
<td>0.006726</td>
</tr>
<tr>
<td>g4 (MPa)</td>
<td>3.03E-05</td>
<td>0.04471</td>
</tr>
<tr>
<td>g5 (MPa)</td>
<td>0.06628</td>
<td>0.2104</td>
</tr>
<tr>
<td>g6 (MPa)</td>
<td>0.9299</td>
<td>0.7326</td>
</tr>
<tr>
<td>tau1 (s)</td>
<td>9608</td>
<td>792</td>
</tr>
<tr>
<td>tau2 (s)</td>
<td>9340</td>
<td>27.7</td>
</tr>
<tr>
<td>tau3 (s)</td>
<td>8240</td>
<td>0.1243</td>
</tr>
<tr>
<td>tau4 (s)</td>
<td>8257</td>
<td>0.002377</td>
</tr>
<tr>
<td>tau5 (s)</td>
<td>0.001971</td>
<td>0.000166</td>
</tr>
<tr>
<td>tau6 (s)</td>
<td>1.91E-05</td>
<td>1.87E-05</td>
</tr>
</tbody>
</table>
It is noteworthy to mention, the GA tools that is employed to find the parameters, works based on the minimizing of the RMS error obtained from difference of the prediction and the test data observed in the experiment. It is tried to narrow down the range of the frequency of the Prony series fitted to the experimental data to improve the accuracy of the model in the range of frequencies applicable for FE analysis.

### 3.10 Conclusion

In this chapter, different material model to characterize the rubber behavior in the FE model is evaluated. Since the rubber sample in the FE model experienced considerable deformation, the conventional material models such Neo-Hookean and Mooney-Rivlin cannot provide sufficient information at the full range of the rubber deformation sliding on the rough surface. As a result, the Ogden model is considered as the best candidate for the material model to be implemented in the FE model. In this chapter, it is shown that model calibration for a single mode of the loading might cause the performance of the material model to predict the rubber behaviors for different modes of loading. As a result, it is shown that, the material model should be calibrated at the same time for all the modes of loading. In order to calibrate the material model, a novel nonlinear programming and Genetic Algorithm (GA) are used. After calibrating the material model, the results obtained from the material model prediction, show a very good agreement with the test data used in this work.
4 FE Multiscale Modeling of Hysteresis Friction

In recent years, thanks to advent of powerful computers, a large number of studies in multi-scale modeling are conducted. The main goal of multi-length scale modeling is first: applying the principals at the lowest scale and transferring the results at the next scale (coarse scale) and second: providing a macroscopic model from the models describing the physics at lower scale. Usually moving from lower scale to upper scale associates with statistical procedures. For instance, here we have used averaging technique to obtain the profile of the rough surface. Furthermore, transferring the results from upper scale to lower scale needs homogenization technique. Since the interaction of the rubber with the rough substrate is influenced by several scales, a multi-length scale frictional constitutive model which is able to solve the contact problem between the rubber and rough substrate in lower scale and then transfer the result to upper scale has been the motivation of this chapter. In this chapter, first a 2D multi-scale FE model is developed and verified with the results available in the literature. Then as a part of the CenTiRe project MODL-2017-B6-11 titled “Comprehensive Multi-Scale Modeling of Tire Abrasion” for the first time a 2D multi-scale FE model on the rough surface is developed and the hysteresis friction coefficient is obtained at micro and macro-scales [66]. To obtain a better understanding over the effect of rough surface asperities on the interface of the sliding rubber with the hard substrate, the model is also extended to a 3D multi-scale FE model. One of the drawbacks associated with the physics-based theory to predict the hysteresis friction is that the surface needs to be assumed self-affine fractal. However, in practice, such an assumption might not be accurate for all the surfaces. However, in the multi-scale FE modeling to obtain the hysteresis friction, since we utilized the surface profile directly
into the model, there is no need to calculate the power spectral density (PSD) of the studied surface. Accordingly, in terms of self-affine fractal property for the studied surface, no limitation exits. More details in this regard are explained in part 4.6.

4.1 Surface Profile Measurement and Decomposition to Generate Macroscopic and Microscopic Profile

The surface profile for sandpaper 120 grit is captured using a profilometer machine and the data is taken from different parts of the sandpaper. The machine that is used to take the raw data is from Nanovea which possesses a non-contact chromatic confocal optical technology to measure the height of the rough surface asperities. In order to measure the height at each point of the surface, a white light from the lens is omitted and the reflection of the light will be captured through the optical pen in the machine. Based on the roughness of the surface, the machine is able to filter the input reflected light frequency to specific range. In order to calibrate the machine, the intensity of the light needs to be adjusted on the specimen. With a single calibration, the machine can scan an area of 2500 mm$^2$ (50 mm $\times$ 50 mm). The lens of the machine moves in x and y direction and measures the z profile. The resolution of the movement in x and y direction is 7 $\mu$m.
Figure 4-1. The surface profile measurement machine used to measure different surfaces

The surface profiles obtained from the scanning are decomposed to 2 different scales (micro and macro). In order to decompose the surface profiles a Butterworth band pass filter is used. The power spectral density (PSD) of the surface includes a range of frequency (wave vector) that two intervals are considered as the decomposition which the wave vector $q$ from $10^3$ to $10^4$ (1/m) is devoted to macro and wave vector $q$ from $10^4$ to $10^5$ (1/m) is devoted to the micro scales. Butterworth filters provide the smooth output that is highly helpful in case of FE analysis. The sharp surface profile usually causes the element distortion and divergence issue in FE modeling. In addition, the Butterworth filters are made of simple transfer function from polynomials that made them easy to use. Figure 4-2 shows the original raw data obtained from the profilometer as well as the decomposed scales.
4.2 Homogenization

In many problems, the normal contact can be modelled using a classical FE boundary condition (BC) such as non-penetration (gap). The law of Coulomb is usually used for tangential contact. Here we need to divide the tangential velocity of the rubber elements into slip/stick parts. It is shown in [67] that slip part is governed by constitutive relation that is more advanced version of Coulomb law and the stick part constitutes a constraint in normal contact. Figure 4-4 shows how the tire tread elements is decomposed to rubber tread block in length scale of [cm] to [mm] and finally they decomposed them to length scale of [mm]) to [µm]. For capturing the effect of adhesion even we need to go to the length-scale nanometer [nm] and molecular dynamic (MD) techniques may be used, however, it is out of the scope for the current study.
\[ t_{N,average} = \frac{1}{S_0} \sum^n_n \int_{S_n^1} (t_{N}^{n} j^n \cdot j^0 + t_{T}^{n} i^n \cdot i^0) \, ds \]  

\[ t_{T,average} = \frac{1}{S_0} \sum^n_n \int_{S_n^1} (t_{N}^{n} j^n \cdot i^0 + t_{T}^{n} i^n \cdot i^0) \, ds \]  

For transferring the load between different scales, we need to calculate the average normal stress within the micro-scale rubber block. To do that, the local stress is divided to local components of the normal and tangential direction. We also have the global normal and global tangential directions shown with \( j^0 \) and \( i^0 \), respectively.

Figure 4-3. Component and tangential and normal stress in local and global systems

The boundary condition that needs to be chosen on the micro-length scale is provided in ABAQUS FE software through choosing the penalty method contact condition between the rubber (slave surface) and hard substrate (master surface) to fulfill the following condition:

\[ g_N \geq 0 \]
Choosing the penalty method enforce the contact constraints to regularize non-penetration condition.

As it shown in the Figure 4-4 in order to capture the effect of roughness at different length scales, they have been separated in different length-scales with the methodology that mentioned before. The decomposed surfaces shown in Figure 4-2 is defined as the analytical rigid surface in ABAQUS. In order to implement the analytical rigid surface, the INP files that are already generated from the ABAQUS CAE are modified and the profiles are added to the codes. The analytical rigid surface is coupled to a rigid body reference node. Since the motion of the surface is coupled by the motion of the reference node, ENCASTRE (fixed all the degrees of freedom) is considered as the boundary condition for the references node. To read the reaction force applied on the analytical surface, the rigid body reference node is used which becomes active only when the analytical surface is in the contact interaction. Accordingly, in order to read the total normal and tangential forces applied to the rubber block, the forces applied to the reference point and associated with the road profile are utilized.
Figure 4-4. An exemplary multiscale modeling setup

The nodes at the top surface of the rubber block are coupled with a referenced point and the degree of freedom (DOF) along the vertical direction are coupled and they move together vertically.

The nominal pressure in contact patch for a passenger car tire is applied at the top surface of the rubber block at macro-scale. Applying the pressure at the top surface of the rubber block at the macro-scale provides a distribution of the contact pressures at the elements in contact with the analytical rigid surface. Since the rubber block at the micro-scale should be representative of the elements close to the interface of the rubber and rigid surface, the average of the pressure distribution within the elements in contact with the rigid surface is used as the input pressure applied on the top surface of the rubber block at micro-scale. In addition, using an equation constraint in the micro-scale model, the displacement of the nodes at the left and right sides of the rubber block are coupled to make sure that the boundary condition at micro-scale and macro-scales are transferred correctly.
Accordingly, the displacement of the left and right nodes at DOF $i$ are equal. The frictionless contact condition is considered between the interface of the rubber block and rigid surface at micro-scale. The material properties for both macro and micro scales are the same and the hyperelastic and viscoelastic properties are obtained using the methodology developed in chapter 3.

4.3 Mesh Convergence Analysis

4.3.1 Macro-scale

The mesh convergence analysis is conducted for both macro and micro scales. Different mesh size has been considered and a 2D rubber block FE model sliding on the smooth surface at macro-scale is studied and the distribution of the stress is compared with the validated model obtained from the literature. The simulation results show that base mesh size of 0.3 mm is sufficient to capture the distribution of the stress within the element close to the interface.

In order to verify the boundary conditions (BCs) applied to the model, the contact pressure obtained from the simulation is compared with the results from the validated model in the literature [68]. The contact pressure along the bottom of the rubber block are compared and shown in Figure 4-5. The simulation results have a good agreement with the results obtained from the literature [68]. In this case study, the normal load of 0.3 MPa and sliding velocity of 10 cm/s is selected as the input to the model. Also, mesh refinement at macro-scale is applied on the leading edge of the rubber sample shown in Figure 4-6. The mesh size in the leading edge is reduced to the half size of the base mesh size at macro-scale. Such a procedure assists the FE code to avoid divergence as the excessive deformation at the leading edge is probable.
With regard to the nominal contact pressure 0.3 MPa, the contact pressure error is defined in Eq. 4-7.

\[
\text{Contact Pressure Error} = \left| \frac{P_r - P_s}{P_{\text{nominal}}} \right| \times 100
\]

where \(P_r\) is the contact pressure presented in [68] and \(P_s\) is the contact pressure obtained from the simulation. The maximum contact pressure error obtained from the comparison of the model and the references is less than 14% and the average of the contact pressure error is less than 5%. With regard to the agreement between the results obtained from the existing model and the validated model in the literature, 0.3 mm base mesh size has been used for the entire work for the FE model at macro-scale level.
4.3.2 Micro-scale

In this section with regard to the defined mesh size for the macro-scale, the goal is finding the optimum mesh size for the micro-scale. In this part, the simulations are conducted using the rough surface profiles obtained from the previews section. The pressure load on the top surface of the rubber block at macro-scale is considered 0.3 MPa and sliding velocity of 10 cm/s is applied to the reference point associated with the nodes at the top surface of the rubber block at macro-scale. The average value for the distribution of the contact pressure of the rubber block at macro-scale and the analytical rigid surface is obtained to be utilized as the input load pressure for the rubber block at micro-scale. A 2 mm by 1 mm rubber block as shown in Figure 4-6 is considered for the micro-scale level simulation.

Distribution of the stress within the rubber block at macro and micro level is shown as follows. The maximum stress occurred at the leading edge that has agreement with the experimental test during the wear test on linear friction tester.
Figure 4-6. Distribution of the stress within the rubber block at macro and micro level

In the next step, a mesh size sensitivity analysis is conducted to study the mesh convergence performance. Different size for the mesh size sensitivity analysis is considered. As mentioned above for this part of sensitivity analysis, 0.3 MPa pressure is applied at the top surface of the micro-scale rubber sample.

Table 4-1. Mesh size sensitivity analysis performed to obtain the optimized mesh size

<table>
<thead>
<tr>
<th>Mesh size (mm)</th>
<th>Mises Stress (MPa)</th>
<th>Mises Relative Error (%)</th>
<th>CPRESS Stress (MPa)</th>
<th>CPRESS Relative Error (%)</th>
<th>RF2(N/mm)</th>
<th>RF2 Relative Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.08</td>
<td>0.488</td>
<td>NA</td>
<td>1.058</td>
<td>NA</td>
<td>0.24</td>
<td>NA</td>
</tr>
<tr>
<td>0.06</td>
<td>0.624</td>
<td>28</td>
<td>1.364</td>
<td>29</td>
<td>0.252</td>
<td>5.1</td>
</tr>
<tr>
<td>0.04</td>
<td>0.724</td>
<td>16</td>
<td>1.596</td>
<td>17</td>
<td>0.268</td>
<td>6.4</td>
</tr>
<tr>
<td>0.02</td>
<td>0.832</td>
<td>15</td>
<td>1.804</td>
<td>13</td>
<td>0.28</td>
<td>4.2</td>
</tr>
<tr>
<td>0.01</td>
<td>0.899</td>
<td>8</td>
<td>1.894</td>
<td>5</td>
<td>0.289</td>
<td>3.3</td>
</tr>
<tr>
<td>0.005</td>
<td>0.917</td>
<td>2</td>
<td>1.951</td>
<td>3</td>
<td>0.292</td>
<td>1.1</td>
</tr>
</tbody>
</table>
The relative error at each mesh size for each variable is obtained with regard to the following formula:

$$\% \text{Relative Error} = \left| \frac{X_i - X_{i-1}}{X_i} \right| \times 100$$

Which $X_i$ is the output variable at the mesh size $i$.

With regard to the obtained results for the CPRESS, RF2 and Mises variables at the elements located at the interface with the rigid surface, the relative errors for each variable is used to obtain the average relative error (~ 2%) of all the studied variables, showing convergence toward acceptable relative error. Accordingly, the 0.01 mm mesh size is considered for the reference mesh size at micro-scale for the rest of studies in this work.

![Figure 4-7. The error bars obtained from the mesh size sensitivity analysis](image)

Figure 4-7. The error bars obtained from the mesh size sensitivity analysis
4.4 Hysteresis Friction Coefficient

In this part, the hysteresis friction coefficient will be calculated from coupling of micro and macro scale models. The friction coefficient is function of the pressure, sliding velocity and the pavement roughness. Since the pavement roughness possesses asperities at different scale, in order to consider the effect of the all asperities on the rubber-surface friction, we need to provide a comprehensive multi-scale including the frequency range of at different scales to obtain the hysteresis friction. Different parts of the rubber block at the bottom are interacting with different asperities, the friction coefficient in each analysis increment is time dependent.

\[ \mu = \mu(p, v, t) \]  

In addition, the friction force can be evaluated via monitoring the tangential and normal forces applied to the rough rigid substrate. It is noteworthy to mention that since the outcome of this work is going to be implemented at wet condition such as hydroplaning problem, being immersed in water causes the temperature increase not to happen. Accordingly, the flash temperature is not comprised in this work. As mentioned before, the force monitoring on the rigid substrate is conducted through monitoring the reference point coupled to the rigid substrate. Furthermore, the friction coefficient out of each simulation will be obtained via time averaging throughout the simulation time.

\[ \mu(p, v) = \frac{1}{T} \int_0^T \mu(p, v, t) \, dt \]  

Figure 4-8 shows the flowchart of the simulation process to obtain the overall friction coefficient from the macro-scale model.
Several simulations need to be executed in the micro-scale to obtain the friction dependency on pressure and sliding velocity. As a result, we need to have a rough estimation over the needed normal load condition at micro-scale mode simulation. Accordingly, a macro-scale simulation with
nominal normal load (300 KPa) is conducted and the contact pressure range is obtained. The obtained contact pressure range assists us to shape the distribution of pressures to run the micro-scale simulation.

Table 4-2. Distribution of the pressure and sliding velocity used as the input parameters for micro-scale simulations

<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Sliding Velocity (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>1E-4</td>
</tr>
<tr>
<td>0.2</td>
<td>1E-3</td>
</tr>
<tr>
<td>0.3</td>
<td>1E-2</td>
</tr>
<tr>
<td>0.4</td>
<td>1E-1</td>
</tr>
<tr>
<td>0.5</td>
<td>1E0</td>
</tr>
</tbody>
</table>

As shown in Table 4-2, for each pressure, 5 different sliding velocities are defined which shape totally 25 simulation conditions for micro-scale model. The simulations are made of 2 steps. At the first step the normal load is applied. At the next step (step 2) the sliding velocity is applied and the pressure from the previous step is propagated to the 2nd step. The simulation time for both steps is considered 0.1 s.
Figure 4-9. The distribution (histogram) of the friction coefficient obtained for the micro-scale model at 0.3 MPa normal pressure and 0.01 m/s sliding velocity

The simulation for each case at micro-scale level is repeated for different part of the rough surface. 200 different rough profiles (lines) are prepared from the different part of the raw data that obtained from the scanning of the sandpaper. The micro-scale simulations for each case on different rough surface are executed and the friction results are obtained. For instance, the distribution (histogram) of the average friction coefficient obtained from the different random samples for the micro-scale model at 0.3 MPa normal pressure and 0.01 m/s sliding velocity is shown in Figure 4-9. As it is evident in Figure 4-9, the friction coefficient obtained from different surface samples are different while comparing the distribution with the probability density function (pdf) for a normal distribution (green curve) shows that they possess an almost normal distribution.
The 0.02 standard deviation of the friction coefficient obtained from 8 rough line samples (shown in Figure 4-10) at micro-scale shows that choosing 8 random rough line to obtain the average of friction coefficient can be reliable for the corresponding simulation cases at the specific normal load and sliding velocity.
Figure 4-11. The box plot of the friction coefficient results obtained for the micro-scale model at 0.3 MPa normal pressure and 0.01 m/s sliding velocity.

Also, it is noteworthy to mention that in GW’s contact mechanics [69], it is assumed that the distance between the summit of the asperities and mean surface height shape a normal distribution. In fact, the summits of the asperities contribute to the formation of the rubber deformation and accordingly causes the dissipative loss. As a result, it is expected when the number of the asperities increases, the friction caused by the dissipative loss shape a Gaussian distribution similar to the GW’s theory.

The shape of the random rough surface and corresponding distribution of the summit distances from the mean surface height [70]

With regard to the obtained results, 0.13 is considered for the friction coefficient of the micro-scale model at 0.3 MPa normal pressure and 0.01 m/s sliding velocity which is shown in Figure 4-11.
In this work, the friction coefficient obtained from the calculation of longitudinal force over the normal force applied to the rubber block. The stick slip condition can be witnessed through looking at the longitudinal force applied from the rubber to substrate and observed from the reference point associated with the rough substrate during the sliding.

The results obtained from the simulation shows that the stick-slip behavior can be captured with the multi-scale methodology presented in this work. The weighted time average of the longitudinal force over the normal force is used to obtain the friction coefficient for each case study.
Figure 4-13. Friction coefficient obtained at different normal load and sliding velocity at micro-scale for 2 different rubber compounds.

In this work, the micro-scale simulations for 2 different compounds are obtained. As it is shown in Figure 4-13, the friction coefficient increases with sliding velocity. In addition, the friction coefficient increases with the normal load on the rubber block. Since, increase of the sliding velocity leads to more deformation of the rubber sliding on the rough surface, more dissipation of heat and corresponding hysteresis friction is expected which has an agreement with the obtained results. In addition, with regard to the viscoelastic properties of the compound A and compound B
studied in this work, compound A shows more dissipative behaviors, leading to more hysteresis friction compared to compound B. Furthermore, it is understood that the sliding velocity is corresponding with the loading frequency on the rubber.

![Image](image.png)

Figure 4-14. A simple rough surface with regular sinusoidal asperity shape

For illustration, we can consider a simple case shown Figure 4-14. In this simple setup, if the rubber sample just moved at X direction and would be fixed in Y direction, we can estimate the rubber deformation with $A \sin \left(\frac{2\pi x}{\lambda}\right)$ which $A$ is the amplitude of the asperities height and $\lambda$ is the wavelength of the roughness. Since the $X$ can be replaced by the $Vt$ in which $V$ is the element velocity at X direction and $t$ is the travel time, we can conclude from this simplified assumption that the increase in the sliding velocity is corresponding to the higher loading frequency on the rubber caused by the rough asperities. Accordingly, the result obtained from Figure 4-13 has agreement with the expectation and show that as the sliding velocity increases the, higher frequency range of dynamic viscoelasticity leads to more dissipation loss as well as hysteresis friction.

In order to obtain the relative area (area ratio) we need to obtain the real contact area over the nominal contact area. There are two ways to obtain the real contact area in FE analysis. First, the
real area can be obtained through the number of the nodes that possess contact forces. The node with contact force over the total number of the nodes at the surface shows the relative area. The other way to obtain the relative area is variable CAREA that shows the contact area. This variable shows the areas of the sides (facets) that obtain contact forces.

Figure 4-15 shows a case study to obtain the area ratio. It is obvious that as the rubber block sliding time on the rough surface increases, the real contact area converges to specific value. It is noteworthy to mention that, to obtain a reliable real contact area from FE analysis, the mesh density should be high enough. This task has been done in the mesh sensitivity analysis in this chapter.
The relative area (area ratio) obtained at different normal load and sliding velocity at micro-scale for the compound A is shown in Figure 4-16. As it is shown in the results, the area ratio decreases with the sliding velocity which has an agreement with the analytical results reported in the literature [56, 71]. Also, it is obvious that increase in the normal load on the rubber block causes more rubber deformation at the interface rubber and rough surface. Accordingly, the more contact area is expected as normal load increases.
In addition to the 2D rubber block sliding model, a 3D model is also developed, and the friction coefficients results are compared with the 2D model.

4.5 Developing a 3D Finite element model for rubber block sliding analysis on the rough surface

In order to develop the 3D fine element model, the rough surface should be incorporated into the multi-scale modeling approach. To do so, using the different approximation the rough surface should be reconstructed. The 3D rubber block with dimensions $20mm \times 10mm \times 3.5mm$ have been used in this work. The methodology that can be used for building a 3D rough surface is similar to 2D, however the pattern of building the 2D rough surface should be repeated in the depth (Z direction), as well. In order to rebuild the rough surface for FE multi-scale modeling, 2 main approaches are suggested by the other researchers[72-74] that are reviewed as follows:

4.5.1 Reconstructing the surface using Power Spectral Density (PSD)

In this method, first the rough surface is scanned and using the Power Spectral Density function (PSD), the roughness of the surface is divided to different scales. For instance, in [72] it is shown that after the PSD is obtained, the surface profile is reconstructed using the Eq. 4-11

$$z(x) = \sum_{i=1}^{N} 2\sqrt{f_0C_{PSD}(q_i)} \sin (q_i x + \phi_i)$$  

4-11

Where $f_0$ is the highest frequency and $\phi_i$ is the random shift phase.
Figure 4-17. Rough surface reconstruction for the FE analysis using surface PSD

Figure 4-17 shows an exemplary surface that can be scanned along the $l_i$ direction. Then the PSD of the surface can be obtained using the $C_{PSD}$ function presented by Persson [56, 75, 76]. The artificial surface can be made through the Eq. 4-11. Re-building the surface with regard to the $C_{PSD}$ has some drawbacks. For instance, the random shift phase causes missing the surface characteristics. Also, defining the cut off frequency dramatically affect the similarity of the re-built and real surfaces and it is difficult to obtain a systematic criterion to choose the cut off frequency to represent the real surface sufficiently.

4.5.2 Setting up the rough surface directly from the surface profile using filtering technique

A new approach to directly rebuild the real surface for the FE analysis is directly using the surface profile [73]. To do that, as it is mentioned in 4.1 the surface is decomposed to difference scales using the filtering technique. For instance, in this work, the surface is decomposed into micro and macro scales. Then the decomposed profile can be used to build the 2D lines and 3D surfaces for FE analyses using analytical rigid line/surface in ABAQUS.
Figure 4-18. Rough surface reconstruction for the FE analysis using filtering technique

One of the advantages of using the surface profile to directly setup the FE analysis is avoiding the transformation process that causes surface characteristic loss. This process has been done without defining any random shift phase that causes the surface characteristics loss.

To develop the 3D model, the 2D rubber block as well as the 2D analytical rigid surface should be extended to the 3D. Developing a 3D rubber block is similar to 2D and after the 3D CAD geometry is provided, we have developed the mesh grid within the rubber block. The mesh element type that is used in this simulation is Plane Strain. It is noteworthy to mention that using a Plan Strain analysis against the Plane Stress analysis helps us to prevent rubber deformation in lateral direction of sliding.

Figure 4-19. The script used in ABAQUS input deck to import 2D and 3D surface

69
In order to develop the 3D rough surface, the output file from the scanner is reformatted to the text file in which X, Y, Z are the column related to the location of the measured points at each coordinate. The text file is imported into the SolidWorks software as the points cloud to develop the 3D rough surface. After the points are imported into SolidWorks, the path that the points needed to be connected are defined. The points are connected to maintain the path that the data is collected from the real surface along line $l_i$ (Figure 4-18). Then, the lofted surface feature is utilized to shape the surface between the point toward developing the whole 3D rough surface. The 3D surface file can be saved in INP format to be imported into ABAQUS software and be used for the rest of the analysis. The location of the 3D rigid elements for the rough surface to be modified within the ABAQUS input deck is shown in Figure 4-19. The rough surface is defined as the rigid shell element in ABAQUS.

4.5.3 Comparison between 2D and 3D FE Model for Rubber Block Sliding

The multi-scale 2D FE model has been already discussed and verified extensively in the literature [72, 73]. In this work the 2D and 3D results will be compared with experimental data. In addition, a comparison is conducted between a reference 2D FE model (using 8 line profiles) and the 3D FE model to verify the surface characteristics and the accuracy of the 3D model.

To compare the 2D and a 3D setup for the macro-scale rubber block sliding, the same rough surface profiles (2D rough line profiles) that has been used for the 2D FE model, has been also utilized for building the 3D FE model. The simulation is conducted at 0.3MPa normal load and 0.1 m/s.
In order to obtain the friction coefficient from the 2D model, 8 rough line profile samples \( l_i \) have been used. All the rough profile line that are used to run different 2D models, have been also used with 0.5 mm distance (\( \Delta Z = 0.5\,mm \)) at Z direction to build the 3D rough surface (see Figure 4-18). An exemplary 2D and 3D setups showing the displacement results at Y direction are shown in Figure 4-20. The friction coefficient obtained from the 2D rubber block sliding on each rough line profile is shown in Figure 4-21.
Figure 4-21. The hysteresis friction coefficient obtained from the 2D rubber block sliding on different rough line profile at 0.3 MPa and 0.1 m/s

To compare the performance of 2D model vs. 3D model, the relative error difference between models is used. Accordingly, the average friction coefficients are obtained by sliding the rubber on different rough line profiles using 2D model. The simple arithmetic average method over the friction coefficients may not be a valid method to estimate the overall friction representative the surface characteristics. Therefore, we use random resampling over various combination of measured friction coefficients reported in Figure 4-21. We use different sample sizes (e.g., n= 2, 4, 7) in random sampling and calculate the average friction in each selected subsample. For example, n=2 refers to selecting the friction coefficients from two different types of the rough line profiles. Needless to say, since there are 8 friction coefficients, the number of random samples of size n over these 8 coefficients is the combination of n out of 8 (e.g., \( \binom{8}{n} \)).

The calculated average frictions on selected resampling results are depicted in Figure 4-22.
Figure 4-22. Average friction coefficient obtained from different sampling combination for 0.3 MPa normal load and 0.1 m/s sliding velocity

For instance, in Figure 4-22, the Sample Index (7) represents the average friction coefficient obtained over the friction coefficients of rough line profile indices 2 and 3 (Figure 4-21).

Given the reliability of obtaining the average friction coefficient from 8 rough surface profile in 2D model (discussed in section 4.4), the average friction coefficient obtained from the 8 rough line profiles is considered as the reference value and the relative friction error for all the other 2D model cases as well as the 3D model is calculated using Eq. 4-12:

$$\text{Relative Friction Error (\%)} = \left| \frac{\mu_{\text{average, ref}} - \mu_{\text{average}}}{\mu_{\text{average, ref}}} \right| \times 100$$  \hspace{1cm} 4-12
Figure 4-23. The Relative friction coefficient error obtained from different sampling combination for 0.3 MPa normal load and 0.1 m/s sliding velocity

It is evident from Figure 4-23 that the averaged friction converges as the sampling size increases. In other word, the averaged friction error decreases with the sampling size which is in accordance with the statistical study presented in section 4.4

As it is shown in Figure 4-23, the relative error obtained from the 3D model has the lowest error compared to the average friction coefficient obtained from the different sampling combination of
2D setup, showing the fact that the 3D rough surface can represent the surface characteristics sufficiently. Also, the relative friction errors obtained from different subsample show that increase the subsample size is corresponding with the accuracy improvement. The relative friction error of 3.6% is obtained for the 3D model. As a matter of fact, the more rough surfaces are used to calculate the averaged friction coefficient for 2D model, the more 2D and 3D results converge to similar solution.

In order to investigate the effect of the pressure on the relative friction error, similar analysis is conducted for the macro-scale model of the rubber block sliding at 0.1 m/s and 30 KPa normal load.

![Bar chart showing averaged relative friction coefficient error for different sampling combinations and normal loads.](image)

Figure 4-24. The Averaged relative friction coefficient error obtained from different sampling combination and different normal load.
It is concluded that the relative friction error at 30 KPa normal load deceases with the sampling size and shows the same behavior as 300KPa. However, the averaged friction coefficient error at different sampling size at 30KPa normal load is more than those of 300KPa normal load. To explain this observation, the rubber block contact area with the rough surface for both cases are shown in Figure 4-25.

![Figure 4-25. Rubber block contact area with the rough surface at different normal load (300 KPa and 30 KPa)](image)

It is obvious that at lower pressure (30 KPa normal load), there exists some gap regions within the contact area and the full contact has not shaped. As a result, some line profiles are not in contact in the 3D model while such a gap is not considered in the 2D setup. In fact, in the 2D model, the line contact will be shaped without consideration of the neighbor regions along the Z direction that cause the error if the contact pressure wouldn’t be enough to shape a full contact throughout the contact area.

At the micro-scale simulation, since the range of the contact pressure is more than the nominal contact pressure at macro-scale, enforcing the full contact at the line-profiles in the 2D setup does not originate any issue. However, for the macro-scale model, relying only on the 2D model specifically at lower nominal contact pressure (less than 30 KPa), associates with some risks and a preliminary analysis with the 3D setup is highly recommended to avoid false incorporating of all line profile in the 2D model.
Although the 2D FE model setup is highly computationally efficient, given the aforementioned issues, we have been motivated to extend the 2D model to a 3D FE model for the rubber block sliding on the rough surface at macro-scale. Accordingly, the friction coefficients at the micro-scale (function of sliding velocity and contact pressure) obtained from the 2D micro-scale setup have been used in both 2D and 3D macro-scale models.

It should be emphasized that implementation of the periodic boundary condition at 3D model setup for the micro-scale modeling is highly challenging.

4.5.4 Distribution of stress within the rubber block at different normal load and sliding velocity

The distribution of the Mises stress is shown in Figure 4-26. The rubber block is sliding at X direction and the normal load is applied to the top surface. The simulation is tried to be similar to the experimental condition for the test setup developed at CenTiRe lab. The nodes at the upper surface of the rubber block coupled and move together at Y direction.

It is shown that the max of the Mises stress increases with the sliding velocity. Also, it is evident that the pick of the Mises stress occurs at the leading edge causing more wear at the leading edge which has a very good agreement with the wear experiments conducted in the CenTiRe lab. The elements at the leading edge of the rubber block are the first series of the elements that contact the rough asperities and experience the highest deformation.
Figure 4-26. Distribution of stress within the rubber block at different normal load and sliding velocity on the rough surface: a) 0.3 MPa normal load and 0.1 m/s sliding velocity, b) 0.3 MPa normal load and 0.2 m/s sliding velocity, c) 0.3 MPa normal load and 0.3 m/s sliding velocity

The strain simulation is also conducted, and the readers are referred to APENDIX to see the results.

4.5.5 Distribution of contact pressure (CPRESS) and principal strain within the bottom of the rubber block

Figure 4-27 shows the distribution of contact pressure (CPRESS) and principal strain within the bottom of the rubber block in contact with the rough surface. One of the advantages of developing a 3D model compared to a 2D model is evident here that we can clearly illustrate the contact pressure at the interface of the rubber block and the rough substrate.
Figure 4-27. Distribution of contact pressure (CPRESS) and principal strain within the bottom of the rubber block in contact with the rough surface: a) CPRESS at 0.3 MPa normal load and 0.1 m/s sliding velocity, b) CPRESS 0.3 MPa normal load and 0.2 m/s sliding velocity, c) CPRESS at 0.3 MPa normal load and 0.3 m/s sliding velocity, d) max principal strain at 0.3 MPa
normal load and 0.1 m/s sliding velocity, e) max principal strain at 0.3 MPa normal load and 0.2 m/s sliding velocity, f) max principal strain at 0.3 MPa normal load and 0.3 m/s sliding velocity

As it is shown in Figure 4-27, the contact pressure at the interface has stronger correlation with the normal load than the sliding velocity. However, the sliding velocity has significant effect on the contact pressure shaped at the elements close to the leading edge. Also, it is evident that the distribution of the stress within the bottom surface of the rubber block is highly related to the shape of the rough surface developed in the FE analysis which is tried to be as close as possible to the 3D scanned surface. Furthermore, the max of the principal strain at the bottom surface of the rubber block is decreased from the leading edge toward the tailing edge. The reason that can be explained for this observation is that the elements close to the leading-edge experience both the highest principal strain L11 and shear strain L12 which are in direction of the sliding velocity. Gradient of the principal strain L11 and shear strain L12 at the interface of rubber and rough substrates is highly depended on the sliding velocity which are shown in Figure 7-2 (APENDIX). The results obtained from the simulation highly agreed with the experimental data obtained from the in-house wear tests.
Figure 4-28. A rubber block sample used for the wear test on the rough surface at 0.1 m/s sliding velocity and 0.3 MPa normal load

From the scratches observed at the bottom of the rubber block, shown in Figure 4-28, it is clear that the leading edge of the rubber sample experiences more wear rate than the other part of the rubber block caused by gradient of the contact pressure at the bottom of the sliding rubber block which is witnessed in the simulation results, as well.

4.6 Comparison of The Finite Element Results with the Results Obtained from Analytical Model Previously Developed at CenTiRe

The hysteresis friction obtained from the macro-scale FE model is compared with the results obtained from the analytical model that previously developed and implemented in Center for Tire Research (CenTiRe) [77, 78]. Since this work is mainly focused on the FE analysis, the goal of showing the analytical model results is to provide the reader with some trends that can be obtained from the analytical point of view. The summery of the equations used to develop the analytical
model is presented in the following table. As it is obvious, in order to obtain the friction coefficient through analytical physics-based theory, the power spectral density of the surface under study should be obtained. To do so, it requires some assumptions such as self-affine fractal property that is if the surface goes through the scale transformation i.e. magnification, the surface seems similar to the original scale. This property can help to relate the power spectrum of the surface to the power law characteristics which is shown in Eq. 4-13.

\[ C(q) \sim q^{-2H+1} \]  

4-13

In this equation, H is the Hurst exponent which relates surface fractal property to the power law. This relation is valid on condition that the surface would be self-affine and within specific range of the wavevector \( q_0 < q < q_1 \). As a result, consideration of stone particles distribution within the asphalt texture is important. For more details in this regard the readers refereed to [77].

Table 4-3. List of the equation used toward developing the analytical model to calculate the friction coefficient based on Persson’s theory [78]

<table>
<thead>
<tr>
<th>Equation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ \Delta E_{diss} = \int_0^T \int_0^V \int_0^T d^3 x dt \dot{\varepsilon} ]</td>
<td>Energy dissipation</td>
</tr>
<tr>
<td>[ \Delta E = \sigma_f A_0 vt ]</td>
<td></td>
</tr>
<tr>
<td>[ C(q) = \frac{1}{(2\pi)^2} \int d^2 x (h(x)h(0)) e^{-iq \cdot x} ]</td>
<td>Surface roughness power spectrum</td>
</tr>
<tr>
<td>[ \frac{\partial p}{\partial \zeta} = f(\zeta) \frac{\partial^2 p}{\partial \sigma^2} ]</td>
<td>Area ratio as a function of different length scales</td>
</tr>
<tr>
<td>[ P(q) = \left( 1 + [\pi G(q)]^2 \right)^{-1/3} ]</td>
<td>Area ratio</td>
</tr>
</tbody>
</table>
\[ G(q) = \frac{1}{8} \int_{q_L}^{q} dq q^3 C(q) \int_{0}^{2\pi} d\phi \left| \frac{E(q v \cos \phi)}{(1 - v^2) \sigma_0} \right|^2 \]

\[ \frac{\partial T}{\partial t} - D \nabla^2 T = \frac{Q(z,t)}{\rho C_v} \]

Flash temperature caused by the generated energy

\[ T_q = T_0 + \int_{0}^{\infty} dq' g(q,q') f(q') \]

Temperature as a function of length scale

\[ f(q) = \frac{v q^4}{\rho C_v} \frac{P(q)}{P(\text{min})} \int d\phi \cos \phi \text{Im} \left( \frac{E(q v \cos \phi, T_q)}{1 - v^2} \right) \]

\[ g(q, q') = \frac{1}{\pi} \int_{0}^{\infty} dk \frac{1}{D k^2} \left(1 - e^{-D k^2 t_0}\right) \frac{4q'}{k^2 + 4q'^2} \frac{4q^2}{k^2 + 4q^2} \]

\[ D = \lambda_k / \rho C_v \]

Heat diffusivity

\[ \log_{10} a_T = - \frac{C_1(T - T_0)}{C_2 + T - T_0} \]

Shift factor

\[ E(\omega, T) = E(a_T \omega, T_0) \]

Module of elasticity

\[ \mu_{\text{hys}}(v_s, q, T_q) = \frac{1}{2} \int_{q_0}^{q_1} dq q^3 C(q) P(q) \int_{0}^{2\pi} d\phi \cos \phi \text{Im} \left( \frac{E(q v_s \cos \phi, T_q)}{(1 - v^2) \sigma_0} \right) \]

Hysteresis friction coefficient for a self-affine surface

\[ P(q, T_q) \approx \left( 1 + \left[ \frac{\pi}{8} \int_{q_0}^{q} dq q^3 C(q) \int_{0}^{2\pi} d\phi \left| \frac{E(q v \cos \phi, T_q)}{(1 - v^2) \sigma_0} \right|^2 \right]^{3/2} \right)^{-1/3} \]

Temperature dependent area ratio

As it is shown in Figure 4-29, the friction coefficients results obtained from the FE has a very good agreement with the analytical model. The FE simulation results out the macro-scale models is obtained at 0.3 MPa normal load at different sliding velocity. The effect of increase in the temperature has not been considered in the FE model which caused some deviation from the analytical model results.
Figure 4-29. FE and analytical Friction coefficient obtained at different normal load at sliding velocity (the analytical results obtained from the model developed in [71])

As it is shown in Figure 4-29, up to 0.3 m/s sliding velocity, the higher normal load is corresponding with the higher hysteresis friction. However, at the sliding velocity more than 0.3 m/s, the higher normal load (pressure) is corresponding with less hysteresis friction. The area ratio obtained from the FE at macro-scale shown in Figure 4-30. It is evident that the relative area (area ratio) decrease as the sliding velocity increases. In addition, as the normal load increases, the
relative area (area ratio) increases which is caused by the more deformation of the rubber elements that are in contact with the asperities of the rough surface.

Figure 4-30. Area ratio (relative area) obtained at different normal load at sliding velocity at macro-scale FE model
4.7 Comparison with the results obtained from the experimental data

Using the linear friction setup developed at CenTiRe lab, the rubber compound A is tested to obtain the friction coefficient at different normal load and sliding velocity. A 120-grit rough surface is used to provide the desired roughness in the experimental setup. The rubber sample has the rectangular cross section of 2.5 cm × 2.5 cm.

The experimental data obtained at dry condition used in this work. Accordingly, the friction coefficients obtained from the experiment include both the hysteresis and the adhesion friction.

To compare the friction results obtained from the FE rubber sliding model, a semi-empirical model is used [55]. In this model, the area ratio obtained from the simulation results used in the adhesion model shown in Table 4-4.

The empirical parameters of the model are obtained through minimizing the adhesion friction relative error. To estimate the adhesion friction from the experimental data $\mu_{adh,test}$, the difference between the measured friction from the experiment and the hysteresis friction from the FE model is considered and used in Eq. 4-14.

$$\% \text{ Adhesion Friction Relative Error} = \left| \frac{(\mu_{adh,test} - \mu_{adh,model})}{\mu_{adh,test}} \right| \times 100 \quad 4-14$$

$$\mu_{adh,test} = \mu_{test} - \mu_{hysteresis} \quad 4-15$$

Accordingly, the results obtained from the model (presented in Table 4-4) is added to the results obtained from FE analysis toward obtaining the total friction coefficient.
Table 4-4. Semi-empirical adhesion model and the corresponding contents used [55]

<table>
<thead>
<tr>
<th>Equation / Constant</th>
<th>Description / Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu_{adh, model} = \frac{\tau_f}{\sigma_0} \left( \frac{A}{A_0} \right)$</td>
<td>Semi-empirical adhesion model</td>
</tr>
<tr>
<td>$\tau_f = \tau_{f0} \exp \left( -c \left[ \log_{10} \left( \frac{v}{v_0} \right) \right]^2 \right)$</td>
<td>Shear stress from the adhesion</td>
</tr>
<tr>
<td>$v_0 (m/s)$</td>
<td>7.18</td>
</tr>
<tr>
<td>$c$</td>
<td>0.1</td>
</tr>
<tr>
<td>$\tau_{f0} (MPa)$</td>
<td>5.98</td>
</tr>
</tbody>
</table>

As it is shown in Figure 4-31, the friction coefficients obtained from the FEM for both 2D and 3D increases with the sliding velocity. The summation of the hysteresis friction from FEM and adhesion friction from the semi-empirical model are compared with the experimental data at 0.3 MPa normal load. The results show that the friction coefficients obtained from the computation approach have very good agreement with those obtained from the experiment.
Figure 4-31. Friction coefficient obtained from the computational model and experimental data at different normal load at sliding velocity

In the previous studies seen in the literature, it is noted that in order to obtain the hysteresis friction directly from the experimental data, the substrate is lubricated to deactivate the adhesion friction. However, it is believed that at low sliding velocity, providing a wet surface with spray cannot cancel out the effect of adhesion effectively and still there exist viscous friction caused by the fluid viscosity that needs to be considered and is beyond the scope of this study. Accordingly, the semi-empirical adhesion model is used to obtain the total computational friction toward comparing with the experimental data.
4.8 Conclusion

In this chapter, a FE-based methodology is presented to consider the effect of the pavement roughness on the friction coefficient. Surface profile measurement and decomposition is conducted to generate macroscopic and microscopic profile of the studied surface. First a 2D multi-scale FE model is developed and verified with the data exist in the literature. Then the model is extended to a 3D multi-scale FE model to predict the hysteresis friction and provide more comprehensive information from the interface of the rubber and rough substrate. The simulation results compared with the experimental data obtained from the tests conducted in the lab as a part of the CenTiRe project MODL-2017-B6-11. The presented methodology is able to consider the effect the roughness at different scales without the limitation associated with the physics-based model such as self-affine fractal assumption. In the results obtained from the 3D FE model, it is obvious that the contact pressure at the bottom of the leading edge of the sliding rubber on the rough surface increases with the sliding velocity which explains the considerable wear rate observed at the leading edge of the rubber block in the lab tests. The hysteresis friction FE model presented in this chapter can be employed in the hydroplaning analysis to provide more accurate input for the friction coefficient at different slip and contact pressure within the contact patch with regard to the pavement roughness.
5 Hydroplaning Model Development

5.1 Introduction

In this chapter, the process of building a tire hydroplaning model is comprehensively explained. The literature review regarding the existing models to predict hydroplaning are extensively discussed in chapter 2 (Literature Review and Background). The evaluation of the existing model summarized in the literature review shows that the available models and approaches had mostly two main limitations and areas for improvements:

1. These studies only solved the single-phase fluid as an Eulerian fluid element assuming inviscid, laminar flow. The assumption of inviscid flow ignores the effect of shear stress and the laminar assumption simplifies hydroplaning

2. The conventional FE modeling of a full-tire single-scale model cannot provide us with the sufficient information to study the effect of the pavement roughness

This chapter focused on enhancing these two limitations. It is noteworthy to mention that for including the effect of the pavement roughness on the hydroplaning analysis, some efforts are made by the other researchers and shaped two main frameworks as follows:

- Hydroplaning Analysis Based on Power Spectrum of Asphalt Pavement and Kinetic Friction Coefficient:

In the first approach presented in [79] the 3D optical scanner used to obtain the different asphalts’ topographies. Then with assumption of self-affine fractal features of the pavement surfaces, the power spectral density (PSD) of the surface roughness is calculated. Using the Persson’s friction
theory and the PSD of the surfaces, the effect of the pavement roughness on the friction coefficient is included in the hydroplaning FE model. The first approach summarized in Figure 5-1.

Figure 5-1. Including kinetic friction coefficient in hydroplaning analysis using Persson’s theory [79]

As it is previously discussed in detail, using the Persson’s theory requires using the fractal features of the surface. The fractal assumption for the surface is valid on condition that the wavelength of asperities would be in the specific rang (between shortest distance cutoff and upper distance cutoff corresponding to surface asperities sizes). This assumption might not be valid in practice for all the surfaces which bring some drawbacks using this methodology to characterize the kinetic friction coefficient of the rubber-pavement interaction.

- Hydroplaning Analysis Based on 3D asphalt surface FE mesh generated from the X-ray CT scan images of the pavement specimens

In the 2nd approach [80, 81], the CT scan images of the pavement specimens are provided using an X-ray computed tomographer (CT). Different components of the pavement such as aggregate, binder and air voids are distinguished, and the FE mesh of the pavement specimens is generated.
Then a full tire rolling analysis over the pavement mesh is performed. Using the total energy dissipation caused by the dissipative behavior of the rubber elements, the hysteresis friction coefficient is calculated and used for the hydroplaning analysis which is shown Figure 5-2.

![Figure 5-2](image)

Figure 5-2. Including kinetic friction coefficient in hydroplaning analysis using total energy dissipation [80, 81]

One of the main limitations that 2nd approach has placed on the prediction of the hysteresis friction coefficient is that only the macro-scale asperities of the rough surface are considered in the friction coefficient prediction. Since it is believed that the effect of the surface roughness on the friction coefficient can be characterized more accurately through a multi-scale modeling approach, the methodology presented in chapter 4 can be utilized in the hydroplaning analysis to provide much more robust model with regard to the surface roughness.

### 5.2 Tire modeling

There are three methods that can be used to model a rolling tire: implicit dynamic, arbitrary Lagrangian-Eulerian and dynamic explicit models [82]. For tire modeling at dry and wet condition, different physics based and numerical models are introduced to predict the tire force [83, 84]. As
shown in Figure 5-3, first the 2D cross section of the tire including the reinforcements are modeled. Next, the single pitch tread is added, and the rim is mounted. In the next step, using the Symmetric Model Generation (SMG) capability in ABAQUS, the 3D full tire model is generated in which a tread pitch is periodically repeated to create 360 degrees of a full tire model. In this step, all of the loads, including inflating the tire and the normal load, are applied. Finally, the contact between the tire tread and pavement is defined, and using a steady-state transport analysis, the overall rolling analysis is completed. The results from steady state rolling step are exported into a transient rolling step to evaluate the effect of the force due to WFT on the tire through co-simulation with CFD code. Eq. 5-1 shows the tire deformation for displacement field $u$.

$$\nabla \sigma(u) + b = \rho \ddot{u}$$  \hspace{1cm} 5-1

In Eq. 5-1, $\rho$ is the mass density, $b$ is the body force and $\sigma$ is the Cauchy stress.

### 5.3 Tire materials modeling

The composite materials used to create a tire structure are very complex making it difficult to predict the material behavior. The tire components that are made of rubber include the tread, sidewall, apex, and inner-liner. Therefore, large elastic deformation theory is implemented into the model [85] to ensure good predictions. With regard to the importance of the materials modeling, the neo-Hookean hyper-elastic model is utilized for its ability to reasonably represent the hyper-elastic characteristics of rubber. A list of the equations used for the hyper-elastic materials model and the corresponding properties are presented in Table 5-1 and Table 5-2.

In this work, first a 180/65 R 15 steel belted radial tire is used for hydroplaning analysis. For the tire components, different element types are used to reflect the real characteristics of the tire components. The belt and ply in 2D use the ABAQUS embedded element SFMGAX1, and in 3D
the ABAQUS element SFM3D4R is used. For all other tire components such as the carcass, sidewall and tread, in 2D the ABAQUS element CGAX4R, and in 3D the ABAQUS element C3D8R are employed. In addition, for both road surface and rim, an analytical surface is utilized.

Table 5-1. The material model used for hyper-elasticity

<table>
<thead>
<tr>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$W = f(I_1, I_2, I_3)$</td>
</tr>
<tr>
<td>$I_1 = \lambda_1^3 + \lambda_2^3 + \lambda_3^3$</td>
</tr>
<tr>
<td>$I_2 = \lambda_1^3 \lambda_2^3 + \lambda_2^3 \lambda_3^3 + \lambda_3^3 \lambda_1^3$</td>
</tr>
<tr>
<td>$I_3 = \lambda_1^2 \lambda_2^2 \lambda_3^2$</td>
</tr>
<tr>
<td>$W = c_{10}(\bar{I}<em>1 - 3) + \frac{1}{D_1}(\bar{J}</em>{e1} - 3)^2$</td>
</tr>
</tbody>
</table>

Table 5-2. The parameters used for each tire's components

<table>
<thead>
<tr>
<th>Element</th>
<th>C10 (MPa)</th>
<th>D1 (2/MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carcass</td>
<td>0.446</td>
<td>0.045</td>
</tr>
<tr>
<td>Sidewall</td>
<td>0.318</td>
<td>0.063</td>
</tr>
<tr>
<td>Rim Strip</td>
<td>0.930</td>
<td>0.021</td>
</tr>
<tr>
<td>Belt</td>
<td>1.022</td>
<td>0.019</td>
</tr>
<tr>
<td>Tread</td>
<td>0.685</td>
<td>0.029</td>
</tr>
<tr>
<td>Apex</td>
<td>3.512</td>
<td>0.006</td>
</tr>
</tbody>
</table>
Figure 5-3. Tire modeling procedure in ABAQUS: cross section and reinforcements modeling, rim mounting and tread implementation, steady state rolling analysis.
5.4 Numerical Methodology for FSI Problem

The main models considered for the theoretical foundations of the hydroplaning simulations include: the tire model, the fluid flow model, and the coupling between the two. The fluid flow model in the presented new approach is made of k-ω turbulence and two-phase flow model.

5.5 Fluids modeling using CFD

The commercial software Star-CCM+ is employed for the CFD work and provides the advantage for FSI with the capability to couple with ABAQUS for the FE modeling. For the fluid flow, the Navier-Stokes equations are applied coupled with turbulence and multiphase flow models. The fundamental equations of fluid dynamics are based on conservation of mass (continuity), momentum and energy, shown as:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \quad 5-2
\]

\[
\frac{\partial (\rho \vec{v})}{\partial t} + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \overline{\tau} \quad 5-3
\]

\[
\frac{\partial (\rho E)}{\partial t} + \nabla \cdot [\vec{v} \cdot (\rho E)] = \nabla \cdot k_{eff} \nabla T + \nabla (\overline{\tau_{eff}} \cdot \vec{v}) \quad 5-4
\]

where \( p \) is the pressure, \( \overline{\tau} \) is the fluid stress tensor, \( \vec{v} \) is the fluid velocity vector, \( E \) is the total energy and \( k_{eff} \) is the effective conductivity.
5.6 SST k-ω turbulence model

The shear-stress Transport (SST) k-ω model has been coupled in the CFD model for its ability to predict the onset and flow separation from surfaces [86]. The SST k-ω model combines the k-ω turbulence model to predict the boundary layer and the k-ε to predict the free stream flow. The advantage is that other turbulence models do not account for the transport of the turbulent shear stress, thus over-predicting the eddy-viscosity. The proper transport behavior can be obtained by using a limiter to the formulation of the eddy-viscosity:

\[ \mu_t = \frac{\rho k}{\omega} \frac{1}{\max\left[\frac{1}{\alpha'}, \frac{SF_2}{\alpha \omega}\right]} \]  

where \( S \) is the strain rate magnitude and \( \alpha \) is the damping coefficient for turbulent viscosity causing low-Reynolds number correction. The turbulence kinetic energy \( k \) and the specific dissipation rate \( \omega \) are solved numerically from the following transport equations using the SST k-ω model:

\[ \frac{\partial}{\partial t} (\bar{\rho} k) + \nabla \cdot (\bar{\rho} \vec{k} \vec{v}) = \nabla \cdot (\Gamma_k \nabla k) + G_k - Y_k \]  

\[ \frac{\partial}{\partial t} (\bar{\rho} \omega) + \nabla \cdot (\bar{\rho} \omega \vec{v}) = \nabla \cdot (\Gamma_\omega \nabla \omega) + G_\omega - Y_\omega \]  

\[ \frac{\partial}{\partial t} (\bar{\rho} E) + \nabla \cdot [\bar{\vec{v}} \cdot (\bar{\rho}E + P)] = \nabla \cdot k_{eff} \nabla T + \nabla(\overline{\tau_{eff} \cdot \vec{v}}) \]

where \( G_k \) represents the generation of turbulence kinetic energy due to mean velocity gradients, \( G_\omega \) is the generation of specific dissipation rate \( \omega \), \( k_{eff} \) is the effective diffusive, \( Y_k \) and \( Y_\omega \) represent the dissipation of \( k \) and \( \omega \) due to turbulence.
5.7 Two-phase flow modeling

The volume of fraction (VOF) is used to model water and air, and their flow surrounding the tire, such as splash, spray and wind effects, by tracking the volume fraction of each fluid throughout the domain. The volume fraction of all phases must sum to unity, which means that the variables and properties of in a given cell are purely a representation of one of the phases, or representative of a mixture, depending on the volume fraction. If the volume fraction \( \phi = 1 \), the cell is full of the primary fluid (e.g., water) while if the value is 0, the cell is full of the secondary fluid (e.g., air). For volume fractions between 0 and 1, the cell contains an interface between the primary and secondary fluids. The tracking of the interface between the phases can be accomplished by solving the conservation of mass equation for the volume fraction of one or more phases. To track sharp interfaces, the convective terms of the volume fraction transport equation are discretized using a high resolution interface capturing (HRIC) scheme [87]. For the primary phase, the equation is:

\[
\frac{\partial \phi}{\partial t} + \mathbf{v} \cdot \nabla \phi = 0
\]

where \( \phi \) represents the space occupied by each phase (water and air) and mass and momentum conservations are specified by each phase individually, where the water and air volume fractions sum to one:

\[
\phi_w + \phi_a = 1
\]

The fluid density in Eqs. (5-1 to 5-6) is solved as the volume-fraction-averaged density, which in each cell is given by:

\[
\rho = \phi_w \rho_w + (1 - \phi_w) \rho_a
\]

where \( \rho_w \) is the water density and \( \rho_a \) is the air density.
5.8 Computational domain and boundary conditions

To analyze the water flow field in the tire grooves during tire-ground contact, the shape of the deformed tire must be known. The assumption of the tire deformation was simplified according to theoretical and experimental work (see, e.g., [88-94]). However, other researchers [23] used the extraction of a deformed tread shape in the tire deformation FE model [23] and acquired the deformation with the wheel load. In the work herein, the computational mesh size was determined by balancing the accuracy for representing the hydrodynamics and tire dynamics with reasonable computational processing demands. Figure 5-4 illustrates the computational domain and boundary conditions with deformed tire geometry extracted from FEM to capture the fluid dynamics, aerodynamics, and interactions with the moving tire. The spindle position of the wheel is used as the reference point such that the tire rotates with angular velocity \( \omega_Y \), which can be calculated as the ratio between the vehicle speed \( V_{\text{tire}} \) and tire radius \( R_{\text{tire}} \). The air and water are defined with the relative velocity magnitude, which is the same as the vehicle speed but in the opposite direction of the moving tire. The pavement is also specified with the same relative velocity magnitude in the \( z \)-direction.
The boundary conditions have been specified to represent, as closely as possible, a realistic hydroplaning scenario. The front plane of the computational domain (labeled “inflow”) in Figure 5-4 specifies a uniform velocity corresponding to the vehicle speed for the inflow of the two fluids, water and air. The pavement is defined with the no-slip wall boundary condition with the same relative velocity. A uniform water film thickness is added above the pavement as the initial condition with the same velocity as the pavement. The rear computational plane (“outflow”) specifies atmospheric pressure. The tire is modeled as a no-slip surface with no axial velocity but a constant angular velocity, corresponding to the rotating speed in the FE model. To reduce the
effects of ambient boundaries that define the flow field, the two side surfaces and the top surface of the computational domain are specified as a free-slip boundaries with zero shear stress.
Figure 5-5. Mesh for the Star-CCM+ computational domain: (a): Bald tire (b) the treaded tire model; (b) the whole fluid domain

The geometry of the tire is imported from the FE tire model and then re-meshed in Star-CCM+ with the fluid domain included. The tire mesh and contact patch is shown in Figure 5-5 (a) and the overall mesh of the fluid model is shown in Figure 5-5 (b). Polyhedral cells were created to ensure sufficient grid resolution to capture the fine details of the tire tread and capture the fluid impact on the tire. The “advancing layer mesher” has been employed to generate layers of prismatic cells around the tire structure and fill the remaining voids with polyhedral cells. The advantage of this surface mesh method is that each side of the interface can be mapped/matched conformally. To save computational time, along the pavement, the mesh gets coarser with the cell size increase by a ratio of 1.3.
5.9 Fluid Structure Coupling

Fluid-structure interaction is achieved by coupling the FEM and the explicit FVM, which provides stable convergence, while other approaches can generate oscillations and can even produce unstable behaviors. Arbitrary thin structures can be handled by the model, including non-linear effects such as the contact between rigid and flexible walls [95]. The coupling between the CFD model and ABAQUS model in Star-CCM+ transfers information between the fluid model and tire model. An outline of the algorithm and coupling approach are shown in Figure 5-6.

The coupling exchanges time-dependent data such as pressure and nodal coordinates during tire deformation between the CFD and FE solvers. Using direct co-simulation coupling results in more efficient data transfer compared to the use of an external “middleware” software.

5.10 CEL Method in ABAQUS

The co-simulation approach using Star-CCM+ will be compared with the conventional CEL method, which utilizes both Eulerian and Lagrangian approaches. The CEL method has been...
shown to be highly efficient compared to the physics-based models when the interaction between tire and water or soil is studied [96]. The Eulerian domain is shaped to capture the fluid motion whereas the Lagrangian approach tracks the movement of the structure as well represents the interface between the fluid and structure. At each time increment, the Eulerian domain is recreated, and new boundaries are shaped. To do so, computed volume fractions from the Eulerian elements are used [97]. In the ABAQUS CEL approach, the tire displacement is solved instead of the velocity. To prevent the elements from dramatically deforming, the Lagrangian mesh is remapped using the original Eulerian mesh through interpolation. Although the method assumes that the fluid is compressible in the Navier-Stokes equations, a very high bulk module $K$ is used to approximate an incompressible fluid. The governing conservation equations for mass and momentum solved in CEL are:

$$\frac{D\rho}{Dt} + \rho \nabla \cdot \frac{Dd}{Dt} = 0$$

5-12

$$\rho \frac{D^2d}{Dt^2} - \rho g - \nabla \cdot \sigma = 0$$

5-13

where $g$ is gravity, $d$ is displacement and $\sigma$ is Cauchy stress tensor that is related to the equation of state (EOS). As shown in Eqs. 5-12 and 5-13, the displacement will be solved instead of velocity. Also, the Mie-Gruneisen form of EOS is used in ABAQUS defining the relation between pressure and density:

$$p - p_H = \Gamma \rho(e - E_H)$$

5-14

where $p_H$, $E_H$, and $e$ are the Hugoniot pressure, Hugoniot specific energy per unit mass and energy density, respectively, which are functions of density only. Using Eq. 5-14 decouples the energy equation from the continuity and momentum equations whereby pressure is only function of
density and not a function of the energy density. For the case of hydroplaning, the EOS for water is simplified:
\[ p = p_0 c_0^2 (1 - \frac{\rho_0}{\rho}) \] 5-15

where constants and are obtained from experimental data.

The aforementioned assumptions can be the origin of errors associated with the conventional CEL method for which the co-simulation approach correctly models. For further details regarding the CEL method, readers are referred to [97-100].

5.11 Results and Discussions

5.11.1 Grid Convergence Study

A grid resolution study is conducted for a bald tire in both FSI approaches, in terms of the overall lift force. The lift forces calculated on the tire surface are computed as:
\[ f = \sum (f^p + f^s) \cdot n_f \] 5-16

where \( f \) is the vertical component of the tire force, superscripts \( p \) and \( s \) are the pressure and shear force, respectively, acting on the tire surface and \( n_f \) is the vertical unit vector. The pressure force and the shear force on the tire surface are computed as:
\[ f^p = (P - P_{ref}) A_t \] 5-17
\[ f^s = -T_f \cdot A_t \] 5-18

where \( P_{ref} \) is the reference pressure, \( A_t \) is the tire face area and the \( T_f \) is the stress tensor exerted on the tire surface area by the fluid combination of air and water.
As it is shown in Figure 5-7, extensive efforts are made for the mesh refinement. Using combination of layers of prismatic cells around the tire structure and fill the remaining voids with
polyhedral cells assist each side of the interface to be mapped/matched conformally and improve the results accuracy.

To save computational time, along the pavement, the mesh gets coarser with the cell size increase by a ratio of 1.3. The grid convergence index (GCI) methodology [12] is used to determine the numerical accuracy of the solutions for each grid resolution and quantify errors that are caused from discretization or truncation in the CFD simulations [101]. The bald tire is studied under a 2100 N weight load with high tire pressure. The tire speed is 17.9 m/s with 5 mm water film thickness. Three mesh densities are selected with the base sizes of $h_1$, $h_2$ and $h_3$ for the fine, medium and coarse meshes, respectively, shown in Table 5-3. The base size is the core characteristic dimension of the grid cell. The meshing refinement function has been kept the same for the three tested cases. The total cell numbers for three mesh densities in Star CCM+ are $1.48 \times 10^5$, $5.10 \times 10^5$ and $1.58 \times 10^6$. Also, the total cell numbers for three mesh densities in ABAQUS CFD are $1.575 \times 10^6$, $4.70 \times 10^5$ and $1.39 \times 10^5$.

Figure 5-8. Contour of the pressure force for bald tire Star-CCM+ simulation using (a) coarse mesh; (b) medium mesh and (c) fine mesh
Figure 5-8 compares the pressure on the bald tire for the coarse, medium and fine grid resolutions in Star CCM+. It is evident that with increasing grid resolution, the effects of the pressure are more resolved. Further details related to the pressure distribution will be discussed with Figure 5-9.
Figure 5-9. Tire pressure distribution and water splash with two-way coupled model: (a) fluid pressure acting on the tread tire, (b) total lift force versus, time (c-e) water splash at time $t = 0.01, 0.015$ and $0.02 \text{ s}$.

To quantify the GCI, the total lift forces have been calculated at a quasi-steady condition with physical time of $0.04 \text{ s}$, in Table 5-3. The comparison of the total lift forces for the three grid resolutions reveals that the lift forces share similar magnitudes in both methods. The global apparent order of accuracy $P$, in the GCI method in Star CCM+ and ABAQUS CFD are $4.54$ and $3.21$, respectively. The GCI values for the predicted total lift forces from the co-simulation of StarCCM+ and ABAQUS are $8.4\%$ and $1.4\%$ for the coarse-medium and medium-fine grids, respectively. Also, the GCI values for the predicted total lift forces in ABAQUS-CEL are $6.3\%$ and $1.6\%$ for the coarse-medium and medium-fine grids, respectively (the full GCI results are given in Table 5-3).

Table 5-3 Discretization error calculation using the grid convergence index [12]

<table>
<thead>
<tr>
<th>Variable</th>
<th>StarCCM+</th>
<th>ABAQUS CFD</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_1$, $h_2$, $h_3$</td>
<td>0.02, 0.03, 0.045 m</td>
<td>1.67, 2.5, 3.75 mm</td>
</tr>
<tr>
<td>$N_{\text{fine}}, N_{\text{med}}, N_{\text{coarse}}$</td>
<td>1.58 m, 510 k, 148 k,</td>
<td>1.575 m, 470 k, 139 k</td>
</tr>
<tr>
<td>$r_{21}, r_{32}$</td>
<td>1.5</td>
<td>1.5</td>
</tr>
<tr>
<td>$P$</td>
<td>4.54</td>
<td>3.21</td>
</tr>
<tr>
<td>$GCI_{23}$</td>
<td>8.4%</td>
<td>6.3%</td>
</tr>
<tr>
<td>$GCI_{12}$</td>
<td>1.4%</td>
<td>1.6%</td>
</tr>
</tbody>
</table>
Based on the GCI values, it is obvious that the refinement of the mesh improves the numerical accuracy significantly. To maintain consistent accuracy and to ensure sufficient mesh density to capture the details of the tire tread for the co-simulation in StarCCM+, the fine base size 0.02 m was used for the remainder of the study for the tread and half tread tire simulations. Applying the same meshing refinement function, the tread tire simulations use 2.73 million grid cells. It is noteworthy to mention that since the fluid Eulerian domain in ABAQUS Explicit of CEL method can move with the tire in the longitudinal direction, it is possible to utilize a smaller CFD domain around the tire compared to computational domain in Star CCM+.

5.12 Model Verification Using Lift and Lateral Force

The critical hydroplaning speed can be used to verify the computational predictions of the CFD-FE co-simulation and the CEL model with empirical relations and published experiments. The NASA empirical equation [35] and the modified equation [102], respectively, are:

\[ v_{cr} = 51.80 - 17.15(AR) + 0.72p_l \]  \hspace{1cm} 5-19

\[ v_{cr} = p_l^{0.5} F_y^{0.2} \left( \frac{0.82}{WFT^{0.06}} + 0.49 \right) \]  \hspace{1cm} 5-20

In order to use Eqs. 5-19 and 5-20, some assumptions are required. For the NASA equation (Eq. 5-19), the normal load on the wheel is considered to be 4000 N. Also, Horne [35] provided a new equation for a 0.3 m passenger car tire based on 7.6 mm WFT on the road. Since Eqs. 5-19 and 5-20 are applicable for conditions in which the WFT is more than the depth of the tire tread, the half tread tire is used to validate both computer simulations. The critical speeds found by
applying Equations 5-19 and 5-20 are 25.4 m/s and 24.0 m/s respectively. Using the same condition with the FSI model for coupling the FE tire model with ABAQUS CFD, the critical speed is 23.3 m/s yielding 5.6% error with the empirical equation. Likewise, the FE tire model with Star-CCM+ predicts a critical speed of 24.5 m/s with 2.7% error. The computational results from both FSI approaches show good agreement with the empirical equations.

Having verified the FSI models from both approaches by comparing the critical hydroplaning speed with empirical equations, the lift forces and cornering forces are examined. Different speeds and slip angles on the pavement with 5 mm WFT are examined for the lift and cornering forces. In addition to the treaded tire that is the main focus of this work, a bald tire is studied as well.

Figure 5-9 shows the tire pressure distribution, water splash and transient development of the total lift force acting on the tire. Figure 5-9 (a) presents the pressure of the fluid acting on the tire and the contact patch (marked as black edges). Most notable is that the contact patch is the region of the highest pressures and resulting when the water strikes the tire surface during the tire rotation. Pressure is also very high at the edges of the tire tread contacting the pavement. This could be because the tread pattern creates narrow channels for water to pass, which greatly increases the turbulence level of the fluid flow and increases the corresponding fluid pressure on the tire. Figure 5-9 (c-e) illustrates the time-dependent interaction between the water and the tire using an iso-surface of water volume fraction equal to 0.3 or larger. Water channels form at the front of the tire through the treads and the is redirected to the sides of the tire as the tire rotates, which is represented as water splash. Figure 5-9 (b) shows the lift force acting on the tire with time, where after 0.02 s, the lift force asymptotes to a constant force. The negative value corresponds to the direction of the forces, which is acting to the opposite direction of the gravity force based on the coordinates defined in Figure 5-10. The lift force shows a spike increase from 0 s to 0.005 s as
water starts to build up at the front edge of the contact area and below the tread. It then decreases since water gets discharged from the tread to the lateral sides.

As shown in Figure 5-10 for a bald tire and treaded tire, the lift forces predicted using the FE tire model with ABAQUS -CEL and Star-CCM+ co-simulations increase with vehicle speed. Also, the lift force for bald tire reaches the saturated value at lower speeds, as expected. Generally, the results obtained from ABAQUS-CEL and integration of FE tire model with Star-CCM+ are similar. The improvement with the Star-CCM+ predictions can be associated with the more accurate turbulence model used to capture the behavior of the fluid at length scales smaller than the smallest element length in the CEL method.

As the lift force caused by the hydrodynamic pressure in the fluid domain changes, the cornering force of the tire also changes. In vehicle dynamics, the slip angle is the angle between
the direction that the wheel is traveling and the direction that it is pointing. The net lateral force on each tire is generated by the lateral deformation of the tire surface, which is nominally zero at the leading edge and linearly increases until localized slip begins to occur. The location along the tire contact patch at which localized slip begins is dependent on the friction and the localized vertical force \[103\]. The capability to generate lateral forces is closely coupled with the ability to generate longitudinal forces (the total tractive force available being approximately equal to the vector sum of the two). Thus, as the lateral force increases, less longitudinal force is available for braking and thereby can significantly increase the hydroplaning risk. A schematic demonstrating slip is shown in Figure 5-11.

![Figure 5-11. Schematic of the vehicle tire with slip angle](image)

In Figure 5-12 and Figure 5-13, the lateral force versus slip angle for 5 mm WFT at 17.8 and 22.4 m/s are presented, respectively. As shown in Figure 5-12, the lateral force increases with slip angle up to a saturation limit at which the maximum available tractive force is reached. Wet
conditions can have at least three detrimental effects on the ability to generate tractive forces. Specifically, at the tire-road contact patch, three things are reduced: the lift force created by the water film reduces the vertical force at the tire-road contact patch (reduced by the lift force exerted at the tire-water interface), the effective friction, and the size of the tire-road contact patch. Since the tire-road interface supports a shear force, but the tire-water interface does not, it is clear that the available lateral force saturates at lower slip angles as caused by the hydroplaning effect. These effects are exacerbated at higher speeds because both the size of the tire-road contact patch and its effective vertical force is reduced due to the increased size and lift force of the tire-water interface.

For lateral forces obtained from the Star-CCM+ co-simulation, the lateral force saturates at higher slip angles as compared to the ABAQUS-CEL model. This is due to slightly lower lift force captured through the co-simulation method using Star-CCM+, which also incorporates a model for two-phase flow. For the dry condition, since there is no need to have the CFD code running in StarCCM+, only the tire model in the ABAQUS Implicit is used.
Figure 5-12. Cornering force at 40 mph for treaded and bald tires using different co-simulation methods for wet conditions and a treaded tire for dry conditions

The cornering force for treaded and bald tires using ABAQUS -CEL at 22.4 m/s is presented in Figure 5-13. It is obvious that the effect of the lift force from hydroplaning is more significant at higher speeds. Accordingly, the tire reaches its cornering saturation at lower slip angles with lower corresponding lateral force. As expected, a lower lateral force for the bald tire as compared to the treaded tire is obtained, which is due to higher lift force caused by lack of presence of grooves for the bald tire. When a vehicle moves on wet pavement, the friction coefficient between tire and pavement is less than for dry conditions. In addition, as the vehicle speed increases on wet
pavement, the dynamic pressure induced by the water pushes the tire upward and increases the propensity to hydroplane.

![Graph showing cornering force at 50 mph for treaded and bald tires using different co-simulation methods for wet and dry conditions.](image)

Figure 5-13. Cornering force at 50 mph for treaded and bald tires using different co-simulation methods for wet and dry conditions

The results from the simulations are compared with the experimental data provided by CALSPAN Tire Research Facility (TIRF) [104] shown in Figure 5-14. Although the tire size tested by CALSPAN is different from that of this study, the experimental tests can be utilized to draw a comparison between the trend of lateral forces generated on wet and dry conditions. Based on the result provided in [104], the experimental results reveal that the peak lateral force at each slip angle in wet conditions is less than that of dry conditions. In addition, the difference between the lateral
forces at dry and wet condition increase with slip angle, which has agreement with the simulation results in this work.

Figure 5-14. Experimental data of the cornering forces for wet and dry conditions at different weight loads [104]

Figure 5-15 summarizes the water-tire interaction with the bald tire and the treaded tire under 0 degree and 2 degrees slip angles and shows the water splash and spray. The fluid free surface is captured with the iso-surface specified with a water volume fraction of 0.3. Comparing Figure 5-15 (a-b) of the water distribution between the bald tire and the treaded tire at 0 degree slip angle, both cases demonstrate that water splashes along the sides of the tire. However, the water distributions for the bald tire and treaded tire are very different. For the tread tire, it is clear that water separates into streams and is displaced with the tire grooves, which is not observed with the bald tire. When roads are wet, tire grooves channel water away, which helps prevent tires from
hydroplaning, unlike the bald tire scenario where the traction is reduced, and hydroplaning occurs more readily.
Figure 5-15. Iso-surface of water volume fraction of 0.3 showing water splash for (a) bald tire with 0 degree slip angle; (b) tread tire with 0 degree slip angle; (c) tread tire with 2 degrees slip angle

The effect of slip angle on water splash is demonstrated in Figure 5-15 (c), whereby the water splash is more significant along the side of the tire, whereby the lateral force increases, as shown in Figure 5-12 to Figure 5-13. More streams of water have been squeezed out of the tire to ensure a good contact between tire and ground because of the tire tread. Comparing between Figure 5-15 (b-c) with different slip angles, the addition of slip angle has greatly increased the water splash on the tire side. Since the force along the lateral direction caused by such water splash is only a small portion of the force generated laterally due to the tire-pavement interaction and slip angle, the cornering forces because of the slip angle are mainly studied from the output in the tire-pavement FE model.

5.13 FE Model Contact Patch Comparison with the Results Obtained from the Intelligent Tire Technology Previously Developed at CenTiRe

5.13.1 Experimental Result

Apart from varying the lateral and lift forces from the model with the data obtained in the literature, the contact patch length is compared with the experimental data using the intelligent tire
technology developed in CenTiRe [105-109]. An instrumented VW Jetta equipped with intelligent tires (sensors inside the tires) is used to collect data on the Virginia Tech Transportation Institute (VTTI) Smart Road. Smart Road’s weather-making towers sprayed water on the road to produce, in average, up to 2.5 inches of rain per hour. Since the number of active towers were decreased to limit the wet region to the specific road textures, more rain rate is obtained in the test region. Using various gages, water film thickness was estimated for three locations on the test surface. Figure 5-16 shows the gage used for measuring the rainfall during the test. Rainfall rates are summarized in Table 5-4.

![Rain Gauge Image](image)

Figure 5-16: The rain gauge used to measure the rainfall rate

<table>
<thead>
<tr>
<th>Tower No.</th>
<th>Total Rain (mm)</th>
<th>Rain Duration (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>36</td>
<td>79</td>
<td>59</td>
</tr>
<tr>
<td>41</td>
<td>70.8</td>
<td>59</td>
</tr>
<tr>
<td>46</td>
<td>129</td>
<td>59</td>
</tr>
</tbody>
</table>

Table 5-4: The measured rain fall rate during the test
With regards to the roughness of the asphalt and the side slop of the road, it is difficult to measure the water film thickness accurately, however, the water film thickness of the test region is estimated at 2 to 3 mm.

5.13.2 Test setup summary of the utilized intelligent tire technology previously developed in CenTiRe

The instrumented vehicle was equipped with the steering wheel sensor to measure the steering wheel angle and torque. Also, inertial measurement unit is used to measure the roll, pitch and yaw during the test. VBox is used to record the vehicle speed. 2 intelligent tires with accelerometer and encoder have been used. Triaxial accelerometers are placed and attached on the tire inner liner using epoxy glue to monitor the interaction between the tire and the road. A data collection routine has been developed in LabView, which uses USBNI DAQ to synchronize and collect all the sensor data. Data sampling rate of 5000 Hz was used for all the sensors, which is high enough for the purpose of this study.

Figure 5-17. Collecting and pre-processing of the tire signals
The instrumented vehicle is shown in Figure 5-18. During initial rounds of the test, it was observed that using the windshield wipers adds considerable noises into the collected data. As a result, it was tried to keep the wipers off during the data collection. Tests were conducted at 2 different days. day 1 devoted to dry condition tests and day 2 devoted to the wet condition tests.

![The instrumented vehicle used for collecting the data](image)

Figure 5-18. The instrumented vehicle used for collecting the data

To estimate the length of the contact patch, several robust algorithms have been already developed in CenTiRe through processing of the different signals such as circumferential and radial accelerations [105-109]. In addition, different methods are reported in the literature to analyze the time dependent signals such as autocorrelation, cross correlation, Root mean square (RMS) and peak analysis [110]. In this work, a peak detection algorithm is employed to identify the leading and trailing edge of the tire from the circumferential acceleration signal toward estimation of the contact patch length. A flowchart including the signal processing steps is shown in Figure 5-19.
Tests were conducted at different speeds, and it was observed that the effect of lift force on the contact patch length is not notable at low speeds. However, it was seen that the effect of the lift force from the water hydrodynamic force increases with the vehicle speed. Figure 5-20 shows the contact patch length at 50 mph on the wet and dry surfaces.
It is obvious that the length of the contact patch decreases due to the reduction of normal load on the tire. The length of the contact patch on the dry surface at 50 mph is nearly 128 mm and the length of the contact patch on the wet surface with the same speed is nearly 118 mm. It is noteworthy to mention that during the tests, considerable efforts are made to keep the vehicle speed constant during the data collection to make sure that the load transfer caused by the braking/acceleration does not affect the length of the contact patch.

### 5.13.3 Contact patch length obtained from the FSI analysis

In this part, the tire hydroplaning model with 3 mm water film thickness is run to obtain the length of the contact patch at 50 mph affected by the hydroplaning force.

Figure 5-21 shows the effects of the lift force and slip angle on the contact patch. CPRESS variable (N/mm²) obviously shows that the length of the contact patch decreases with existence of the water on the surface.
Figure 5-21. Effect of slip angle on the contact patch W/WO existence of water film at 40 mph, a: 0 deg slip angle dry pavement, b: 2 deg slip angle 5mm water film thickness and c: 0 deg slip angle 5mm water film thickness.

The volume fraction of the water is shown in the Figure 5-22. The tire is moving from right to left, as a result, the water volume fraction before the tire reaches to the left region is 1 and by passing the tire through the water film, the water is pushed to sides and water volume fraction at contact edge decreases.

Figure 5-22. Volume fraction of the water flowing in the tire pattern groove with 5 mm water film thickness at 40 mph

Figure 5-23 shows the length of the contact patch at three different simulation time steps. At the beginning of the simulation (Figure. 5-23 (a)), the contact patch has not shaped yet, as a result we
need to have progress in the simulation to evaluate the initial contact patch which is shown at the next time step (Figure. 5-23 (b)).

The length of the contact patch at the end of the simulation is shown in Figure. 5-23 (c). The length of the contact patch at the end of the simulation is 118.4 mm which has a very good agreement with the length of the contact patch obtained from the intelligent tire test. The length of the contact patch on the dry surface from the experiment is less than the one obtained from the FSI analysis with almost 9% difference.

### 5.14 Further investigation from the FSI Simulations

In this section, using the developed and verified FSI model, a matrix of the cases including different speeds, tire pressures, slip angles and water film thicknesses (WFTs) were run. These runs illustrate the effect of WFT on the lateral and vertical (lift) forces on the tire for all the simulation cases. For example, the first three rows show how the lift force increases and the lateral force decreases with increase in the WFT.

Table 5-5. FSI simulation results for lift and cornering force at different dry/wet conditions
<table>
<thead>
<tr>
<th>Vertical Load (N)</th>
<th>Speed (mph)</th>
<th>Tire Pressure (psi)</th>
<th>Slip Angle (Degree)</th>
<th>Tire Tread</th>
<th>WFT (mm)</th>
<th>Lift Force (N)</th>
<th>Lateral Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4100</td>
<td>40</td>
<td>32</td>
<td>2</td>
<td>New tread</td>
<td>2</td>
<td>1050</td>
<td>1650</td>
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<td>2</td>
<td>New tread</td>
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<td>1676</td>
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<td>New tread</td>
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<td>1250</td>
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<td>New tread</td>
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<td>720</td>
</tr>
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<td>4100</td>
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<td>2376</td>
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<td>2100</td>
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<td>Bald tread</td>
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<td>1250</td>
<td>1270</td>
</tr>
<tr>
<td>2100</td>
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<td>32</td>
<td>10</td>
<td>Bald tread</td>
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Figure 5-24 shows that the lift force caused by the hydrodynamic pressure from the water increases with the WFT. The effect of the WFT on the lateral force is also shown which decreases with the WFT.
Figure 5-24. The effect of WFT on vertical and lateral forces at 2 degrees slip angle and 40 mph obtained from the baseline FSI model.

The FSI model is modified with the kinetic friction coefficients obtained from section 4.4. As a result, the matrix of the simulations is generated for a few cases with the modified friction coefficient and the results obtain from the new analysis shown in Table 5-6.

Table 5-6. The results from the FSI simulation with the modified friction coefficient

<table>
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<tr>
<th>Vertical Load (N)</th>
<th>Speed (mph)</th>
<th>Tire Pressure (psi)</th>
<th>Slip Angle (Degree)</th>
<th>Tire Tread (mm)</th>
<th>WFT (mm)</th>
<th>Lift Force (N)</th>
<th>Lateral Force (N)</th>
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The results obtained from the FSI model with modified kinetic friction coefficient illustrate that change in the friction coefficient leads to change in both the lift force and the lateral force toward providing more information from the pavement roughness to shape the simulation closer to the real scenario.

### 5.15 Conclusions

In this chapter, two different approaches were developed to study the effects of hydroplaning on lift and cornering forces for a passenger car tire. Both approaches were verified with empirical equations and compared well overall. The conventional hydroplaning modeling approaches mostly rely on solving a single-phase fluid as an Eulerian fluid elements which are assumed inviscid and laminar flow. Such an assumption places some limitations on the simplified hydroplaning model such as ignoring the effect of shear stress at the interface.

In the new approach to the address the limitation associated with the conventional approaches the Navier-Stokes equations are applied for the fluid flow and are coupled with turbulence and multiphase flow models. The shear-stress Transport (SST) \( k-\omega \) model has been coupled in the CFD model for its ability to predict the onset and flow separation from surfaces The SST \( k-\omega \) model
combines the $k-\omega$ turbulence model to predict the boundary layer and the $k-\varepsilon$ to predict the free stream flow.

In this work, it was shown that the new approach using ABAQUS and Star-CCM+ co-simulation had slightly better agreement with the empirical models showing a 2.9% improvement. Also, the lateral forces obtained from simulation show a good agreement with the experiment data reported in the literature.

It is noteworthy to mention that the new approach using Star-CCM+ needs more computation time and memory, however, with advent of supercomputers it can be more applicable for the future hydroplaning simulations. In addition, in the new hydroplaning modeling approach, free surface flow with moving boundaries to build and track the amount of water as well as the volume fraction of other fluids can be captured, which can be used to study tire tread design and its relationship to water splashing and aerodynamics. In fact, the new proposed approach also provides the opportunity for the R&D engineers to study the air flow around the moving tire and develop advance technologies to control the air flow near the tire surface towards developing tires with better aerodynamics performance [111].

Last but not least, in this work, a multi-scale modeling approach to predict the hysteresis friction coefficient (presented in chapter 4) is introduced as a methodology that can be utilized in the hydroplaning analysis towards providing much more robust model, considering the surface roughness.
6 CONCLUSION AND FUTURE WORK

6.1 Conclusion

In this work, a novel multi-scale finite element model is presented to enhance the hydroplaning model capabilities. Before using a full-scale tire model, interactions of the tread block with a specific surface is studied. To do so, several mechanical tests data such as uniaxial, biaxial, planar (shear), and DMA are used to build a comprehensive material model to predict the hyper-viscoelastic properties of the rubber accurately.

Calibrating visco-hyperelastic material model using MATLAB genetic algorithm at 3 different loading modes improved the FE model prediction.

Using multi-scale modeling techniques, the friction coefficient between the tire and pavement, for wet conditions, is characterized via developing 2D and 3D model, representing the rubber tread interacting with the rough surface. In this work, the roughness of the specific surface included into the FE model. Decomposing the surface roughness into different scales facilitate considering the surface roughness in FE analysis. The multi-scale rubber block sliding model employed in this work, facilitates obtaining the hysteresis friction coefficient at different normal load and sliding velocity which is suggested to be used instead of a single value for the friction coefficient which is previously utilized in the hydroplaning analysis.

Furthermore, using a validated tire model, fluid-structure interaction (FSI) between the tire-water-road surfaces are investigated through two approaches. In the first approach, the coupled Eulerian-Lagrangian (CEL) formulation was used. The drawback associated with the CEL method is the laminar assumption that the behavior of the fluid at length scales smaller than the smallest element
size is not captured. To improve the simulation results, in the second approach, an FSI model incorporating finite-element methods and the Navier-Stokes equations for a two-phase flow of water and air, and the shear stress transport k-ω turbulence model, was developed and validated, improving the prediction of real hydroplaning scenarios.

6.2 Limitations and Future Work

6.2.1 Limitations

In this work, a multi-scale modeling approach is presented to obtain the hysteresis friction coefficient toward obtaining more information of the road surface roughness into the friction coefficient to define the interaction of tire and wet road surface. Although, the suggested procedure includes a pressure and slip rate dependent friction instead of a single friction coefficient, the effect of the other components of the tire such as side wall stiffness has not been considered. In this model, the water film thickness (WFT) is considered as an input. As a result, the effect of the pavement roughness and the road slop on WFT are neglected.

6.2.2 Future work

Recent study reveals that the friction theory based on hysteresis energy sometimes failed to evaluate the wet traction performance in extremely low speed. Studies show the effective friction coefficient is only 3% larger than the friction coefficient due to energy dissipation in the contact area. The small sliding velocity in the front part of the contact patch (deformation region) supporting adhesion, dewetting may play a more important role concerning safety during cornering or ABS-braking. As a result, as part of the future work, the friction behavior of rubber in wet condition can be studied more precisely.
Accordingly, an analytical model to predict the frictional losses due to viscous resistance at the contact interface at wet condition can be developed which is part of new project that is accepted by CenTiRe’s Industry Advisory Board.

Using digital image correlation (DIC) system and high-speed IR camera to characterize the contact interface can be considered as a part of the future work which the suggested related test setup is shown in Figure 6-1.

Figure 6-1. The digital image correlation (DIC) test setup suggested for the further study of the rubber sliding on the linear friction tester

In this work, a 120-grit surface is used as a rough surface however, different material such as concrete and asphalt can be used for the multi-scale modeling which are shown in Figure 6-2.
As a part of the future works, it is suggested that the outcome of the FSI model would be used to generate synesthetic data toward training an artificial deep neural network (ADNN). In addition, advent of new technology such as computer vision toward prediction of the pavement friction can be incorporated with a hydroplaning feed forward predictive model to estimate the tire force with regard to the road condition. The suggested machine learning-based hydroplaning model (MLHM) can be incorporated with a vehicle dynamic model to be implemented in the future vehicles. The suggested framework is shown in Figure 6-3.
Figure 6-3. Suggested framework for extending the current work for MLHM for implementation in the future vehicles
7 APENDIX

7.1 Stress/strain contours from the 3D model setup

The summary the strain and the stress for the several cases studied in this work are as follow:

7.1.1 Distribution of strain within the rubber block at different normal load and sliding
Figure 7-1. Distribution of strain within the rubber block at different normal load and sliding velocity on the rough surface: a) max principal strain at 0.3 MPa normal load and 0.1 m/s sliding velocity, b) max principal strain at 0.3 MPa normal load and 0.2 m/s sliding velocity, c) max principal strain at 0.3 MPa normal load and 0.3 m/s sliding velocity, d) strain component LE12 at 0.3 MPa normal load and 0.1 m/s sliding velocity, e) strain component LE12 at 0.3 MPa normal load and 0.2 m/s sliding velocity, f) strain component LE12 at 0.3 MPa normal load and 0.3 m/s sliding velocity.

7.1.2 Distribution of strain within bottom of the rubber block in contact with the rough surface

Comparing the shear strain LE23 and LE13 shown in Figure 7-3, illustrates that these two components of the strain within the bottom of the rubber block are highly dependent on the roughness of the surface than the sliding velocity.
Figure 7-2. Distribution of strain within bottom of the rubber block in contact with the rough surface: a) strain component LE11 at 0.3 MPa normal load and 0.1 m/s sliding velocity, b) strain component LE11 at 0.3 MPa normal load and 0.2 m/s sliding velocity, c) strain component LE11 at 0.3 MPa normal load and 0.3 m/s sliding velocity, d) strain component LE12 at 0.3
MPa normal load and 0.1 m/s sliding velocity, e) strain component LE12 at 0.3 MPa normal load and 0.2 m/s sliding velocity, f) strain component LE12 at 0.3 MPa normal load and 0.3 m/s sliding velocity

As it is obvious in the Figure 7-3, the distribution of the aforementioned components of the strain are similar at different sliding velocities and the location of the asperities are playing more important role than the sliding velocity.
Figure 7-3. Distribution of strain within the bottom of the rubber block in contact with the rough surface: a) strain component LE23 at 0.3 MPa normal load and 0.1 m/s sliding velocity, b) strain component LE23 at 0.3 MPa normal load and 0.2 m/s sliding velocity, c) strain component LE23 at 0.3 MPa normal load and 0.3 m/s sliding velocity, d) strain component LE13 at 0.3 MPa normal load and 0.1 m/s sliding velocity, e) strain component LE13 at 0.3 MPa normal load and 0.2 m/s sliding velocity, f) strain component LE12 at 0.3 MPa normal load and 0.3 m/s sliding velocity

7.1.3 Distribution of stress and strain within the front of the rubber block at different normal load and sliding velocity on the rough surface

Distribution of stress and strain within the front of the rubber block at different normal load and sliding velocity on the rough surface are shown in Figure 7-4. The principal stress at Y direction (S22) increases with the sliding velocity. However, the shear stress at YZ plate (S23) does not show strong correlation with the sliding velocity.
Figure 7-4. Distribution of stress and strain within the front of the rubber block at different normal load and sliding velocity on the rough surface: a) principal stress $S_{22}$ at 0.3 MPa normal load and 0.1 m/s sliding velocity, b) principal stress $S_{22}$ at 0.3 MPa normal load and 0.2 m/s sliding velocity, c) principal stress $S_{22}$ at 0.3 MPa normal load and 0.3 m/s sliding velocity, d) shear stress $S_{23}$ at 0.3 MPa normal load and 0.1 m/s sliding velocity, e) shear stress $S_{23}$ at
0.3 MPa normal load and 0.2 m/s sliding velocity, f) shear stress $S_{23}$ at 0.3 MPa normal load and 0.3 m/s sliding velocity
7.2 Full tire contact patch stress at different normal load and velocity

Figure 7-5. Pressure distribution within the contact patch at 2000 lbf normal load at static condition

Figure 7-6. Pressure distribution within the contact patch at 3000 lbf normal load at static condition
Figure 7-7. Pressure distribution within the contact patch at 4000 lbf normal load at static condition

Figure 7-8. Pressure distribution within the contact patch at 5000 lbf normal load at static condition
Figure 7-9. Pressure distribution within the contact patch at 6000 lbf normal load at static condition

Figure 7-10. Pressure distribution within the contact patch at 4000 lbf normal load and 50 mph with 5mm WFT (step 1)
Figure 7-11. Pressure distribution within the contact patch at 4000 lbf normal load and 50 mph with 5mm WFT (step 2)

Figure 7-12. Pressure distribution within the contact patch at 4000 lbf normal load and 50 mph with 5mm WFT (step 3)
Figure 7-13. Pressure distribution within the contact patch at 4000 lbf normal load and 50 mph with 5mm WFT (step 4)
Figure 7-14. Pressure distribution within the contact patch at 4000 lbf normal load and 50 mph with 5mm WFT (step 5-Complete Simulation)
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