Thermo-Mechanical Instabilities in Next-Generation Friction Materials in High-Speed Sliding Systems

Kingsford Koranteng

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Abstract
For centuries, the manufacturing industry has incorporated metals like copper into friction materials to enhance thermal properties and minimize thermo-mechanical instabilities (TMI) in high-speed sliding systems. Unfortunately, these metals have adverse environmental effects due to the emission of hazardous particulate matter. As a result, there is a growing movement towards adopting next-generation friction materials as an alternative solution.

The study begins by conducting experimental and numerical investigations to examine the instabilities found in metal-based friction materials. The primary objective is to utilize the insights gained from the investigations to computationally explore effective strategies for mitigating various instabilities that may arise in next-generation friction materials when used in high-speed systems.

To study the instabilities in metallic-based friction materials, a Cu-based friction material was developed. The study analyzed the friction properties and wear rate of this friction material while sliding against 65Mn steel using a Universal Mechanical Tester-5, and investigated the effects of sliding speed and temperature on the engagement process. The results offer valuable information on the relationship between the friction coefficient and wear rate; instabilities and critical sliding speed, as well as the growth rate of hotspots in metallic friction materials.

To assess the ability of a carbon fiber-reinforced hybrid composite friction material, which is free of copper, to withstand thermo-mechanical instabilities in sliding materials, a nonlinear transient thermo-mechanical model using Finite Element Code was used. The model was used to identify material properties that significantly affect the onset of TMI and which properties need enhancement for efficient use of the friction material in brakes and clutches.

Furthermore, to tackle instabilities like vibration and noise in automotive disc brakes and clutches, a viscoelastic friction material was proposed as another alternative. A mathematical model was developed to examine the instability of this material, by considering three physical material parameters: relaxation time, elasticity, and thermal conductivity. The study provides an intuitive means of predicting the onset of thermo-mechanical instabilities in viscoelastic friction materials and a better understanding of the influence of viscoelastic parameters in sliding systems.

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Thermo-Mechanical Instabilities In Next-Generation Friction Materials In High-Speed Sliding Systems

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of the Requirements for the Degree
Doctor of Philosophy

by
Kingsford Koranteng
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Advisor: Professor Yun-Bo Yi
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For centuries, the manufacturing industry has incorporated metals like copper into friction materials to enhance thermal properties and minimize thermo-mechanical instabilities (TMI) in high-speed sliding systems. Unfortunately, these metals have adverse environmental effects due to the emission of hazardous particulate matter. As a result, there is a growing movement towards adopting next-generation friction materials as an alternative solution.

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NOMENCLATURES

Physical Variables and Properties

- $a_i$: Half-thickness of the sliding layer
- $b$: The growth rate of the perturbation
- $E_i$: Elastic modulus of the layer
- $k_i$: Thermal diffusivity of layer
- $h_i$: Height of the layer
- $K_i$: Thermal conductivity of the layer
- $L$: Length of the sliding layer
- $m$: The wave number of the perturbation
- $n$: Number of hotspots
- $P_i$: Pressure at the contact surface
- $x$: Sliding distance
- $T_o$: Constant Temperature
- $a$: The sliding velocity of the layer
- $V$: Critical sliding speed of layer $i$
- $\delta$: Damping modulus
- $\phi$: The phase angle between stress and deformation
- $\tau$: Relaxation time
- $\sigma$: Elastic stress
- $\mu$: Coefficient of friction
- $\omega$: Frequency of the cyclic load
- $\alpha$: Coefficient of thermal expansion of layer $i$
- $1$: Suffix related to the friction layer
- $2$: Suffix related to the metal layer
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<td>Copper-Based Friction Material</td>
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<td>PFM</td>
<td>Paper-Based Friction Material</td>
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<td>TEI</td>
<td>Thermo-elastic Instabilities</td>
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<tr>
<td>TMI</td>
<td>Thermo-mechanical Instabilities</td>
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<td>65Mn</td>
<td>65 Magnesium</td>
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<td>UMAT</td>
<td>Universal Mechanical Tester</td>
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<td>YLL</td>
<td>Year Life Lost</td>
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CHAPTER I

INTRODUCTION

1.1 Overview

Friction materials are a crucial component of sliding mechanisms in various engineering systems such as aircraft brakes, ship clutches, mining machines and military vehicles (e.g. tanks). These materials are carefully designed to transmit or resist motion when in contact with each other. They are commonly used in brake systems, such as brake pads, as well as transmission systems for the design of clutch discs \cite{4, 5}. Other applications include bearings, seals and turning operations. When these materials come into contact and slide against each other, frictional heating occurs, which can increase the temperatures on the contacting surfaces \cite{6, 7, 8, 9, 10, 11}. The amount of heat generated depends on various factors, such as the type of materials in contact, the geometry and surface roughness of the surfaces, sliding speed, and contact pressure.

As the sliding speed increases, the frictional force also increases until it reaches a maximum value, known as the critical sliding speed. Beyond this point, the heat generated may not have enough time to dissipate, leading to an accumulation of heat on the contacting surfaces \cite{12}. Moreover, because the sliding interaction between the friction material and the disc involves two distinct materials, there are oscillations
in the temperature evolution on the contacting surfaces. This variation leads to localized heating and cooling, causing the temperature to fluctuate between high and low values on the disc. Consequently, the disc’s surface experiences expansion and contraction [13, 14]. When this thermal phenomenon is coupled with any form of mechanical stress, it induces non-uniform thermoelastic distortion. This phenomenon is known as "Thermoelastic Instability" (TEI), which was first discovered by Barber in sliding systems [15, 16]. The onset of TEI poses significant constraints on the design of brake and clutch systems. Researchers have been paying attention to the choice of materials to overcome TEI.

To better understand and solve the problem of TEI, researchers have been exploring different physical fields that include not only the traditional thermoelastic instability discovered by Barber but also several distinct physical interactions, such as wear, convective cooling, material nonlinearities, and geometry. These interactions are studied under the umbrella term "thermo-mechanical instabilities" (TMI), which involves the coupling of any of these interactions: wear, thermal buckling, vibration, noise, etc.

In summary, friction materials play a critical role in ensuring the proper functioning of sliding mechanisms, such as brake and clutch systems. The phenomenon of Thermoelastic Instability (TEI) poses a significant challenge to the design of such systems. Researchers are working to better understand the underlying physical interactions and develop new materials and designs to overcome the onset of TEI and other TMI-related instabilities in friction materials.
1.2 Impact of Instabilities

Thermo-mechanical instabilities (TMI) can have detrimental effects on the performance and longevity of friction materials as they lead to material damage and wear. Other damages in the friction pair include hotspots formation, thermal buckling, creep, thermal fatigue, thermal shock, noise, vibrations and judder [17, 6, 18].

![Figure 1.1: Number of registered vehicles in the United States from the year 2010 to 2021](image)

From an environmental perspective, the frictional contact between the sliding pairs generates particles of various sizes leading to pollution [19]. As the number of registered cars in the United States continues to increase, the emission of particulate matter due to wear and the disposal of worn-out friction materials becomes an increasingly pressing issue.

According to the U.S. Department of Transportation Federal Highway Administration, there were approximately 242 million registered vehicles in the
United States in 2010, which had increased to about 276 million by 2020. In 2021, this number was estimated to be about 282 million, including various types of vehicles such as passenger cars, trucks, buses, motorcycles, and others (Figure 1.1). With the estimated number of vehicles registered in 2021, it can be inferred that 1.128 billion disc brakes and 2.256 billion brake pads were utilized in vehicles registered in 2021. Furthermore, over the course of a car’s lifespan, which is approximately 14,912 miles [1, 20], it is necessary to replace the brake discs and pads multiple times in order to maintain optimal performance. According to a study by Gradin and Åström [20], 14 spare parts of brake pads and 2.5 spare parts of discs are needed per disc brake of the vehicle. This implies that a considerable amount of worn-out friction materials will need to be disposed of for vehicles registered in 2021 alone, throughout their lifespan due to TMI.

In Figure 1.2, we can observe a collection of damaged brake discs that are intended for recycling. The key issue with these discs is the environmental pollution that is generated throughout their lifespan. Moreover, brake pads are frequently replaced
due to cracking and substantial wear, leading to the release of a significant amount of particulate matter into the environment caused by TMI.

Figure 1.3: Transition temperature for different testing conditions [2]

1.2.1 Temperature effect on particulate matter

To reduce the emission of particulate matter, it is critical to maintain the system temperature as low as possible since the temperature greatly influences particle emission. Besides, wear typically produces coarse airborne particles that are greatly influenced by temperature, leading to variations in particle size. Shin et al. [21] established that as temperature of the sliding interface increases, ultrafine particles are produced rather than coarse particles. In addition, Alemani [3] observed that the concentration level of these particles increases as temperature increases.

As depicted in Figure 1.3, as temperature increases, the concentration level of particulate matter increases which is dependent on the system parameters, material properties, and applied pressure.
The concentration of the emission can increase by several magnitude orders within the range of 170-200°C. Therefore, it is crucial to recognize the importance of addressing TMI to minimize the environmental impact of friction materials.

1.2.2 Problem with traditional friction materials

Significant efforts have been undertaken to limit the occurrence of excessive wear and tear, caused by TMI, in brake and clutch applications. This has been accomplished by augmenting the thermal and mechanical properties of the sliding components with the addition of metals such as copper in the friction materials.

![Figure 1.4: Years of Life Lost (YLL) in European urban areas due to different PM sources [3]](image)

However, as environmental pollution continues to worsen due to factors like the emission of particulate matter from brake pads containing harmful metals and the recent COVID-19 pandemic, governments worldwide have implemented policies aimed at reducing particulate matter released from brake wear, thereby safeguarding the
environment and human health. The inclusion of copper in friction materials creates a serious problem in that, as the temperature increases on the sliding interface, ultra-fine copper particles are released into the environment due to wear, exacerbating environmental pollution.

Suleiman et al. [22] and Alemani [3] estimated the Years of Life Lost (YLL), a metric for gauging the reduction in life expectancy, attributable to particulate matter emissions. According to their research, the average life expectancy in European urban areas is reduced by 0.63 years per person, with peak reductions of 1.2 Years of Life Lost. Brake particulate matter in Europe is responsible for 0.043 YLL per person on average, which is equivalent to 16 days of life lost, with peak values reaching 0.08, equivalent to 27 days of life lost, as illustrated in Figure 1.4.

1.2.3 Significance of the research

The significance of this work cannot be overstated, as it offers an innovative approach to mitigate thermo-mechanical instabilities that are known to cause thermal cracks, hotspots, thermal buckling, etc. in friction materials. These instabilities have detrimental impacts on the lifespan of friction materials, necessitating frequent replacement of brake pads and discs, leading to high costs. Additionally, the work on metal-free friction materials copper goes to support the Copper-Free Initiative of the United States Environmental Protection Agency, as directed by the National Pollutant Discharge Elimination System (NPDES) [23]. The initiative aims to reduce the amount of copper in brake pads to below 0.5 percent by weight by the year 2025 as particulate matter emitted from friction material during sliding can cause respiratory and cardiovascular problems. More importantly, this research contributes to the reduction of copper’s impact on aquatic organisms.
As brake pads gradually deteriorate, the brake dust containing copper and various other metals accumulate on road surfaces, eventually being carried into streams and rivers by erosion agents.

The toxicity of copper poses a significant threat to fish and other species residing in these aquatic environments. It hampers their olfactory senses, rendering them more susceptible to predators and impeding their ability to navigate back to their spawning streams. Further, the research helps to mitigate wear dust pollution into the air which can cause respiratory and cardiovascular diseases to the public. The environmental impact of particulate matter from metals such as copper can result in air pollution leading to acid rain and global warming, among other environmental issues (Environmental Impact of Disc Brakes Life Cycle 2020).

The work examines the performance of metallic, non-metallic, and viscoelastic friction materials when utilized in brake and clutch systems in order to minimize the onset of TMI. By minimizing thermo-mechanical instabilities, the rate of brake pad and disc replacement can be reduced, leading to cost savings. Furthermore, mitigating the release of particulate matter from friction materials will reduce the respiratory and cardiovascular problems associated with such emissions, leading to improved public health. Additionally, the reduction of air pollution will minimize environmental phenomena such as acid rain and global warming caused by the emission of particulate matter from sliding. Finally, by reducing the disposal of damaged pads and discs, water, and soil pollution caused by the emission of particulate matter from braking can be minimized.
In summary, this work provides an innovative and sustainable approach to addressing the challenges posed by thermo-mechanical instabilities in friction materials. The benefits of this approach include cost savings, improved public health, and reduced environmental impacts.

1.3 Methods of estimating instabilities

1.3.1 Theoretical and experimental approach

Theoretical and experimental studies have been conducted to address the occurrence of Thermo-Mechanical Instability (TMI) in sliding systems. As previously mentioned, Barber first discovered this phenomenon. Parker and Marshall [24] further examined it during a railway test, where they observed temperature concentration in a particular localized area of their sliding system. TMI is a contact problem that arises when the sliding speed of the system is sufficiently high, causing temperature localization and resulting in material damage, wear, noise, and vibrations. Barber's discovery of instabilities in sliding systems has been the subject of theoretical and experimental research by many scholars.

1.3.1.1 Stability analysis

Several researchers have employed various theoretical approaches to predict thermo-mechanical instabilities and one of the commonly used techniques is the perturbation method. Dow and Burton [25] did extensive work by utilizing this approach by demonstrating that the contact between two half-planes would become unstable if the sliding speed exceeds a critical value. The instability behavior observed strongly relies on the wavelength of the applied perturbation.

Moreover, Burton et al. [26] work revealed that when one of the friction pairs has significantly better conductivity than the other, the perturbation becomes nearly
stationary in the good conductor, resulting in a low critical speed. However, when the friction pairs have comparable conductivities, the perturbation moves with respect to both bodies, leading to a higher critical speed. Meanwhile, the conclusion from experiments conducted by Berry and Barber’s [27], and Barber [15, 16] were at variance with Burton's results. It is worth noting, however, that the critical speed values obtained using the perturbation method may underestimate the experimentally derived values for practical applications, as reported by Anderson and Knapp [6]. Despite this limitation, the perturbation method remains a valuable tool for predicting TMI behavior.

In addition to Burton's work, further research has been carried out on this topic. One example is the study by Barber et al. [28], which aimed to explore the stability of different steady-states by investigating the circumstances in which a small imposed perturbation can exhibit exponential growth over time. The results indicated that such perturbations can only occur for certain eigenvalues of the exponential growth rate, and stability is ensured only when these eigenvalues are negative. Several researchers [29, 4, 30] have conducted a similar study on the non-linear problem of a thermoelastic instability by utilizing the perturbation approach by focusing on stability and steady-state solution to predict the system behavior.

1.3.1.2 Steady state problem

In stability analysis, the imposed initial perturbation will grow exponentially with time when the system becomes unstable. Hence, it is highly anticipated that the system will tend to a steady state if there is no wear on the contacting surfaces. Several steady-state solutions have been established for two -to three-dimensional sliding systems. Barber [31] introduced harmonic potential function into the
general solution for steady-state problems and verified his solution by solving the axisymmetric problem.

Further, Al-Shabibi [32] used a transient solution to solve the nonlinear thermoelastic instability problem in two contacting surfaces. The author obtained the transient evolution of temperature field for a linear contact problem with frictional heating and constant sliding speed by superimposing the solution of the perturbation problem to the steady state solution. It was observed that when the sliding speed is below the critical value, there is moderate reduction in the contact area and the system takes relatively longer to reach a steady state. Also, a transient finite element simulation was conducted by Zagrodzki et al. [33] to investigate the contact problem of a stationary layer between two sliding layers with frictional heating by considering a steady-state case. It was discovered that when separation occurs, there is a non-monotonic transition to a steady state with the contact region separated by the same wavelength.

1.3.1.3 Transient problem

In practical applications, the engagement process of clutches and brakes occurs quickly under normal conditions. Therefore, using a transient approach provides a better representation of real-world conditions than a steady-state approach. It is worth noting that in order to achieve a steady state, the process must first go through a transient state. However, solving the transient thermoelastic contact problems can be difficult and complex, as shown in the study conducted by Azarkhin and Barber [34]. The solution used the transient Green’s function for temperature and thermoelastic displacement given by Barber and Martin-Moran [35], but the algorithm posed some difficulties because bifurcation occurred when a sufficiently large ratio of the initial width of the contact area to that in the steady state was used.
As a result of this complexity, simulating the contact problem in the time domain using a finite element method is the preferred method.

Additionally, Kennedy and Ling [36] used a finite element method to simulate thermoelastic instability and reported that transient changes occurred at the contact of the friction surface. To account for the effect of wear at the contact interface, a wear criterion was proposed to predict wear rates for disk brakes, which produced results that were found to be close to experimental data. Zagrodzki [37] used a transient approach to reveal the mechanisms of excitation of unstable modes during a modal analysis. Also, Zagrodzki et al. [38] conducted an analysis of the non-linear transient behavior of a sliding system for a two-dimensional thermoelastic contact problem. The study revealed that, following separation, there is a transition from the transient to a steady state where the contact region is separated by the same wavelength. Likewise, Azarkhin and Barber [39] investigated the time-dependent problem of a non-conducting half plane sliding against a conducting material using a transient approach. Li and Barber (2008) utilized an expression of temperature and stress fields to solve a transient thermoelastic contact problem based on a fast-speed expansion method.

Several researchers have employed a finite approach to deal with transient problems, and numerical methods have proven effective in handling complicated practical geometries, as demonstrated by Sonn et al. [40]. Yi et al. [41] implemented Burton’s method numerically for the limiting case where the friction pads are non-conductors and found that Lee and Barber [42] idealization provides a good estimate of the more exact behavior in most cases when examining more realistic descriptions of the disk brake geometry. However, the numerical approach requires a significant amount of effort and is computationally intensive, as the solution of
the transient heat conduction equation with a very fine mesh requires small time increments for stability and convergence.

1.3.1.4 Finite element approach

The finite element method is commonly used to investigate transient problems, particularly for complex three-dimensional geometries, as demonstrated by Yeo and Barber [43]. Their algorithm was based on the linearity of the system, including all boundary conditions, to the perturbation from the steady-state and the existence of a serration variable. In contrast to purely analytical methods, which are limited to certain geometric length scales, the finite element method is more versatile.

Du et al. [44] also employed the finite element method to study sliding between a thermally conducting body and a non-conducting body, where the instability was governed by a real eigenvalue and the stability boundary was determined by the presence of a zero eigenvalue. Using finite element analysis, Du obtained a matrix relation between the heat source $q$ at the contact nodes and the nodal contact force $P$, given by the heat balance equation $q = fVP$, where $f$ is the frequency coefficient and $V$ is the sliding speed. This approach has been used by Zagrodzki [45], Azarkhin and Barber [39], Yi et al. [46] and other researchers in solving thermoelastic contact problems with transient analysis.

1.3.1.5 Experimental determination

Several experiments have been conducted to observe the phenomenon of thermoelastic instability using various equipment. Strain gauges, thermocouples, and infrared can provide a rough estimate of instabilities such as stress, hotspots, wear, thermal cracks, and temperature. However, for accurate and detailed information, a well-defined setup is required.
The first experimental approach to observe the existence of local heating in railway brake tests was carried out by Parker and Marshall [24]. However, the mechanism of TEI was not fully understood until Barber [16] conducted a detailed investigation using both experimental and theoretical methods. It was discovered that thermal deformation causes the contact geometry to change, and when the effect of thermal deformation exceeds wear, the contact area becomes unstable. Since then, the phenomenon has been experimented in various applications, particularly in railway and automotive brakes [47, 48]. Lee and Barber [49] conducted a bench experiment to examine the temperature variation during sliding interaction in disk brakes.

In a similar vein, Abbasi et al. [50] carried out a pin-on-disc test to study the thermomechanical interaction of railway braking materials and a numerical model was developed to compare the results with the experiment. The results were found to be consistent with each other. Additionally, Jiaxin Zhao [51] conducted a bench experiment to investigate the friction coefficient of a wet clutch by taking into account the rotational speed and torque. The authors observed that when the fluid pressure increases, the friction coefficient changes due to the variation of the real contact area of the sliding pairs.

1.4 Overview of dissertation

Chapter II

In this chapter, a Cu-based friction material (CFM) was manufactured through powdered metallurgy techniques to understand some of the instabilities associated with metallic friction materials when used in dry clutch systems. The CFM was subjected to testing to determine its ability to withstand thermo-mechanical instabilities under various conditions. To evaluate the material’s frictional properties and wear rate, it was slid against 65Mn steel using a Universal Mechanical Tester.
under two operating variables: sliding speed and temperature. The effect of these variables during the engagement process of the friction pairs was examined.

By knowing the normal applied force and the dimensions of the clutch disc, the dynamic friction coefficient was converted into the friction torque capacity over time. The impact of operating temperatures and sliding speeds on the thermal buckling and thermoelastic instability of the friction disc were also investigated. The study demonstrates how temperature and sliding speed influence instabilities in metallic friction materials.

Chapter III

For this chapter, a comprehensive two-dimensional non-linear transient thermo-mechanical model was developed using a finite element approach to investigate a novel material with anisotropic properties. This material is free from metals and is made to slide against a steel disc. The study aims to investigate the impact of the friction material’s properties on thermo-mechanical instability when used in automotive disc clutches or brakes. The validity of the model was established by comparing its results with an existing eigenvalue approach.

A parametric study on material anisotropy was carried out, which revealed some interesting findings. The findings of this study provide critical insights into the behavior of metal-free anisotropic friction materials under various conditions, which can contribute to the development of more reliable and efficient automotive disc clutches or brake friction components in the future.

Chapter IV

The research carried out in this chapter considers a mathematical model to assist in forecasting the emergence of thermo-mechanical instabilities in disc brakes and
clutches while utilizing viscoelastic friction materials, as this type of material can be used to eliminate noise, vibration, and squeal in sliding systems.

The proposed model is based on Burton’s fundamental theory for thermoelastic instabilities in pure elastic materials but extends to include three physical material parameters: relaxation time, elasticity, and thermal conductivity.

Prior to this study, the impact of these material properties on thermoelastic instabilities had not been fully grasped. Therefore, a finite element analysis is performed to confirm the mathematical model by evaluating the variation of the critical speed as a function of thermal conductivity. This research is pivotal in comprehending the influence of viscoelastic parameters in sliding systems and offers an intuitive method for predicting the onset of thermo-mechanical instability. Of greater significance, the work here presents researchers with a substitute material that can be utilized in clutch and brake pads, along with methods for enhancing its suitability.

Chapter V

A 7-month industrial training was conducted at CalmCar Vision System LLL to gain practical insights and explore different methods of encountering thermo-mechanical instabilities in electric/autonomous vehicles. The training also emphasized automatic parking and various scenarios utilizing a range of sensors and electrical components. This chapter provides a concise overview of the work conducted during the industrial training.

Chapter VI

After analyzing the results obtained from each individual task, conclusions are drawn and potential areas for future work are suggested.
SECTION II

INSTABILITIES IN METALLIC FRICTION MATERIAL

2.1 Introduction

Friction material formulations incorporate sintered-metal particles to provide unique mechanical and thermal properties during sliding interactions. These particles, which may contain copper or iron, are mixed with other friction agents such as wear-resistant agents, resins, and fillers to enhance friction characteristics and lubrication to improve sliding performance [52]. Adding sintered metals to friction pairs help control the onset of instabilities. However, the negative environmental impact of metal particles has led to a need for more eco-friendly materials. Additionally, several States within the United States have enacted laws restricting the use of these metals in brake pad formulations. For instance, the Washington Better Brake Law legislation and the California Motor Vehicle Brake Friction Material Law prohibit the use and sale of brake pads containing more than trace amounts of copper and other metal particles in the States. The goal is to decrease the release of copper and other metal particles into water bodies due to wear.

While the primary objective of the entire research is to investigate and develop strategies for reducing instabilities in next-generation friction materials, an insight
into instabilities in traditional materials such as metallic-based friction materials is necessary to better formulate effective approaches to control instabilities in next-generation friction materials. Moreover, metallic-based friction materials are still in use despite their environmental related problems. As a result, this work not only aims to address instabilities in next-generation metal-free friction materials but also offers ways to mitigate instabilities associated with metal-based friction materials.

2.1.1 Friction pairs

Friction pairs are crucial components in sliding systems, such as brakes and clutches, as they provide sufficient friction torque to transmit power or decelerate sliding parts. In most contact interactions, minimizing friction between the friction pairs is crucial to avoid excessive wear and energy loss. However, clutches require high and consistent dynamic friction to transmit a continuous but smooth torque [53]. As a result, manufacturers of automatic transmissions and researchers have devoted extensive efforts to enhancing friction pairs to improve driving comfort, performance, and fuel efficiency.

Today, composite materials are replacing conventional friction materials to achieve greater efficiency [54]. For clutch applications, Cu-based and paper-based composite friction materials are widely utilized to specify the contact interaction between the engine and the transmission system, owing to their distinctive thermal and mechanical characteristics. Research conducted by Yu et al. [55] indicates that copper-based friction material exhibits significantly higher friction and wear characteristics during sliding interactions than paper-based friction material.
2.1.2 Paper-based friction material (PFM)

Paper-based composite materials, whose matrix primarily comprises binders, fillers, fibers, and friction modifiers [56, 57, 58, 59], have become popular in the automobile industry for designing clutch friction discs for automatic transmission systems [53]. However, when paper-based friction materials are used in dry clutches, they undergo wear, distortion, and high temperatures, resulting in excessive noise, vibration, and poor friction behavior. [57, 60].

Figure 2.1: Surface of paper-based friction disc before and after sliding

Figure 2.1 depicts the surface profile of a dry friction disc before and after multiple clutch engagements. The post-engagement surface of the paper-based friction disc shows burns and a darkened appearance due to high-temperature generation and the influence of high operating conditions such as high thermal loading. These factors can affect the dynamic friction behavior and wear rate of the friction material, resulting in poor torque transmission and noise during clutch engagement [58]. Therefore, despite the presence of metals in paper-based friction materials, they are still susceptible to exhibiting instabilities.
The focus of this study was primarily on copper-based friction material, which contains a high amount of copper as a metallic material to understand the role of such metal in the onset of thermo-mechanical instability.

2.1.3 Copper-based friction material (CFM)

Cu-based composites have been observed to exhibit good thermal behavior and friction torque in dry clutches, despite their negative impact on the environment [61]. The dynamic behavior of Cu-based friction material is more evident in a study conducted by Jang et al. [62]. Further, research conducted by Kwabena Gyimah et. al [63] indicated that Cu-based composites’ friction and wear properties can be improved through powdered metallurgy techniques.

The friction torque of Cu-based composites during clutch engagements depends not only on the material composition or preparation methods used [64, 65, 66], but also on factors such as temperature, sliding speed, and normal load [67, 68]. However, research has shown that the normal load may not significantly influence clutch frictional characteristics [69, 70], except in some cases where a slight increase in friction coefficient may be observed. Friction generally decreases with increasing applied loads during sliding contact, as indicated by numerous research studies [59, 71, 72, 73]. The primary reason for this decrease in friction with increased normal load is the temperature generated at the contacting surface. For this reason, temperature and sliding speed were the main considerations in this work.

For this study, the Cu-based composite matrix was prepared based on previous research by Wei & Chen [74] on different compositions of Cu-based friction material. We used graphite with a size of 300~600µm at a content of 10wt%, while the friction component of the matrix Al₂O₃ was increased to 4wt% from the 3wt% used by Wei & Chen [74] to provide more stable and higher friction properties [75, 76].
The choice of 65Mn steel was made due to its ability to resist wear. In addition, the presence of manganese improves both the hardness and hot working ability of the material. During sliding contact between friction pairs, the occurrence of thermoelastic instabilities (TEI) and thermal buckling is expected when the temperature or operating speed exceeds critical values [37]. This is due to the non-uniform pressure distribution across the surface of the friction plate, which results in localized high temperatures on certain areas of the friction plate. Such phenomena lead to the formation of hotspots on the clutch disc and eventually reduce its lifespan.

2.1.3.1 Engagement behavior of CFM

The investigation of the engagement behavior of Cu-based friction material in clutches is crucial for improving gear change smoothness, maximizing torque transmission, and enhancing overall life and comfort in automatic transmission systems. To achieve these goals, it is essential to understand the relationship between friction, torque, and operational variables during an engagement. The performance of the clutch during engagement is influenced by various factors, including operating conditions and design parameters. While some of these effects can be predicted, others, such as the impact of sliding speed and corresponding torque on friction and thermoelastic instability, are more challenging to anticipate. As a result, researchers often conduct a series of experiments to better understand the behavior of friction pairs before proceeding to the development stage. In most cases, experiments on clutch friction characteristics are performed using the SAE#2 machine, which operates as an inertial dynamometer. However, this apparatus only takes into account rotating inertia as an input parameter.
Using a Universal Mechanical Tester, the friction characteristics of a dry clutch friction material during an engagement process are obtained in this study. The resulting friction characteristics are then converted into the torque friction capacity produced during the engagement process. Several methods have been employed to estimate the friction torque of the clutch. For example, a model for calculating the friction torque of a dry clutch was developed by Lin et al. [77], and it was verified by bench tests, demonstrating accurate results. Additionally, Pica et al. [78] created a torque model with temperature and slip speed for a dry clutch, and their findings showed that temperature changes can determine the degradation of clutch engagement characteristics. Furthermore, the change in friction coefficient during sliding can impact various operating parameters, such as normal force and sliding speed [79].

2.1.3.2 Instabilities in Cu-based friction pairs

Abdullah and Schlattmann [80] investigated the transient thermal behavior of an axisymmetric clutch friction disc. They observed that during the engagement process, the temperature was lowest at the inner radius and highest at the outer radius. This temperature difference was due to the increase in velocity along the radial to the circumferential region. The amount of heat generated during the sliding of friction pairs depends on the type of friction pairs involved. Ali et al. [54] studied the effect of transient thermal load on dry friction clutch and reported that the material composition used for the friction material affects the life span and stability of the clutch disc. Additionally, non-uniform temperature distribution during sliding contact can cause excessive heat to promote thermoelastic instability, as demonstrated by Chen et al. [81] and Yi et al. [46].
It is worth noting that the existing literature mainly covers the chemical compositions, wear, and friction characteristics of Cu-based composites, but there is little work that relates the tribological experimental results to the onset of thermoelastic instability and thermal buckling in real clutches. A comprehensive understanding of thermoelastic instability during sliding is crucial to assess the lifespan and performance of a clutch. Thus, this study aims to expand the tribological investigation conducted by Wei and Chen [74] to include thermoelastic instability (TEI) and thermal buckling. The study investigates the impact of sliding speed and temperature on the friction torque capacity of the friction disc, wear rate, and their contribution to thermal buckling and TEI during the engagement process.

Chen et al. [81] conducted a numerical investigation on the interplay between thermal buckling and thermoelastic instability and found that temperature non-uniformity caused by the latter could influence the temperature distribution in the former. At high sliding speeds, the resulting temperatures may become concentrated at certain areas on the clutch disc, forming hotspots [6, 82]. Thermoelastic instability-induced thermal buckling is responsible for the formation of these hotspots [83, 46]. Additionally, the onset of thermoelastic instability in sliding systems has been observed to be marked by anti-symmetric perturbation. Therefore, the friction properties, wear rate, and thermal behavior of the friction disc are crucial factors to consider during the engagement process. This study begins by investigating the friction characteristics of the friction pairs, which is fundamental in the analysis of thermoelastic instability.

The friction material used in this study was fabricated using powder metallurgy techniques and the constituents listed in Table 2.1. To improve the friction behavior
of the Cu-based composite, a graphite content of 10wt% and a size range of \(300 \sim 600 \mu m\) were added, as recommended by Wei & Chen [74].

2.2 Experimental setup and procedures

2.2.1 Sample preparation

The constituents were weighed according to their chemical compositions and mixed thoroughly in an electric blender at 200rpm for 6 hours. The resulting mixture was then compacted at a pressure of 800MPa into circular shapes with a thickness of 6mm and a diameter of 68mm using a compacting machine.

Table 2.1: Composition of Cu-based composite (wt%)

<table>
<thead>
<tr>
<th>Cu</th>
<th>Fe</th>
<th>Sn</th>
<th>Cr</th>
<th>Al$_2$O$_2$</th>
<th>ZrSiO$_4$</th>
<th>Graphite</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>4</td>
<td>7</td>
<td>2</td>
<td>4</td>
<td>3</td>
<td>10</td>
</tr>
</tbody>
</table>

The shapes were sintered in a furnace tube at a temperature range of \(800^\circ C\) to \(900^\circ C\) in a controlled atmospheric pressure of 0.01MPa for 30 minutes. The high carbon steel 65Mn (pin), with a cylindrical shape having a diameter and height of 6mm and 18mm, respectively, was industrially prepared according to the composition presented in Table 2.2. Ultrasound hardness testing devices were used to conduct hardness tests on the samples, with five tests conducted for each sample to obtain an average hardness.

Table 2.2: Chemical composition of 65Mn steel

<table>
<thead>
<tr>
<th>Element</th>
<th>C</th>
<th>Si</th>
<th>Mn</th>
<th>S</th>
<th>P</th>
<th>Cr</th>
<th>Ni</th>
<th>Cu</th>
</tr>
</thead>
<tbody>
<tr>
<td>Composition</td>
<td>0.62~0.70</td>
<td>0.17~0.37</td>
<td>0.62~0.70</td>
<td>(\leq 0.036)</td>
<td>(\leq 0.036)</td>
<td>(\leq 0.25)</td>
<td>(\leq 0.25)</td>
<td>(\leq 0.25)</td>
</tr>
</tbody>
</table>
The purpose of the experiment was to study the friction characteristic under different operating conditions and its related instabilities such as wear, cracks, etc. when used in clutch applications. The torque friction characteristics for the clutch disc (Cu-based) and the separator (65Mn) were obtained using a Universal Mechanical Tester (UMT-5) as shown in Figure 2.2.

![Experimental set-up procedures](image)

**Figure 2.2: Experimental set-up procedures**

2.2.2 Description & operation of the UMT-5 Machine

With a speed range of approximately 0-5000 r/min, the UMT-5 can adjust the load sensor within a range of 0-2000 N and achieve a maximum heating of 1000°C in the rotary chamber. The sliding track on the friction disc was configured with a rotation radius of 25 mm, while the 65Mn steel pin was secured on the 2-D sensor module. Center bolts were used to attach the friction disc to the rotary chamber.
The experiment followed these steps:

1. The UMT-5 machine was set up and connected to a computer.
2. The two friction pairs were cleaned and installed onto the UMT-5 machine.
3. The necessary speed, temperature, load, and duration of the test were established.
4. The friction disc was heated and set to rotate at the specified speed and temperature, and the 65Mn steel pin was then applied to the rotating disc at the desired load.
5. Each measured parameter was tested for a duration of 5 minutes, after which the friction pairs were separated and cleaned.
6. The dynamic friction coefficient output was recorded and saved on the computer.

2.2.3 Experimental operating conditions

The experiment was conducted under high operating conditions and in the absence of lubrication. In order to relate the findings to the actual engagement process of a clutch, the following test parameters were examined: The rotations per minute (rpm) were set at 1, 10, 100, 500, 1000, 1500, 1500, and 2000, while maintaining a constant normal applied load of 600 N and a temperature of 175°C. This variation was performed to assess the impact of the resulting friction coefficient on clutch friction torque. Additionally, the normal load and speed remained consistent at 600 N and 500 rpm, respectively, while the temperature was modified at 25, 100, 175, 325, and 400 degrees to determine its effect.

2.2.4 Surface morphology and wear rate determination

Due to the increased production of unpleasant-smelling gases during a preliminary test at higher temperatures, the surface morphology at various temperatures was examined to gain a better understanding of the surface behavior.
This was necessary as the presence of these gases indicated material decomposition due to heat. Additionally, the wear rates of the friction disc were studied by measuring the mass before and after each test under the varied parameters [84, 85]. The wear rate was calculated from the pin on the disc test and discussed in the context of the experiment. Equation 2.1 was utilized to compute the wear rate.

\[
W = \frac{w_1 - w_2}{\rho d_s F_N}
\]  

(2.1)

Where \( w_1 \) is the initial mass of the sample (\( kg \)), \( w_2 \) is the mass of the sample after the test (\( kg \)), \( \rho \) is the density of the specimen \( kg/m^3 \), \( d_s \) is the sliding distance (\( m \)), \( F_N \) is the normal applied load (\( N \)).

2.2.5 Determination of torque-friction capacity

The assumption made in this study is that if the same sliding conditions are employed in a real clutch for the two sliding pairs utilized in the experiment, the resulting friction coefficient will be relatively similar with slight variations. This assumption enables us to relate the experiment results to real clutch applications. By utilizing Equation 2.2, the measured friction coefficient can be converted into torque friction capacity generated during the clutch engagement, provided the dimensions of the clutch disc and applied normal force are known.

\[
T_c(t) = \frac{2}{3} \mu(t) \frac{R_o^3 - R_i^3}{R_o^2 - R_i^2} F_a
\]  

(2.2)

Where \( F_a \) (\( N \)) is the normal force applied to the clutch disc, \( \mu(t) \) is the friction coefficient with time, \( R_o \) (\( m \)), and \( R_i \) (\( m \)) are the inner and outer radius of the friction disc surface.
The applied normal force $F_a$ was 600N and the inner and outer radius of the friction pairs were 0.062m and 0.087m respectively. During the sliding of the friction pairs, kinetic energy is converted into heat energy. The heat produced at any time during sliding on the clutch surface is given as:

$$q(r, t) = \gamma \mu Pr\omega_0$$

(2.3)

Where $\gamma$ is the heat partition coefficient, $\mu$ is the friction coefficient, $P$ is the applied pressure, $r$ is the radius of the disc, and $\omega_0$ is the initial angular velocity.

Figure 2.3: Friction torque capacity at different temperatures vs. time
2.2.6 Friction torque capacity

2.3 Experimental results

The presented graph in Figure 2.3 illustrates the friction torque capacity between the friction disc and the separator under constant applied normal load and sliding speed of 600N and 500rpm, respectively, during a 3-minute engagement period at various temperatures. The data shows that temperature has a noteworthy impact on the friction torque capacity. At 25°C, the observed friction torque capacity exhibits a fluctuating pattern over time, which suggests a strong interaction between the two friction pairs influenced by the friction coefficient. This occurrence may result in clutch juddering during the engagement process, which can negatively affect driving comfort and is therefore not recommended. However, as the temperature rises from
25°C to 175°C, the friction torque capacity notably increases from 32 $Nm$ to 37 $Nm$.

2.3.1 Surface morphology and wear rate at varied temperature

The surface morphology analysis in Figure 2.4 showed that at 175°C, the surface roughness increased due to wear, contributing to the higher friction value observed in Figure 2.5. However, at temperatures above 175°C, the torque friction capacity (Figure 2.3) or friction values decreased (Figure 2.5) significantly. This could be due to increased degradation of the material composition on the friction plate surface, which affected the generated friction coefficient.

The sliding contact surface of the Cu-based composite degraded significantly as the temperature increased further. This is because the strength of the Cu-based composite matrix is reduced at higher temperatures, leading to the formation of scratches on the surface at 25°C and large grooves at 325°C, which eventually resulted in high delamination of the worn surface at 400°C.

![Graph of Friction Coefficient vs. Temperature](image)

*Figure 2.5: Friction coefficient vs. temperature*
The rough surface, scratches, and delamination observed on the surfaces of the Cu-based composite friction material had a significant impact on noise generation during sliding, making it unstable and high. Additionally, the wear rate of the Cu-based friction material increased with the rise of temperature as depicted in Figure 2.6. The wear rate at 25°C was observed to be $9.3 \times 10^{-16} m^3/Nm$, while at 400°C, it was $2.3 \times 10^{-16} m^3/Nm$. The increase in wear rate with the temperature rise is attributed to surface wear caused by material deterioration, as shown in Figure 2.4.

Overall, the results show that a more stable friction torque capacity with time was observed at temperatures of 100°C, 175°C, and 400°C, but the friction coefficient decreases as temperatures increase beyond 175°C, as shown in Figure 2.5. Thus, we can conclude that as temperature increases, the rate at which torque is transmitted from the engine to the transmission system is minimized between the two friction pairs.

Figure 2.6: Wear rate at varying temperatures
However, at an extreme temperature of 400°C, the torque capacity generated is unstable with time. This is due to the high possibility of the operating temperatures exceeding critical values, which may excite the axisymmetric mode and make the system unstable. Additionally, surface deterioration, as seen in the surface morphology study, may occur, which can influence the friction values between the sliding pairs and explain the observed friction torque at 400°C.

2.3.2 Friction torque and wear rate at varied sliding speeds

The torque friction capacity with time for different sliding speeds is presented in Figure 2.8. At 1rpm, unstable torque friction capacity was observed, which is similar to what was observed when the temperature was varied at 25°C (Fig. 2.3). This can be attributed to the vigorous interaction between the surface of the friction pairs at very low operating conditions, which led to vibration as reported in previous studies [86].

Figure 2.7: Friction coefficient vs. sliding speed
Fluctuation in the friction coefficient may contribute to this phenomenon, resulting in clutch juddering during torque transmission [87, 88, 89]. One way to mitigate torque fluctuation and improve low-speed performance of the transmission system is to reduce the friction coefficient [79]. However, the friction values exhibited at 25°C and 1rpm were above 0.35, which is significantly high (Figures 2.4 & 2.8 respectively), hence the expected fluctuation in torque.

Furthermore, when the sliding speed exceeds the critical speed, the entire clutch system may become unstable, leading to torque fluctuations. This behavior is investigated in a later section. Above a sliding speed of 500rpm, the torque friction capacity decreases from 37Nm to 35Nm (See Figure 2.8) due to the reduction in friction value (Figure 2.7). This observed phenomenon has a significant impact on clutch power output during an engagement.

![Figure 2.8: Friction coefficient at different sliding speeds vs. time](image)
Figure 2.9: Wear rate at varying sliding speeds

Figure 2.9 illustrates that the wear rate increased considerably from $1.42 \times 10^{-15} m^3/Nm$ to $2.71 \times 10^{-15} m^3/Nm$ when the sliding speed increased from 10rpm to 100rpm. This can be attributed to the influence of surface temperature, which rose sharply due to the sudden increase in speed, resulting in more material removal. This explains the high friction coefficient observed at 100rpm (Figure 2.7) due to the rough surface produced by the material removal. The highest wear rate ($4.01 \times 10^{-15} m^3/Nm$) occurred at a sliding speed of 2000rpm.

In general, a more stable friction torque capacity was obtained but with time fluctuations when varying the sliding speeds. To gain a better understanding of this phenomenon and its impact on clutch lifespan, buckling analysis and thermoelastic analysis were carried out numerically to determine the critical temperature and speed at which the system becomes unstable, leading to fluctuations in torque friction capacity.
2.3.2.1 Temperature effect on the Cu-based friction material

Thermal buckling is not solely affected by friction coefficient values, but also by temperature distributions. In this study, we conducted a thermal buckling analysis to investigate how different temperatures affect the Cu-based friction disc and to relate our findings to the operation of a real clutch. We assumed that the real clutch operates under the same conditions as those in our experiment, taking into account its thermo-mechanical properties. The temperature distribution profile we used for the friction disc assumed a monotonically increasing temperature from the inner to the outer radius, as described by Equation 2.4.

\[ T = \Delta T \left( \frac{r - R_i}{R_o - R_i} \right) \]  \hspace{1cm} (2.4)

where \( \Delta T \) is the temperature gradient, \( r \) is the changing radius along the radial of the friction disc, \( R_i \) is the inner radius and \( R_o \) is the outer radius. Figure 2.10 shows the FEA meshed model of the friction disc used in the buckling analysis. The thermo-mechanical properties of the Cu-based friction material and the separator (65Mn) are shown in Table 2.3.

A numerical analysis using ABAQUS software was performed and the results were validated using an analytical method, as documented by J. Zhao et al. [90]. The computed eigenvalues and critical buckling temperatures are presented in Table 2.4. The first buckling mode at different temperatures is shown in Figure 2.11.
Figure 2.10: FEA meshed model of the friction disc

Table 2.3: Thermo-mechanical properties of Cu-based friction material used in the simulation

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Clutch disc (Cu-based)</th>
<th>Separator (65Mn)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic modulus, $E$ (GPa)</td>
<td>211</td>
<td>100</td>
</tr>
<tr>
<td>Poison’s ratio $\nu$</td>
<td>0.288</td>
<td>0.25</td>
</tr>
<tr>
<td>Thermal conductivity, $(W/(mK))$</td>
<td>48</td>
<td>0.5</td>
</tr>
<tr>
<td>Specific heat, $C$ $(J/\text{kg}K^{-1})$</td>
<td>450</td>
<td>1000</td>
</tr>
<tr>
<td>Mass density, $\rho$ $(kg/m^3)$</td>
<td>7820</td>
<td>1400</td>
</tr>
</tbody>
</table>
Figure 2.11: First buckling mode a) indicating the number of hotspots b) the wave formation of the disc

Table 2.4: Critical buckling temperature computed by the Finite element method and Analytical approach

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Buckling temperature</th>
<th>Eigenvalue</th>
<th>Eigenvalue (analytical)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>1780.3</td>
<td>71.21</td>
<td>72.30</td>
</tr>
<tr>
<td>100</td>
<td>1767.0</td>
<td>17.67</td>
<td>10.20</td>
</tr>
<tr>
<td>175</td>
<td>1771.0</td>
<td>10.12</td>
<td>10.20</td>
</tr>
<tr>
<td>250</td>
<td>1770.0</td>
<td>7.08</td>
<td>7.19</td>
</tr>
<tr>
<td>325</td>
<td>1764.8</td>
<td>5.43</td>
<td>5.50</td>
</tr>
<tr>
<td>400</td>
<td>1768.0</td>
<td>4.42</td>
<td>4.45</td>
</tr>
</tbody>
</table>

It was observed that the buckling mode had a wavenumber of n=14 along the circumferential direction for all tested temperatures, suggesting the possible formation of hotspots on the disc. However, their corresponding eigenvalues were distinct, as displayed in Table 2.4. The axisymmetric mode was not stimulated at these
temperatures since the critical buckling temperatures did not surpass the operating temperatures. Nevertheless, it was apparent that hotspots could develop, resulting in the onset of thermoelastic instability. This observed behavior is considerably influenced by the temperature distribution across the disc during the engagement process, indicating that temperature fields rather than friction coefficient values directly affect thermal buckling.

The critical speed of the experiment was analyzed in relation to the clutch application using the HotSpotter software, which was developed by the University of Michigan. In this analysis, the friction coefficient obtained from the experiment and other properties of the friction pairs were used. The setup model used in the HotSpotter software is shown in Figure 2.12. The stator was set as the separator, and the rotor was the friction disc. The objective was to determine the critical sliding speed at which the system becomes unstable and to examine the growth rate of hotspots at various sliding speeds.

2.3.2.2 Sliding effect on thermoelastic instabilities

Using the HotSpotter software developed by the University of Michigan, an investigation was conducted on the correlation between the critical sliding speed of the Cu-based friction disc and the 65Mn steel separator and the formation of hotspots. The analysis integrated the friction coefficient obtained from the experiment and other pertinent properties of the friction pairs. Figure 2.12 displays the model used in the software setup. The critical sliding speed at which the system becomes unstable and the growth rate of hotspots at different sliding speeds were determined.

The results revealed that the critical sliding speed of the Cu-based friction disc and the 65Mn steel separator was approximately 200rpm, indicating that the system becomes unstable beyond this speed.
This instability could have contributed to the torque capacity fluctuations and vibrations observed during the sliding contact. Four (4) hotspots on the Cu-based friction disc necessitated a critical sliding speed of about 1000rpm, while fourteen (14) hotspots required a critical speed slightly above 4000rpm. Nonetheless, no clear linear association was observed between the formation of hotspots and the critical sliding speed.

The computation of the growth rate of hotspots at various sliding speeds was based on the friction coefficient obtained from the experiment, elastic modulus, thermal conductivity, and other relevant properties.
Figure 2.13: Critical sliding speed vs. the number of hotspots

Figure 2.14: Growth rate of hotspots
As demonstrated in Figure 2.14, the growth rate of hotspots was either zero or exhibited small negative values at sliding speeds of 500rpm and below. This can be attributed to the fact that the critical sliding speed, as illustrated in Fig. 2.13, was determined to be 200rpm. Thus thermoelastic instability is not excited below this speed. The negative growth rate of hotspots at 1rpm, 10rpm, and 100rpm can also be ascribed to this reason.

It is intriguing that the growth rate was zero at 500rpm, despite being above the critical sliding speed. This is because, at a zero growth rate, the system hovers between stability and instability, and any slight influence can trigger either a positive or negative growth rate.

Furthermore, the maximum growth rate of hotspot formation on the Cu-based clutch plate for sliding speeds greater than 500rpm happened when there were six hotspots. Thus, surpassing a certain sliding speed does not automatically mean an increase in the growth rate. The relationship between the growth rate and the number of hotspots is non-linear and relies heavily on the sliding speed. For instance, for the formation of six hotspots on the disc, the growth rate at a sliding speed of 1000rpm was 2.15, whereas at 2500rpm, it was 10.35. This is because, at higher sliding speeds, there is an increase in temperature formation, which stimulates thermoelastic instability. Below 500rpm, the growth rate was negative, indicating stability and the lack of hotspots on the disc.

2.4 Deduction and Conclusion

In this study, a Cu-based sintered metallic friction material was employed to investigate the response of metals to thermo-mechanical instabilities. The outcomes of this investigation are valuable for understanding and predicting the performance of non-metallic friction materials. The study revealed that at low operating conditions,
the dynamic friction coefficient between the friction pairs was high and fluctuated in
the output torque capacity, causing instability in the system. However, this did not
necessarily trigger the formation of hotspots but could result in vibration and noise
during clutch engagement.

On the other hand, when the sliding speed exceeded the critical speed of 200rpm,
the axisymmetric mode was activated, followed by the formation of hotspots on the
disc. The onset of the axisymmetric mode rendered the entire clutch unstable, leading
to thermoelastic instability. The presence of these hotspots had several negative
effects on the life of the clutch. Additionally, the growth rate of hotspot formation
on the disc was negative when the sliding speed was below the critical speed.

Based on both the experimental and numerical results, it can be concluded that
the inclusion of metals in friction pair is not a sufficient solution to address the
issue of thermo-mechanical instabilities. Depending on the operational conditions
and material compositions, the system may still be susceptible to the onset of
thermo-mechanical instabilities. Furthermore, the sliding velocity and temperature
were determined to have a significant impact on the initiation of stability.
CHAPTER III

INSTABILITIES IN METAL-FREE FRICTION MATERIAL

3.1 Introduction

As per the previous chapter, even with the inclusion of copper in friction pairs, there remain various possibilities of instability and environmental contamination. It is justifiable to introduce regulations limiting excessive copper content in friction materials and promoting the adoption of substitute materials like graphite, carbon fibers, nanotubes, and ceramics.

Nevertheless, the utilization of these alternative materials for friction purposes can potentially introduce operational safety concerns, primarily related to diminished friction caused by thermo-mechanical instabilities. Hence, it is of utmost importance to comprehensively understand the conditions leading to thermo-mechanical instabilities in these materials. Researchers have proposed these non-metallic materials as a promising alternative choice to replace metallic materials. Figure 3.1 demonstrates that the advantages and disadvantages of all current friction materials, including asbestos-based, steel fiber-based, steel, fiber and rubber combination, and copper-based friction material, have been widely documented and understood. However, there has been a lack of research on thermo-mechanical instabilities associated with copper-free friction materials during sliding.
3.1.1 Thermo-mechanical instability

The term "Thermo-mechanical instability" (TMI) refers to the interplay between various physical fields, including traditional thermoelastic instabilities and other distinct physical interactions such as wear, convective cooling, and material nonlinearities, including viscoelasticity. Manufacturers have become increasingly interested in this topic as sliding systems involve several of these physical fields. Frictional material properties have been found to significantly impact TMI in sliding interactions.

Depending on the material properties involved, the resulting temperature distributions may be highly non-uniform, leading to thermoelastic distortion [16]. This distortion, in turn, influences the contact pressure distribution and ultimately leads to instability of the system [91].

3.1.1.1 Estimating thermo-mechanical instabilities

Several researchers have developed models for thermoelastic instability (TEI) and thermo-mechanical instability (TMI), as well as related factors such as material properties, surface roughness, wear, and more, in homogeneous materials. For instance, researchers have developed models to predict the onset of TEI in a friction pair, stating that if the sliding speed of the system surpasses a critical value, instability occurs. This method was first introduced by Burton et al. [26]. It involves introducing a harmonic perturbation in the form of pressure or temperature between the sliding surfaces and monitoring the growth of the perturbation. The system becomes unstable when the imposed perturbation grows uncontrollably. This instability can lead to hotspot formations, wear, noise, vibration, and material damage on the disc [6, 92].
Figure 3.1: Diagram illustrating the pros and cons associated with existing friction materials
Furthermore, several studies [6, 39, 93] have demonstrated that the sliding speed between two half-planes can lead to instability when a small harmonic perturbation is imposed. Lee and Barber [94] expanded this concept to include a layer with a finite thickness sliding between two half-planes.

Additionally, Jang and Khonsari [93] developed an analytical model for symmetric and antisymmetric modes to analyze the contact between an insulator and a conductor. They found that critical speed is influenced by five independent dimensionless parameters. These authors also utilized the condition that thermoelastic instability occurs when the sliding speed surpasses a critical value to determine the threshold of thermoelastic instability in friction pairs, accounting for the effect of surface roughness [95].

3.1.1.2 Methods for estimating thermo-mechanical instabilities

The majority of studies on instability in sliding materials have been based on the eigenvalue approach. However, it is recommended to use the non-linear and transient modal approach for efficient numerical analysis of TMI [96]. Unlike the eigenvalue approach, which assumes conforming contact, the transient approach deals with nonlinearities such as surface separation, wear effect, asperity contact, etc. A truncated series approach (reduced-order models) was used by Al-Shabibi and Barber [4] for a simple approach to the transient solution of TEI. Meanwhile, Li and Barber [96] used the fast speed expansion method. To ensure accuracy, a finite element discretization method is a straightforward approach that can be implemented in finite element analysis software such as ABAQUS and ANSYS. An early attempt at using the numerical approach to study TEI was made by Kennedy [36]. Furthermore, numerical approaches have been used to investigate the transient thermo-mechanical phenomena for multi-disk clutches [45] and the nonlinear behavior of a sliding system.
with TEI [30], where the authors investigated the migration speed generated during sliding contact.

This study aims to investigate the effect of material anisotropy on TMI in metal-free friction materials by developing a non-linear transient solution. Specifically, a carbon-fiber-reinforced hybrid composite friction material with direction-dependent properties will be analyzed to better understand the behavior of anisotropic materials under sliding interactions. This approach distinguishes this study from previous research that mainly focused on homogeneous materials and did not consider the effect of material anisotropy on TMI.

Previous studies on the influence of material properties on TMI have focused on homogeneous materials. However, the growing demand for metal-free friction materials as a replacement for copper friction materials highlights the need for a better understanding of the response of these materials to TMI. Zhao et al. [97] attempted to investigate the effects of thermal conductivities, elastic moduli, and expansion coefficient on TEI using the eigenvalue approach. However, the influence of these properties in different directions (longitudinal, shear, and transverse) as observed in anisotropic materials were not considered. This study aims to investigate the influence of metal-free friction material properties in both the longitudinal and thickness directions on TMI using a transient approach. The study also incorporates both contact with surface separation effect and a full-contact regime for accuracy.

3.2 Finite element formulation of the TMI problem

3.2.1 Material consideration

This study utilizes a metal-free friction material composed of a carbon-fiber-reinforced composite. The material’s mechanical stiffness matrix
and thermal properties are determined using the rule of mixture approach by modifying a friction material model presented by Biczó et al. [98], where the copper component is excluded. Since the TMI model is two-dimensional, the material properties in the longitudinal (x-direction) and transverse (y-direction) directions are considered. The longitudinal direction represents the sliding direction, while the transverse direction represents the thickness direction, which is perpendicular to the sliding surface.

Table 3.1: Mechanical and thermal properties of the carbon-fiber-reinforced composite friction material

<table>
<thead>
<tr>
<th>Thermomechanical properties of the metal-free friction material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic modulus ((GPa))</td>
</tr>
<tr>
<td>\textit{Longitudinal} (E_{11})</td>
</tr>
<tr>
<td>\textit{Transversal} (E_{22})</td>
</tr>
<tr>
<td>Shear modulus (G_{12}(GPa))</td>
</tr>
<tr>
<td>Poison ratio, (\nu_{12})</td>
</tr>
<tr>
<td>Thermal conductivity, (K) ((W/(mK)))</td>
</tr>
<tr>
<td>\textit{Longitudinal} (K_{11})</td>
</tr>
<tr>
<td>\textit{Transversal} (K_{11})</td>
</tr>
<tr>
<td>Coefficient of thermal expansion (\alpha_{11}) ((10^{-6}K^{-1}))</td>
</tr>
<tr>
<td>\textit{Longitudinal} (\alpha_{11})</td>
</tr>
<tr>
<td>\textit{Transversal} (\alpha_{22})</td>
</tr>
</tbody>
</table>

Table 3.1 displays the mechanical and thermal properties of the metal-free friction material employed in the TMI investigation. It should be noted that the scope of the parametric investigation in this study is limited to the carbon-fiber-reinforced hybrid composite friction material (which is metal-free) utilized.
The elastic modulus values range between 9 and 11 GPa, thermal conductivity is between 1 and 4 W/mK, and the coefficient of friction ranges from 10 to 13 $\mu C^{-1}$.

Table 3.2: Shows the properties of the conducting material used in this analysis

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Steel disc</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of Elasticity, $E$(GPa)</td>
<td>210</td>
</tr>
<tr>
<td>Thermal conductivity,$K$ (W/(mK))</td>
<td>57</td>
</tr>
<tr>
<td>Coefficient of thermal expansion</td>
<td>12</td>
</tr>
<tr>
<td>Poisson ratio, $\nu$</td>
<td>0.4</td>
</tr>
<tr>
<td>Specific heat, $C$ (J/kg$^{-1}$K$^{-1}$)</td>
<td>460</td>
</tr>
<tr>
<td>Mass density, $\rho$(kg/m$^3$)</td>
<td>7250</td>
</tr>
</tbody>
</table>

3.2.2 Heat transfer formulation (Thermal analysis)

The general form of the two-dimensional transient conduction equation for both discs in the Cartesian coordinate system is described as:

$$\frac{\partial}{\partial x} \left( K_i \frac{\partial T_i}{\partial x} \right) + \frac{\partial}{\partial y} \left( K_i \frac{\partial T_i}{\partial y} \right) = \rho_i C_i \left( \frac{\partial T_i}{\partial t} + V_i \frac{\partial T_i}{\partial x} \right) \tag{3.1}$$

3.2.3 TMI model formulation

For the TMI analysis, two-disc blocks were used: a friction disc block with material properties listed in Table 3.1 and a steel disc block with properties presented in Table 3.2.
Both blocks were considered to be elastic and thermally conducting. The schematic of the two-disc blocks of different materials is shown in Figure 3.2, where Block 1 is the composite friction material and Block 2 is the conducting steel disc. Block 1 moves in-plane with a sliding speed of $V$ with respect to Block 2.

Furthermore, Block 1 can move in both the $x$ and $y$ directions, while Block 2 is constrained to move in the $y$-direction only. The two blocks have equal lengths $L$ and heights of $h_1$ and $h_2$ respectively, and when $y = y_1$, it represents the contact interface between the two blocks. It is worth noting that the range of the parametric study in this work is based on the carbon-fiber-reinforced hybrid composite friction material (metal-free) used. The specific heat, density, sliding speed and conductivity are represented by $C$, $\rho$, $V$, and $K$, respectively, where the subscript $i=1, 2$. When $i=1$, it indicates the expression for the friction plate with $V_1 = V$. On the other hand, when $i=2$, it represents the conduction equation for the disc, where $V_2 = 0$. 

Figure 3.2: Schematic of the sliding components
During sliding contact between the two blocks, the heat generated at the contact surface is defined by:

\[
q = f V P_i
\]

where \( f \) is the friction coefficient; \( V \) is Sliding velocity \((m/s)\) and \( P \) is the contact pressure \((Pa)\) at the interface.

### 3.2.3.1 Convective term/ Convergence problem

Equation 3.1 has a dual conductive-convective nature, which can lead to convergence problems with numerical solutions when using the standard Galerkin finite approach. This occurs because the convective term \( V \frac{\partial T}{\partial x} \) is not accurately discretized by this method. When the Peclet number is greater than 2, the algorithm can produce numerical oscillations that are physically meaningless, and for clutch and brake models, the Peclet number is usually greater than 100. To overcome this convective problem, the Petrov-Galerkin algorithm was used, which employs an upwinding approach [99, 100]. The backward-difference scheme is used for the dual conductive-convective term instead of the central-difference scheme, and non-symmetric weighing functions are used in the finite element to maintain the same shape functions as in the standard Galerkin finite element. In this analysis, the commercial finite element package ABAQUS [101] was used, which is equipped with the Petrov-Galerkin algorithm.

### 3.2.3.2 Boundary conditions

If the surfaces \( y = y_1 \) are in perfect contact, the clearance \( \Delta u_{y_1} = 0 \) and the contact is given implicitly by the conservation of energy condition as follows:

\[
K_1 \frac{\partial T_1}{\partial y} \bigg|_{y=y_1} - K_2 \frac{\partial T_2}{\partial y} \bigg|_{y=y_1} = q
\]

(3.3)
In the case of separation occurring at certain regions of the sliding interface, the heat generation in that region is zero since \( P_i = 0 \). Thus, we made the assumption that no heat conduction occurs across the gap during separation, which can be expressed as follows:

\[
\frac{\partial T_1}{\partial y} \bigg|_{y=y_i} - \frac{\partial T_2}{\partial y} \bigg|_{y=y_i} = 0
\]  

(3.4)

We introduced surface film condition \( \Omega_i(0.5W/m^2K) \), where \( i = 1,2 \), at an ambient temperature of 273K on the two exposed edges of the model due to the heat exchange with the environment as shown in Fig.3.2, giving:

\[
-K_1 \frac{\partial T_1}{\partial y} \bigg|_{y=0} = \Omega_1(T(x,0) - T_0)
\]  

(3.5)

\[
K_2 \frac{\partial T_1}{\partial y} \bigg|_{y=y_2} = \Omega_2(T(x,y_2) - T_0)
\]  

(3.6)

This process is important to prevent heat build-up resulting from poor heat dissipation. Additionally, cyclic and symmetrical boundary conditions are applied to the sides of the two blocks at \( x = 0 \) and \( x = L \) to ensure that the temperatures at the two opposite edges of each block are identical. This is to model the behavior of a complete circular disc sliding, rather than two separate blocks sliding.

\[
T_1 \bigg|_{x=0} (y,t) = T_1 \bigg|_{x=L} (y,t)
\]  

(3.7)

\[
T_2 \bigg|_{x=0} (y,t) = T_2 \bigg|_{x=L} (y,t)
\]  

(3.8)
3.2.4 Mechanical formulation (Static elastic analysis)

The resulting pressures during the contact of the two sliding blocks were computed using the general elastic contact in ABAQUS. With reference to Figure 3.2, at the edges where \( y = 0 \) and \( y = y_2 \), the following boundary conditions were imposed.

\[
\begin{align*}
(u_y)_{1,1} \bigg|_{y=0} &= 0 \\
(u_y)_{1,1} \bigg|_{y=0} &= \text{Constant}
\end{align*}
\] (3.9)

The displacement of the nodes in the y-direction is denoted by \( u_y \). At this boundary condition, an external force \( F_y \) was applied to act as a control in the y-direction, preventing sudden separation between the two blocks. Additionally, cyclic symmetry boundary conditions were defined at the side edges of the blocks where \( x = 0 \) and \( x = L \) for the displacement of nodes in both x and y directions. This representation was used to model the sliding behavior of two complete circular discs, rather than that of two blocks.

\[
\begin{align*}
(u_x)_{1,2} \bigg|_{x=0} &= (u_x)_{1,2} \bigg|_{x=L} \\
(u_y)_{1,2} \bigg|_{x=0} &= (u_y)_{1,2} \bigg|_{x=L}
\end{align*}
\] (3.10)

Furthermore, block 2 was restricted at \( y = y_2 \) to prevent any further movement in the positive y-direction as a result of the externally applied force \( F_y \).

3.2.5 Finite element mesh

Considering the two sliding bodies, the perturbation in contact pressure or temperature moves at a relatively high velocity close to the sliding velocity \( V \), with respect to the poor conductor. Therefore, it was assumed there is a high possibility
that the Peclet number associated with the friction material would be high and may lead to a high-temperature gradient in the y-direction near the contact surface [4, 30]. Therefore, mesh refinement was performed, particularly around the interface, to ensure that the model was capable of reproducing a strong variation of temperature in the skin layer by estimating the temperature distribution perpendicular to the interface [30]. To achieve this, the finite element mesh of the poor conductor was biased at 1.25 close to the contact surface, and a mesh refinement was carried out. Figure 3.3 indicates that the temperature perturbation in the y-direction normal to the contact surface was fully captured by the mesh points of the poor conductor, using a total of 750 elements.

Figure 3.3: Character of temperature distribution in the skin layer of the finite mesh of the poor conductor

For the thermal analysis, a 4-node convection/diffusion quadrilateral element type (DCC2D4) was utilized for the friction material. This was done to incorporate a convective term for the moving block 1. On the other hand, a 4-node linear heat transfer quadrilateral element type (DC2D4) was used for the non-moving conductor
material (block 2). In the elastic analysis, a 4-node bilinear plane strain quadrilateral element type (CPE4) was used for both the conducting and non-conducting materials.

3.3 Solution to the TMI formulation

3.3.1 Procedures used for TMI solution

- The initial step involves defining a small sinusoidal perturbation on the mean contact pressure, denoted by $P_m$, with a specific wavenumber of $(m = 2n\pi/L)$, where $n$ represents the number of hotspots. The perturbation could be any noise that is imposed on the mean contact pressure. This noise can be decomposed into a Fourier series, where each term represents a sinusoidal wave. Thus, to introduce this noise onto the contact interface between the two sliding bodies, we can directly add the sinusoidal wave to the mean pressure.

\[
P(x, t) = P_m + A\sin(\omega x + \phi)
\]  

(3.11)

where $P_m$ is the mean pressure and the remaining term represents the harmonic pressure perturbation; $A$ is the amplitude of the perturbation; $x$ is the distance (m); $\phi$ is the phase angle; $\omega$ is the angular velocity (rad/s).

Equation 3.2 is then used to calculate the frictional heat generated, where the variable contact pressure obtained from Equation 3.11 is used. Similarly, the mass flow is computed by using the composite friction material’s density, $\rho$, and the sliding velocity, $V$.

- A transient heat analysis is carried out using ABAQUS code and the resulting nodal temperatures are extracted from the contacting interface at $y = y_1$. The extracted nodal temperatures are fed into the elastic part of the analysis.
In the elastic analysis, the resulting pressure distribution is extracted from the contact interface and the new nodal contact pressure is used to compute the heat flux for the next transient heat analysis.

The process is repeated using an iterative scheme established using MATLAB. In the process, both the amplitude of the perturbation and the location of its peak pressure in the sliding direction will change until the solution converges where the form of the perturbation and its location remains unchanged within a specified accuracy through a single iteration.

When the sliding speed is lower than the critical speed, the expected contact pressures decay and converge to a uniform contact pressure at each iteration. On
the other hand, if the sliding speed \( V \) exceeds the critical speed \( V_c \), the imposed pressure perturbation grows and eventually causes some nodes to separate. However, to investigate how material properties affect TMI, we ensured that the sliding speed \( V \) was initially greater than the critical speed \( V_c \) during the analysis so that TMI can be observed. Figure 3.4 shows the entire iterative process explained.

3.3.2 Validation of the model

The focus of this investigation was to determine the circumstances in which a sinusoidal disturbance applied to constant pressure would experience exponential growth over time. If these conditions are met, we can infer that thermo-mechanical instability has occurred, as the disturbance will continue to grow indefinitely. Here, the expression for the contact pressure with time can be written as:

\[
P(x, t) = P_m + A \sin(\omega x + \phi) e^{bt}
\]  

(3.12)

where \( b \) is the growth rate. As the imposed perturbation increases, the mean pressure remains constant. In a stable system, it is anticipated that the growth rate will decrease over time. Dividing both sides of Equation 3.12 by the mean pressure gives \( P_m \):

\[
\frac{P(x, t)}{P_m} = 1 + \frac{A}{P_m} \sin(\omega x + \phi) e^{bt}
\]

(3.13)

Assuming that TMI occurs, it was considered that the first term on the right is significantly smaller than the second term, based on Equation 3.13. Therefore, the first term on the right was excluded. Taking the natural logarithm of both sides of the remaining expression, an equation of a straight line is obtained as expressed in Equation 3.14, where the slope of the line represents the growth rate, \( b \).
Figure 3.5: Graph of the (a) exponential growth of pressure perturbation with time, \( t \ [s] \) and (b) natural logarithm of the contact pressure perturbation as a function of time, \( t \ [s] \).
The condition for instability occurs when the slope of the line is positive.

\[ \ln\left( \frac{P(x,t)}{P_m} \right) = bt + c \]  

(3.14)

The equation represents a straight line, and the growth rate, \( b \), is indicated by its slope. A positive slope of the line indicates instability.

Figure 3.6: Comparison of growth rate vs. sliding velocity for an existing eigenvalue method and the currently developed model

For a velocity of 3.0\( m/s \), Figure 3.5a displays the exponential expansion of pressure perturbation at different points in time. The exponential growth indicates that the system is unstable, thereby initiating thermomechanical instability. Figure 3.5b depicts a straight line with a positive slope, resulting from plotting the natural logarithm of variable pressure divided by the mean pressure. The slope of the line represents the growth rate of the pressure perturbation. Additionally, during the initial phase, a specific mode of perturbation was triggered, representing only the
general pattern of the eigenmode. The actual mode gradually takes shape during the initial phase.

3.3.2.1 Comparison with eigenvalue approach

The developed algorithm involving a full-contact regime was validated by comparing its results with those of an eigenvalue approach using Hotspotter [102], which is an open-source software that estimates the growth rate, critical speed, and number of hotspot formations in sliding systems. Table 3.1 listed the material properties utilized, and since Hotspotter only permits isotropic material properties for thermal conductivity entry, we calculated the effective thermal conductivity and implemented it in Hotspotter. Both models assumed a friction coefficient of 0.2. Figure 3.6 displays a comparison of the growth rate against velocity for the two models. The results obtained from the developed transient model agree extremely well with the results from Hotspotter for full-contact regime analysis. The model’s accuracy instills confidence in the simulation results generated in this study.

3.4 Contact type used in the TMI Analysis

3.4.1 Full-contact regime (Contact without separation)

The model considered a full-contact engagement without any separation, where all contact surface nodes remained in complete contact during the sliding process. This assumption resulted in a linear thermo-mechanical problem since full contact was maintained without any separation. This approach is similar to the eigenvalue approach used in early studies of thermoelastic instability [26, 25], which was further enhanced by Yi et al. [102] using a finite element implementation method. In this work, a full-contact regime analysis was performed using ABAQUS code to investigate the dominant material parameter in TMI formation. The sliding speed was fixed
at approximately $V = 3.0\text{m/s}$, and the simulation was conducted for a total time of $t = 80\text{s}$, with a step time $\Delta t \leq 4 \times 10^{-4}\text{s}$ computed using a Courant number $Cu = V\Delta t/h_x \leq 1$, where $h_x$ denotes the mesh size in the x-direction. In this case, a specified time was used for the simulation because the problem is linear, and there was no node separation due to non-linearity. As a result, the pressure and temperature could continue to grow exponentially, leading to very high peak values. Hence, it was necessary to terminate the iteration process for a specific time period.

3.4.2 Contact with separation

In actual practice, the thermo-mechanical behavior of clutch and brakes is nonlinear, mainly due to loading, geometry, and material nonlinearity. Therefore, it is crucial to consider the separation effect since above certain temperatures, the problem may involve regions of separation and become nonlinear. Separation occurs when the imposed harmonic perturbation increases at each iteration, leading to some contact nodes starting to separate. This usually results in a rapid contraction of the contact area and a corresponding increase in the maximum contact pressure due to flash temperature. This phenomenon may be due to the local reduction or constriction of the heat path, which increases the density of thermal flux lines at the contact spot, causing thermal spikes that represent the hot spot formation on the discs. The amplitude of the perturbation continues to increase at the contact surface as more nodes separate with further iterations until the solution converges, and the form of the perturbation and its location remains unchanged using a step time $\Delta t \leq 4 \times 10^{-4}\text{s}$. The non-uniform contact pressure distribution on the contact surface leads to the non-uniform distribution of temperature that causes hotspot, judder, and thermal buckling, as observed on brake and clutch discs [18, 103, 104].
As previously stated, the properties of the material significantly affect the formation of thermo-mechanical instability. Hence, we investigate the effects of the dominant material properties by considering a non-linear steady-state solution that allows for surface separation.

3.5 Effect of Material Properties on TMI

3.5.1 Effect of thermal conductivity

The role of material properties in determining the stability of the system, particularly thermal conductivity, has been highlighted in previous studies such as Zhao et al. [105]. It has been observed that increasing the thermal conductivity of friction materials can lead to an increase in the critical speed, emphasizing the significance of material properties in ensuring system stability. In this study, the impact of thermal conductivity in both the longitudinal and transverse directions on the stability of the sliding system is investigated using a transient approach.

3.5.1.1 Longitudinal direction ($K_{11}$)

The investigation begins by varying the thermal conductivity in the longitudinal direction while holding other material parameters constant at $K_{22}$, $E_{11}$, $E_{22}$, $\alpha_{11}$, and $\alpha_{22}$ values of $4W/(mK)$, 11GPa, 11GPa, $13 \times 10^{-6}C^{-1}$, and $13 \times 10^{-6}C^{-1}$, respectively. Specifically, $K_{11}$ was varied in the range of $1 - 4W/(mK)$. Figure 3.7a shows the contact pressure distribution without separation, while Figure 3.7c shows the distribution with separation effect. Our findings reveal that increasing $K_{11}$ did not significantly affect the system instability.

Figure 3.7b, which is a magnified portion of Figure 3.7a, indicates that increasing the thermal conductivity $K_{11}$ results in a slight increase in contact pressure. However, when $K_{11}$ is equal to the constant thermal conductivity $K_{22}$ in the thickness direction
(4W/(mK)), the ratio of contact pressure $P(x, t)/P_m$ decreases from 47591 to 47074 in the full contact regime and from 11.14 to 10.79. Further investigation reveals that the contact pressure increases again when $K_{11}$ is greater than $K_{22}$.
Figure 3.7: Surface contact pressure distribution for thermal conductivity in the longitudinal direction for: a) full-contact regime b)magnified portion of the full-contact regime c) contact with separation effect.

Figure 3.8: Temperature distribution along the contact interface for thermal conductivity in the longitudinal direction with separation effect.
When the thermal conductivity in the thickness direction \((K_{22})\) equals the thermal conductivity in the longitudinal direction \((K_{11})\), the former dominates, leading to a slight drop in temperature as seen in Figure 3.8.

However, the maximum contact pressure increases slightly with increasing \(K_{11}\), as observed in the magnified portion of Figure 3.7b. To further understand the influence of \(K_{11}\) on TMI, we computed the growth rates of contact pressure by increasing \(K_{11}\). The growth rates at \(K_{11} = 1, 2, 3, 4\) W/(mK) were found to be 0.126, 0.269, 0.2670, and 0.2671 s\(^{-1}\), respectively. Therefore, increasing \(K_{11}\) causes the growth rate to increase slightly, resulting in a reduction in the critical speed and making the system unstable.

### 3.5.1.2 Transverse direction/ Thickness direction \((K_{22})\)

The thermal conductivity in the transverse direction was also varied \(K_{22} = 1, 2, 3, 4\) W/(mK) while keeping all other material properties constant. Thus, keeping \(K_{11}, E_{11}, E_{22}, \alpha_{11}, \alpha_{22}\) at 4 W/(mK), 11 GPa, 11 GPa, \(13 \times 10^{-6} C^{-1}\), \(13 \times 10^{-6} C^{-1}\) respectively. Figures 3.9a and 3.9b depict the resulting contact pressure distribution for contact with and without separation respectively, and highlight the significant effect of contact pressure on system stability. Increasing the thermal conductivity in the transverse direction, \(K_{22}\), results in a steady decay of the imposed pressure perturbation, which stabilizes the system. This effect is particularly pronounced in the full-contact regime shown in Figure 3.9a.

For contact with separation, as shown in Figure 3.9b, when \(K_{22} = 1.0\) W/(mK), separation occurred in certain regions, accompanied by a rapid contraction of the contact area at \(0.52 < x/L < 0.52\). The maximum value of \((P(x,t)/(P_m))\) observed was 10.96.
Figure 3.9: Surface contact pressure distribution for thermal conductivity in the transverse direction a) full-contact regime b) with separation effect
Further increasing $K_{22}$ to 2, 3, and 4 W/(mK) resulted in certain regions having more contact interaction, leading to the evolution of new local maximum pressures. This is due to the enhancement of the uniformity of the contact pressure, resulting in a gradual improvement in the temperature distribution on the contact surface, as shown in Figure 3.10. With a thermal conductivity of $K_{22} = 1.0$ W/(mK), the temperature distribution across the contact surface is highly non-uniform. At a contact distance of $0 < x/L < 0.47$, the temperature was approximately $T=8^\circ C$, and suddenly increased rapidly to a maximum value of $T=202.5^\circ C$.

Meanwhile, as $K_{22}$ gradually increases to 4.0 W/(mK), the temperature profile becomes more uniform. The temperature at the start is $T=88.4^\circ C$ and gradually decreases to $T=13^\circ C$ at $x/L = 0.37$, before slowly increasing again to a local maximum value of $T=82.3^\circ C$ at $x/L = 0.49$. The maximum temperature is $T=85^\circ C$ at $x/L = 0.94$. 

Figure 3.10: Temperature distribution along the contact interface for thermal conductivity in the transverse direction with separation effect.
The study found that the growth rate $b$, [$s^{-1}$] of contact pressure varies with the thermal conductivity $K_{22}$, with values of 0.4374, 0.4004, 0.374, and 0.2650 for $K_{22}$ equal to 1, 2, 3, and 4 $W/(mK)$, respectively. These growth rates explain the observed behavior of contact pressure shown in Figure 3.9. In addition, decreasing the growth rate leads to an increase in the critical sliding speed of the system, which results in a more stable system.

3.5.2 Effect of elastic modulus

Previous studies have demonstrated that materials with a high elastic modulus tend to promote the formation of hotspots, which can lead to instability in the system. However, there has been a lack of research on the role of materials with direction-dependent properties, and how they contribute to the formation of thermal-mechanical instability (TMI). Therefore, we conducted an investigation into the unstable behavior of the friction material caused by the elastic modulus in both the longitudinal direction ($E_{11}$) and the thickness direction ($E_{22}$).

3.5.2.1 Longitudinal direction ($E_{11}$)

In order to analyze the effect of elasticity on thermal-mechanical instability (TMI), we conducted a study in which we kept several parameters constant, including $K_{11}$, $K_{22}$, $E_{22}$, $\alpha_{11}$, and $\alpha_{22}$ at values of $4W/(mK)$, $4W/(mK)$, 11GPa, $13 \times 10^{-6}C^{-1}$, and $13 \times 10^{-6}C^{-1}$, respectively. We then varied the elasticity $E_{11}$ by setting it to values of 9, 10, 11, and 12 GPa, and compared the resulting contact pressure distributions with and without the surface separation effect, as shown in Figures 3.11a and 3.11b, respectively.
Figure 3.11: Surface contact pressure distribution for elastic modulus in the longitudinal direction a) full-contact regime b) with separation effect.
It is important to note that in the full contact regime (no separation), as seen in Figure 3.11a, the maximum contact pressure increases as the elastic modulus $E_{11}$ is increased, except for $E_{11} = 11$. This is because the problem becomes nonlinear at a certain point but is restricted to linear, resulting in no contact separation and continuous heat generation at the contact interface. This approach makes it difficult to predict the actual behavior of the system at extremely high temperatures since the contact is constrained.

![Figure 3.12: Temperature distribution for elastic modulus in the longitudinal direction with separation effect](image)

To better understand the behavior of the system at extremely high temperatures, it is crucial to consider the non-linear case by introducing contact with the separation effect. In contrast to the full-contact regime, the maximum pressure was found to decrease as $E_{11}$ increased. This is because, at a steady state, the temperature may cease to grow when equilibrium is reached. Furthermore, we observed that the corresponding growth rate, $b, s^{-1}$, decreased slightly as $E_{11}$ increased, with values of
0.256, 0.2366, 0.218, and 0.215 for \( E_{11} = 9, 10, 11, \) and \( 12 \) GPa, respectively. This suggests that increasing \( E_{11} \) leads to an increase in the critical velocity of the system.

In addition, the temperature profiles indicate a minor decrease in the maximum temperature from \( T = 211.25^\circ C \) to \( T = 206^\circ C \) as the elastic modulus \( E_{11} \) is increased (refer to Figure 3.12). For instance, at the end of the contact pressure distribution profile where \( x/L = 0.94 \), the temperatures were measured to be \( T = 7.98, 14.36, 21.9, \) and \( 22.1^\circ C \) for \( E_{11} \) values of 9, 10, 11, and \( 12 \) GPa, respectively. This suggests that increasing \( E_{11} \) slightly improves the uniformity of temperature.

3.5.2.2 Transverse direction (\( E_{22} \))

Figures 3.13a and 3.13b display the evolution of contact pressure distribution for two cases: the full-contact regime and the regime with separation effect. The elastic modulus in the thickness direction, \( E_{22} \), was varied between 9 and 12 GPa while all other parameters were held constant. The results demonstrate that the elastic modulus in the thickness direction plays a crucial role in promoting the formation of TMI. In the full-contact regime, as the value of \( E_{22} \) was increased, the contact pressure grew rapidly. Similarly, in the regime with the separation effect, an increase in \( E_{22} \) also led to the growth of contact pressure. However, in this case, the growth was gradual, as illustrated in Figure 3.13b. By increasing \( E_{22} \) at 9,10,11 and \( 12 \) GPa, the corresponding growth rates \( b \ (s^{-1}) \) were found to be 0.094, 0.111, 0.190, and 0.215, respectively.

The results of the analysis showed that the growth rates of TMI decrease as the critical sliding speed increases with a decrease in the elastic modulus, \( E_{22} \). This trend can be attributed to the fact that the thickness direction of the material becomes more dominant with an increase in \( E_{22} \), leading to higher contact pressure.
Figure 3.13: Contact pressure distribution for elastic modulus in the transverse direction a) full-contact regime b) with separation effect.
This higher contact pressure, in turn, generates more friction heat, making the friction material more susceptible to TMI.

Moreover, as shown in Figure 3.14, an increase in $E_{22}$ leads to a slight rise in the temperature of the friction material. Further increasing the value makes it susceptible to TMI. Therefore, the critical sliding speed required to trigger TMI decreases with an increase in $E_{22}$, indicating that materials with higher values of $E_{22}$ are more susceptible to TMI.

3.5.3 Effect of coefficient of thermal expansion

3.5.3.1 Longitudinal direction ($\alpha_{11}$)

In a similar vein, the study investigated the effect of varying the thermal expansion in the longitudinal direction, $\alpha_{11}$, while holding all other parameters, including $K_{11}$, $K_{22}$, $E_{11}$, $E_{22}$, and $\alpha_{22}$, constant at $4W/(mK)$, $4W/(mK)$, 11GPa, 11GPa, and $13 \times 10^{-6}C^{-1}$, respectively.
Figure 3.15: Surface contact pressure distribution for the coefficient of thermal expansion in the longitudinal direction a) full-contact regime; and b) with separation effect
The analysis revealed that an increase in $\alpha_{11}$ led to an improvement in stability, as observed in the full-contact regime depicted in Figure 3.15a. However, when $\alpha_{11}$ was equal to $\alpha_{22}$, the opposite effect was observed in both contact analyses. Specifically, the growth rate of contact pressure increased slightly, indicating that $\alpha_{11}$ started to dominate over the other parameters. The results demonstrated that for contact with separation, the coefficient of thermal expansion, $\alpha_{11}$, had only a slight impact on TMI, as shown in Figure 3.15b.

The analysis yielded growth rates of 0.0374, 0.0375, 0.0378, and 0.0380 for the coefficient of thermal expansion, $\alpha_{11}$, for values of $10 \times 10^{-6} C^{-1}$, $11 \times 10^{-6} C^{-1}$, $12 \times 10^{-6} C^{-1}$, and $13 \times 10^{-6} C^{-1}$, respectively. The results indicated that increasing $\alpha_{11}$ had a slightly discouraging effect on TMI. The overall conclusion was that the influence of thermal expansion in the sliding direction was not significant, particularly in the case of contact with separation. In other words, the coefficient of thermal expansion of a friction material had a low impact on sliding stability when contact separation occurred.

3.5.3.2 Transverse direction ($\alpha_{22}$)

A similar approach was taken to investigate the effect of the coefficient of thermal expansion in the thickness direction, with values of $\alpha_{22} = 10, 11, 12, 13 \mu C^{-1}$ being considered. The evolution of contact pressure distribution for the full contact regime was observed, and it exhibited the same behavior as seen by varying the coefficient of thermal expansion in the sliding direction. This behavior can be visualized in Figure 3.16a.

In the case of contact with separation effect, it was found that the maximum peak value of the perturbation $P(x, t)/(P_m)$ was almost the same for each varied coefficient of thermal expansion in the thickness direction ($\alpha_{22} = 10, 11, 12, 13 \mu C^{-1}$).
Figure 3.16: Surface contact pressure distribution for the coefficient of thermal expansion in the transverse direction a) full-contact regime; and b) with separation effect.
The corresponding growth rate for increasing $\alpha_{22}$ values were found to be 0.0374, 0.0372, 0.0381, and 0.0380, respectively. These findings suggest that the coefficient of thermal expansion in the thickness direction has only a minor effect on TMI. Additionally, as the coefficient of thermal expansion was increased, the direction of the perturbation moved toward the left. Despite these observations, the effect of the varied parameters on the system stability was insignificant, except for the position of the perturbation, which changes in the contact with the separation effect, as shown in Figure 3.16b.

3.6 Conclusion

This study aimed to investigate the impact of material properties such as elastic modulus, thermal conductivity, and coefficient of thermal expansion, on thermo-mechanical instability (TMI) in metal-free friction materials. To achieve this, a thermomechanical model was developed and validated using an existing eigenvalue approach. The analysis was performed for both full-contact regime and contact with separation effect.

The results showed that elastic modulus $E_{11}$ and thermal conductivity $K_{22}$ were effective in reducing TMI formation. Conversely, $K_{11}$ and $E_{22}$ were found to increase the susceptibility of the material to TMI. Furthermore, the variation of $E_{11}$ in full-contact regime analysis promoted instability.

Regarding the coefficient of thermal expansion, it was found to have a minor role in TMI formation when separation occurs. However, it was observed to influence the direction of the contact pressure perturbation and, thus, the position of hotspot formations during sliding interactions in the thickness direction.
Overall, this study sheds light on the critical material properties that affect TMI in metal-free friction materials and provides insights into how to mitigate this phenomenon in practical applications.
4.1 Introduction

It is widely understood within the field of mechanical engineering that the interaction between the friction ring section of a rotating disc, also known as a rotor, and the brake or clutch lining material is often accompanied by vibrations. These vibrations occur at the interface where the two materials slide against each other and have the potential to excite the entire rotor, causing unwanted noise. Unfortunately, this noise can be transmitted to other parts of the vehicle and may result in discomfort for the vehicle occupants.

Research has shown that the noise generated by this interaction is more pronounced when metallic or semi-metallic brake lining materials are used [106, 107]. To address this issue, manufacturers have taken an innovative approach by applying damping elements to the backing plate of the brake assembly. This approach aims to reduce the transmission of noise by introducing materials that are specifically designed to dampen vibrations.

One example of this approach is the use of a laminate that includes one or two layers of viscoelastic adhesives bonded to one or more intermediate materials. These materials are designed to form the brake pad assembly and act as a damping layer that helps to reduce the transmission of noise. This innovative approach has proven effective in reducing noise and improving the comfort of vehicle occupants [108].
4.1.1 Thermo-mechanical Instabilities

According to several studies [109, 110, 111], it has been demonstrated that the utilization of damping materials in brake components can effectively dissipate the vibration and noise produced by brake systems. However, it has been observed that the application of damping materials to the backplate alone is not sufficient to reduce the transmission of unwanted noise to the occupants of the vehicle [112]. This outcome is not surprising as the noise and vibration in brake systems mainly arise due to thermomechanical instabilities that occur at the interface of the sliding pairs rather than at the backing plate. Therefore, the most effective method to mitigate the noise and vibration generated in brake systems is to consider a specific class of friction materials that possess acceptable thermophysical characteristics, a stable friction coefficient, optimal elastic behavior, and damping properties as suggested by Sergienko et al. [113].

Despite the potential benefits of utilizing low-viscoelastic materials as friction linings to eliminate noise, vibration, and squeal in sliding systems, there have been limited studies on how such materials could impact brake performance. Therefore, this work aims to investigate the impact of physical material properties on the onset of thermo-mechanical instabilities in brake pads when a low-viscoelastic material is used as a friction lining. This type of material has been found to effectively dampen noise and vibration in sliding systems [18, 108]. The study will specifically examine the contact behavior between the friction material and the rotor, focusing on how the material’s physical properties could affect the onset of thermomechanical instabilities.

4.1.2 Contact behaviour

When two solid bodies come into contact, they generate contact stresses at the interface due to an applied load. In the case of sliding, the temperature can also
influence the contact stresses. The contact stresses can cause thermo-mechanical instability, especially when peak values are reached. To prevent this instability, it is necessary to reduce the peak values of the contact stresses and make the interface pressure as uniform as possible. This can be achieved by using viscoelastic materials and understanding the mechanism of contact systems.

There have been few studies on the effect of viscoelastic materials on thermo-mechanical instabilities in automotive disc brakes and clutches. Decuzzi developed an analytical model to analyze the effect of viscoelastic and poroelastic properties of friction materials on the onset of frictionally excited thermoelastic instability. However, this model had some limitations. It assumed that the critical sliding speed of the pure metallic material was close to zero, while that of the viscoelastic material was equal to the sliding speed. Additionally, the model assumed that the thermal conductivity of the friction material was zero, which leads to an overestimation of the critical sliding speeds and higher instability predictions for real materials.

Moreover, the model in question fails to offer an effective means of simultaneously obtaining the critical sliding speed of both sliding materials. It is crucial to note that while there have been numerous research works on the contact interactions of solids involving viscoelastic materials, only a few have focused on the interfacial behavior of surfaces and how it affects sliding instability.

For instance, Yu et al. [114] derived viscoelastic contact models that offer a more realistic contact analysis. Additionally, they developed an empirical method-based model to determine the localized properties of a viscoelastic material. In another study, Úradniček et al. [115] numerically and experimentally investigated the influence of material-dependent damping on brake squeal in disc brake systems.
They found that non-proportional material-dependent damping could significantly affect the stability of the brake system. Furthermore, Zhao et al. [116] conducted an important study on friction analysis, focusing on an anisotropic surface by considering a viscoelastic material. This study extended Persson’s [117, 118] work on studying sliding friction and the effect of the elastic modulus on the contact area.

Feng et al. [119] conducted a study that aimed to establish the relationship between viscoelastic friction and wear by examining a polymer composite friction lining material and wire rope. The study resulted in the development of a mathematical model that utilizes a numerical approach to predict the critical sliding speed of friction material with viscoelastic parameters. Materials with viscoelastic properties are commonly used in the automotive industry because of their ability to absorb noise and vibrations from systems.

The research here specifically looked into how viscoelastic materials respond to thermo-mechanical instability when used as friction materials in brake pads and clutch discs to control noise and vibration. However, the study did not take into consideration the frequency response and its consequent vibration and noise effects during sliding interaction. Instead, the focus was on the response of the viscoelastic parameters to thermomechanical instability.

The primary goal of this study is to predict the critical sliding speed for both viscoelastic and pure elastic friction materials during sliding in order to determine the onset of instabilities. This is a key distinction from previous research as the model takes into account thermal conductivity parameters, which were not considered in the limited case examined by Decuzzi.
4.2 Formulation of the problem

4.2.1 Mathematical model

Multiple models have been established to describe the behavior of viscoelastic materials. However, the simplest model that adequately predicts this behavior, and is widely used in solid mechanics, is the standard linear solid model also known as the three-parameter model [120]. The three-parameter model is established by either adding a spring in series or parallel to a Maxwell model [121].

The Maxwell model consists of a spring and dashpot connected in series. In this work, a spring placed in parallel with a series connection of a spring and a dashpot is considered to represent the viscoelastic model shown in Figure 4.1. The physical model, which consists of a viscoelastic friction material and a steel disc, is treated as two straight blades on a single plane, in contact along a straight common interface as shown in Figure 4.2.

Figure 4.1: Standard solid model describing the viscoelastic behavior
The model does not consider thermal radiation nor thermal convection heat exchange with the surrounding air. The fundamental mathematical derivation, which describes the thermo-mechanical behavior was obtained from Burton et. al., [26].

4.2.1.1 Derivation of the mathematical model

If the linearly viscoelastic plate designated as body 1 is loaded with a sinusoidal pressure, the strain response is also sinusoidal with the same frequency as the applied pressure but lags by a phase angle $\delta$. Thus,

$$\varepsilon_1 = \varepsilon_0 \cos(\omega x - \delta)$$  \hspace{1cm} (4.1)

Where $\varepsilon_0$ is a constant. Moreover, Equation (4.1) can be rewritten as:

$$\varepsilon_1 = \varepsilon_0 e^{i\omega x - \delta}$$  \hspace{1cm} (4.2)

$$\varepsilon_1 = \varepsilon_0 e^{i\omega x} e^{-i\delta}$$
Considering the linear elastic plate designated as body 2, the strain response does not lag as in the case of the viscoelastic material and can be expressed as:

\[ \varepsilon_2 = \varepsilon_0 e^{i\omega x} \Rightarrow \varepsilon_0 e^{i\omega x} \] (4.3)

According to Burton’s [122], Equation (4.4) and Equation (4.5) can be used to express the surface pressure on bodies 1 and 2 respectively, assuming the surface is held flat.

\[ P'_1 = \frac{E_1 \alpha_1 T_0 k_1}{V_{c_1}} \left[ \left( \omega - b_1 \right) \cos (\omega x) + a_1 \sin (\omega x) \right] \] (4.4)

\[ P'_2 = \frac{E_1 \alpha_1 T_0 k_2}{V_{c_2}} \left[ \left( \omega - b_2 \right) \cos (\omega x) + a_2 \sin (\omega x) \right] \] (4.5)

Here:

\[ a_i = \left[ -\frac{\omega^2}{2} + \frac{\omega}{2} \left\{ \omega^2 + \left( \frac{V_{c_1}}{k_1} \right)^2 \right\} \right]^{\frac{1}{2}}, \quad b_i = \left[ \omega^2 + \left( \frac{V_{c_1}}{k_1} \right)^2 \right]^{\frac{1}{2}}, \quad i = 1, 2 \]

Assuming that each surface undergoes equal and opposite displacement when the two bodies are in contact, the pressure on body 1 can be expressed as \( P = P'_1 - P''_1 \) if we consider a cosine wave distribution of the displacement. Where:

\[ P''_1 = -\frac{E_1 \omega \varepsilon_1}{2} - \frac{E_1 \omega \varepsilon_0 e^{i\omega x} e^{-i\delta}}{2} \] (4.6)
For body 2, the pressure on the body is given as \( P = P'_2 - P''_2 \). Where:

\[
P''_2 = \frac{E_2 \omega \varepsilon_2}{2}
= \frac{E_2 \omega \varepsilon_0 e^{i \omega x}}{2}
\]  

(4.7)

If we take into account the equilibrium condition where the pressure (P) is the same on both bodies, we can express P as:

\[
P = \frac{E_1 P'_2 + E_2 P'_1 e^{-i \delta}}{E_1 + E_2 e^{-i \delta}}
\]  

(4.8)

By substituting \( P'_1 \) and \( P'_2 \) into Equation (4.8) We obtain:

\[
P = \frac{E_1 E_2 T_0}{E_1 + E_2 e^{-i \delta}} \left[ \left( \frac{\alpha_1 k_1 (b_1 - \omega)}{V_{c_1}} e^{-i \delta} - \frac{\alpha_2 k_2 (b_2 - \omega)}{V_{c_2}} \right) \cos(\omega x)
+ \left( \frac{\alpha_1 a_1 k_1 e^{-i \delta}}{V_{c_1}} \frac{\alpha_2 a_2 k_2}{V_{c_2}} \right) \sin(\omega x) \right]
\]  

(4.9)

In order to achieve equilibrium, the heat generated by friction must be equal to the heat conducted from the interface, as expressed below:

\[
\mu P(V_{c_1} + V_{c_2}) = q_{\text{net}}
\]  

(4.10)

Where: \( q_{\text{net}} \) is the net heat flow between the bodies as expressed in Equation (4.11). Reference to the derivation can be found in Burton et al. [26]

\[
q_{\text{net}} = q_1 + q_2
= T_0 \left\{ (K_1 b_1 + K_2 b_2) \sin (\omega x) + (K_2 a_2 - K_1 b_1) \cos (\omega x) \right\}
\]  

(4.11)
Substituting Equation (4.9) and Equation (4.11) into Equation (4.10). We get:

\[
\frac{\mu E_1 E_2 (V_{c1} + V_{c2})}{E_1 + E_2 e^{-i\delta}} \left[ \left( \frac{\alpha_1 k_1 (b_1 - \omega) e^{-i\delta}}{V_{c1}} - \frac{\alpha_2 k_2 (b_2 - \omega)}{V_{c2}} \right) \cos \omega x \right.
\]
\[
\left. + \left( \frac{\alpha_1 a_1 k_1 e^{-i\delta}}{V_{c1}} + \frac{\alpha_2 a_2 k_2}{V_{c2}} \right) \sin (\omega x) \right] = \left\{ (K_1 b_1 + K_2 b_2) \sin (\omega x) + (K_2 a_2 - K_1 b_1) \cos (\omega x) \right\} \tag{4.12}
\]

Where \( e^{-i\delta} = (\cos \delta - i \sin \delta) \). Note that, \( i \sin \delta \) is the imaginary component. Rearranging the coefficients of the sine and cosine terms to be equal on both sides of Equation (4.12) gives:

\[
\frac{\mu E_1 E_2 (V_{c1} + V_{c2})}{E_1 + E_2 (\cos \delta)} \left[ \left( \frac{\alpha_1 k_1 (b_1 - \omega)(\cos \delta)}{V_{c1}} - \frac{\alpha_2 k_2 (b_2 - \omega)}{V_{c2}} \right) \right]
\]
\[
= (K_2 a_2 - K_1 b_1) \tag{4.13}
\]

\[
\frac{\mu E_1 E_2 (V_{c1} + V_{c2})}{E_1 + E_2 (\cos \delta)} \left[ \left( \frac{\alpha_1 a_1 k_1 (\cos \delta)}{V_{c1}} + \frac{\alpha_2 a_2 k_2}{V_{c2}} \right) \right]
\]
\[
= (K_1 b_1 + K_2 b_2) \tag{4.14}
\]

To obtain \( V_{C1} \) and \( V_{C2} \) for the viscoelastic friction material and the steel material respectively, Equations (4.13) and (4.14) are solved using numerical techniques such as Gaussian elimination.

To derive the expression for the phase lag, the standard linear model is utilized, which includes a spring \( E_2 \) in parallel to a series connection of a spring \( E_1 \) and a dashpot \( \zeta \).
This model is illustrated in Figure 4.1 and was originally obtained by Findley et al. [123]. The expression for the phase lag is given as

\[ \delta = -\arctan \left( \frac{\omega \zeta E_2^2}{(E_1 + E_2)\omega^2 \zeta^2 + E_1 E_2^2} \right) \] (4.15)

Where the load frequency \( \omega \) is given as:

\[ \omega = \frac{V_m}{2\pi} \] (4.16)

Where: \( V = V_{c_1} + V_{c_2} \) is the sliding velocity and \( m \) is the wave number of the imposed perturbation.

Figure 4.3: Schematic of the two sliding bodies

4.2.2 Finite element approach

4.2.2.1 Modeling of the problem

To validate the derived expressions for the critical speed of the viscoelastic friction material siding against the metal material, a finite element model is utilized.
Two thermally conducting plates are used for this purpose. Plate 1 represents the viscoelastic material that moves in-plane with a sliding velocity $V$ with respect to plate 2, which is the conducting material. While plate 1 moves in both the $x$ and $y$ directions, plate 2 is constrained in both the $x$ and $y$ directions.

A schematic of the sliding bodies used in this study is depicted in Figure 4.3. Both plates have equal lengths $L$ and heights of $h_1$ and $h_2$, respectively. The contact interface between the two plates is represented by $y = y_1$. Further details regarding the applied boundary conditions, mesh refinement procedures, and model accuracy can be found in Koranteng et al. [124].

In this analysis, the commercial finite element software package called ABAQUS [101] is used. It is equipped with the Petrov-Galerkin algorithm, which is a numerical method used to solve complex partial differential equations. The analysis considers a viscoelastic behavior of a material using a single Maxwell model. In this model, the viscoelastic parameters that govern the material behavior are the shear modulus, bulk modulus, and the relaxation time, denoted by $G$, $K$, and $\tau$ respectively. These parameters are computed and incorporated into the ABAQUS input script. The relaxation time $\tau$ is calculated as the ratio of the viscous damping coefficient $\zeta$ to the elastic modulus $E_1$. The input script applies these parameters to the simulation, allowing for the accurate modeling of the viscoelastic behavior of the material.

4.2.2.2 Solution procedure for the viscoelastic model

The following list outlines the procedure used in this study to solve the viscoelastic model.
• Impose a sinusoidal perturbation having a specific wavenumber \( m = 2n\pi/L \) onto a mean contact pressure as:

\[
P(x, t) = P_m + Asin(\omega x - \phi)
\]  

(4.17)

To obtain a response from viscoelastic materials under short-time loading, a sinusoidal load is preferred over a static load as stated by Findley et al. [123]. To compute the frictional heat at the contact interface \( y = y_1 \), the pressure from Equation (4.17), the friction coefficient and the sliding velocity is used. The mass flow onto the moving body is defined as the product of the density, \( \rho \), of the moving body and the sliding velocity, \( V \). These parameters are then incorporated into a user-defined Subroutine in ABAQUS.

• A heat transfer analysis is conducted to obtain the nodal temperatures at the contact interface during sliding where \( y = y_1 \). These temperatures are then utilized as input for a dynamic viscoelastic step in ABAQUS.

• The contact pressure resulting from the viscoelastic analysis is extracted from the sliding interface. The new nodal contact pressure is used to compute the heat flux for the next heat transfer analysis.

• Repeat the process using an iterative scheme as shown in the simulation procedure in Figure 4.4. For each iteration, extract the maximum contact pressure or temperature, while taking note of the time steps involved.

• Obtain the natural logarithm of the data obtained in step 3 and create a graph that shows the relationship between the natural logarithm of the contact pressure or temperature and time. The graph will have a linear shape, with either a positive or negative slope. The slope of the graph reflects the rate of growth of the perturbation.
Figure 4.4: Simulation solution procedure
A positive slope indicates instability, where $V > V_{C_1}$, while a negative slope indicates stability within the system, where $V < V_{C_1}$. As the critical speed is not known at first, adjust the sliding speed while monitoring the maximum temperature or contact pressure observed in each iteration.

- When determining the critical speed, a linear interpolation method is utilized that relies on the premise that the growth rate, $b$ equals zero at the critical speed. If the system is stable, then the critical sliding speed will be relatively high. Conversely, a lower critical sliding speed suggests that the system is unstable.

4.2.2.3 Convective term and convergence problem

In conductive-convective problems, numerical errors due to convergence issues are common. These issues arise particularly when the Peclet number exceeds 2. In the case of clutch and brake problems, the Peclet number tends to be higher than 100. Furthermore, using the standard Galerkin finite element method often results in significant numerical errors. To overcome these issues, this investigation utilizes ABAQUS [101], a commercial finite element package that is equipped with a Petrov-Galerkin algorithm that can handle conductive-convective problems. The algorithm uses an upwinding approach [99, 100], which replaces the central-difference scheme with the backward-difference scheme for the conductive-convective term.

The high Peclet number associated with the brake/clutch model is due to a perturbation moving at a fast speed close to the value of the sliding velocity with respect to the poor conductor. This causes a high-temperature gradient in the y-direction near the contact surface of the poor conductor [30, 125, 126]. To address this issue, the finite element mesh of the poor conductor is biased at 1.25 near the contact surface, and a mesh refinement exercise is conducted to ensure that the model
can reproduce a strong variation of temperature in the skin layer by estimating the temperature distribution perpendicular to the interface [30] as shown in Figure 4.5.

![Figure 4.5: Character of temperature distribution in the skin layer of the finite element mesh of the poor conductor](image)

Table 4.1: Shows the properties of the sliding materials used in this analysis

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Steel disc</th>
<th>Friction material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of Elasticity, E</td>
<td>210</td>
<td>6</td>
</tr>
<tr>
<td>Damping Modulus, (GPa)</td>
<td>-</td>
<td>0.2</td>
</tr>
<tr>
<td>Thermal conductivity,K (W/(mK))</td>
<td>57</td>
<td>0.1-1.0</td>
</tr>
<tr>
<td>Coefficient of thermal expansion</td>
<td>α11 (10^{-6}K^{-1})</td>
<td>12</td>
</tr>
<tr>
<td>Poisson ratio, ν</td>
<td>0.25</td>
<td>0.25</td>
</tr>
<tr>
<td>Specific heat, C (Jkg^{-1}K^{-1})</td>
<td>460</td>
<td>120</td>
</tr>
<tr>
<td>Mass density, ρ(kg/m^3)</td>
<td>7250</td>
<td>2000</td>
</tr>
</tbody>
</table>
Figure 4.6: Comparison of the effect of a) thermal conductivity and b) phase angle on the critical sliding speed of the viscoelastic material for different models.
4.2.3 Validation of the Model

4.2.3.1 Comparison of models

The validity of the results obtained from Equations (4.13) and (4.14) is confirmed through a comparison of the effect of thermal conductivity of the viscoelastic friction material on the critical sliding speed of the system with the results obtained from finite element simulations. Due to the range of physical parameters involved in this study, it is difficult and expensive to conduct experimental work. Therefore, the results are compared with Burton’s model, which is a well-known and experimentally validated model. The comparison is shown in Figure 4.6a and the material properties used in the analysis are presented in Table 4.1.

The results from the simulated finite element approach are in good agreement with the results obtained from the derived mathematical model. Furthermore, it is expected that the critical sliding speed may be decreased by the presence of the damping element in the viscoelastic model. Therefore, the results from Burton’s model are different from the other two models with viscoelastic parameters.

To further demonstrate the accuracy of the mathematical model, a comparison was made between the Decuzzi model and the developed mathematical model. In order to compare the two models, certain adjustments were made. Firstly, the thermal conductivity of the friction material in the developed model was set to zero \( (K_1 = 0) \), since the Decuzzi model ignores the effect of thermal conductivity of the friction material. Additionally, the developed model assumes that the surface displacement in the \( y \)-direction is zero \( (\mu|_y = 0) \), which implies that there will be no lateral expansion of the model. Therefore, the Poisson’s ratio \( (\nu_1) \) is nearly zero. Consequently, the smaller the Poisson’s ratio used in the Decuzzi model, the more it agrees well with the developed model.
Figure 4.7: The variation of dimensionless critical speed as a function of the thermal conductivity ratio for a) the viscoelastic friction material b) the pure elastic conducting material
The comparison between the two models is shown in Figure 4.6b. Note that a 1:1 ratio could be obtained when $\nu_1 = 0$. The results from Figure 4.6a and 4.6b give confidence in the output results from the developed mathematical expression, which can be used to investigate the critical sliding speed of viscoelastic friction material. This comparison provides further evidence that the developed mathematical model is accurate and can be used to predict the behavior of viscoelastic friction material under various conditions.

4.2.4 Parametric study of physical parameters

4.2.4.1 Variation of thermal conductivity ratio for different friction values

To investigate the relationship between the normalized critical sliding speed and other physical material parameters, the mathematical formulation is utilized. The procedure involves solving Equations (4.13) and (4.14) simultaneously, using a numerical approach. In this study, the sliding speed $V$ and wave parameter, $m$ are normalized in terms of the half-thickness, $a_2$ of the metal material and its thermal diffusivity, $k_2$.

The results of the investigation are presented in Figures 4.7a and 4.7b, which depict the variations of the dimensionless critical sliding speed in the viscoelastic friction material and the metal material, respectively. The figures show the impact of different ratios of the thermal conductivities at different friction coefficient values. The wavenumber represented by $m = 2\pi/L$, is fixed at 32, which is the upper limit for actual sliding bodies.

In the viscoelastic material, the normalized critical speed $Va_2/k_2$ is notably higher than that of the metal material when $K_1/K_2$ is varied from 0 to 1.
Figure 4.8: The variation of the dimensionless critical speed as a function of the wave parameter for a) different values of the relaxation time b) different values of the elastic parameter
This trend, originally observed in Burton’s linear elastic model [26], is also present in the viscoelastic model. However, the curve deviates significantly from other $\mu$-plots when $\mu$ is set to 0.2 for both materials. Upon closer inspection, it becomes apparent that as $\mu$ increases, the curves flatten out and exhibit a substantially reduced critical speed. These graphs suggest that the impact of thermal conductivity becomes less influential for larger $\mu$ values. Consequently, raising $\mu$ lowers the critical sliding speed of the system, potentially leading to issues such as buckling, hot spots, thermal fatigue, and vibration. Achieving sliding stability requires an appropriate balance of friction coefficient $\mu$ and thermal conductivities.

4.2.4.2 Variation of wave parameter for different relaxation times and elastic parameters

In order to understand the influence of the relaxation time $\zeta/E_1$ when $(K_1/K_2) = 0.02$ on the stability of the system, Figure 4.8a examines the variation of the dimensionless wave number, $ma$. It is worth noting that the damping effect during sliding increases with a higher value of $\zeta/E_1$. For the pure elastic solution, where $\zeta/E_1 = 0$, the normalized critical speed $V\alpha_2/k_2$ as a function of the wave parameter is linear and relatively high. This indicates a more stable system but may not necessarily minimize noise and vibration over time. Besides, when $\zeta/E_1 > 0$, a nonlinear reduction in the critical sliding speed as a function of wave parameter, $ma$ is observed. This becomes more pronounced as $\zeta/E_1$ is increased. Thus, it follows that the relaxation time decreases the effect of the wave parameter $ma > 0.2$ on the critical speed.

In Figure 4.8b, the dimensionless critical speed is plotted as a function of the wave parameter for various elastic parameters $E_1/E_2$ ($\zeta/E_1 = 0.02$).
Figure 4.9: The variation of the dimensionless critical speed as a function of the thermal conductivity parameter for a) different values of the relaxation time b) different values of the elastic parameter.
It is observed that as the elastic parameter $E_1/E_2$ increases, the critical speed $Va_2/k_2$ decreases significantly. This suggests that increasing the elastic parameter $E_1/E_2$ reduces the impact of the wave parameter on the stability of the system. Hence, it can be concluded that when the ratio of elasticity between the two sliding materials is relatively high, the stability of the system cannot be ensured.

4.2.4.3 Variation of thermal conductivity ratio for different relaxation times and elastic parameters

Figure 4.9 considers the variation of the normalized critical speed $Va_2/k_2$ with the thermal conductivity parameter $K_1/K_2$ at different relaxation times $\zeta/E_1$ and elastic parameters $E_1/E_2$. In Figure 4.9a, the significant impact of the relaxation time on the stability of the system as the thermal conductivity parameter $K_1/K_2$ is increased can be observed. The critical speed decreases as the relaxation time increases. For instance, at $(\zeta/E_1) = 0$, the resulting critical speed is found to be dominant. Further increase in $\zeta/E_1$ causes the overall critical speed $Va_2/k_2$ to reduce slightly. Similarly, in Figure 4.9b, as the elastic parameter is increased, the overall critical speed decreases. Therefore, it can be inferred that increasing the relaxation time $\zeta/E_1$ and the elastic parameter reduces the effect of the thermal conductivity parameter and makes the system more susceptible to instability.
Figure 4.10: The variation of the dimensionless critical speed with the elastic parameter at different a) relaxation times b) thermal conductivity ratios
4.2.4.4 Variation of elastic parameter at different relaxation times and thermal conductivity ratios

Figure 4.10a and Figure 4.10b show the variation of the normalized critical speed $V a_2/k_2$ with the elastic parameter for different values of the relaxation time parameter $\zeta/E_1$ and thermal conductivity parameter $K_1/K_2$, respectively. The critical speed decreases rapidly as the elastic parameter $E_1/E_2$ is increased.

This is intuitive because an increase in $E_1/E_2$ leads to higher stress levels, which explains the decrease in critical speed. Furthermore, increasing $\zeta/E_1$, as seen in Figure 4.10a, causes the curve of the critical speed as a function of the elastic parameter $E_1/E_2$ to reduce. On the other hand, by increasing the parameter $K_1/K_2$, as seen in Figure 4.10b, the curves of the critical sliding speed as a function of the elastic parameter increase.

Therefore, it can be inferred that increasing the thermal conductivity parameter of the viscoelastic material reduces the effect of the elastic parameter on the system stability while increasing the relaxation time increases the effect of the elastic parameter.
4.2.4.5 Variation of the relaxation time for different elastic parameter values and thermal conductivity ratio

The variation of the critical speed $V_{a_2}/k_2$ as a function of the relaxation time parameter $\zeta/E_1$ for the elastic parameters $E_1/E_2$ and thermal conductivity parameters $K_1/K_2$. Figure 4.11a illustrates the effect of the relaxation time parameter $\zeta/E_1$ on the critical speed $V_{a_2}/k_2$ for different elastic parameters $E_1/E_2$. For small values of $E_1/E_2$, the resulting critical speed is higher and is significantly influenced by the relaxation time. As shown in the figure, the critical speed decreases as the relaxation time increases. This is because the phase angle $\delta$ is highly influenced by both $E_1/E_2$ and $\zeta/E_1$ according to Equation 4.15. As the phase angle increases, the critical speed decreases.
Moreover, for relatively small values of $E_1/E_2$, there exist critical values of the relaxation time $\tau$ at which the critical speed starts to rise and fall. In other words, there is a range of values for $\zeta/E_1$ for which the critical speed is not monotonic. The behavior of the critical speed for small values of $E_1/E_2$ is more sensitive to changes in the relaxation time parameter compared to larger values of $E_1/E_2$.

When the elastic parameter $E_1/E_2$ is relatively small, the normalized critical speed is higher, but it is significantly affected by the relaxation time $\tau$. In Figure 4.11a, it can be observed that the critical speed decreases as the relaxation time parameter $\zeta/E_1$ is increased. This behavior can be explained by the fact that the phase angle $\delta$ is highly influenced by $E_1/E_2$ and $\zeta/E_1$ according to Equation 4.15. As the phase angle increases, the critical speed decreases. There are critical values of the relaxation time $\tau$ at which the critical speed starts to rise and fall for relatively small values of $E_1/E_2$.

The behavior of the normalized critical speed as a function of the elastic parameter $E_1/E_2$ and relaxation time parameter $\zeta/E_1$ is complex and varies with the values of the parameters. When $E_1/E_2 = 0.002$, the critical speed exhibits a rapid decrease from 1603.7 at $\zeta/E_1 = 0$ to 580.5 at $\zeta/E_1 = 1$. Subsequently, the critical speed gradually rises to a peak value of 808.5 when $\zeta/E_1 = 1.47$ before starting to decrease gradually with increasing $\zeta/E_1$. This trend is observed for all increasing values of $E_1/E_2$, except for when $E_1/E_2 = 0.012$. In this case, at $\zeta/E_1 > 1.91$, there is a sharp rise in the critical speed. This indicates that there is a specific critical value above which increasing $\zeta/E_1$ causes the critical speed to rise. This behavior arises because as $\zeta/E_1$ becomes larger and the phase angle approaches zero, the linear elastic problem is recovered. This may explain the sharp rise in critical speed as $E_1/E_2$ is increased to 0.012 when $\zeta/E_1 > 2$. 

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In Figure 4.11b, the curves exhibit a rapid decrease as $E_1/E_2$ is further increased with an oscillation trend. As $E_1/E_2$ increases, the profile becomes less pronounced and flattens to a more uniform profile. This phenomenon occurs due to a shift in the phase angle $\delta$, which increases as $\zeta/E_1$ increases, resulting in a reduction in the critical speed.

On the other hand, in Figure 4.11c, increasing the thermal conductivity parameter leads to an improvement in the critical speed. For small values of the thermal conductivity parameter (0.01-0.04), increasing $\zeta/E_1$ above 2 causes the critical speed to increase rapidly. However, when $K_1/K_2 > 0.04$, the reverse is observed. This phenomenon is attributed to the change in the phase angle $\delta$. These observations regarding the behavior of the viscoelastic friction material during sliding interactions are particularly useful in determining the threshold at which it can either stabilize noise and vibration or encourage the onset of instabilities.

4.3 Conclusion

The study aims to investigate the behavior of viscoelastic friction materials in brakes and clutches systems. To do this, a mathematical model was developed to predict the critical sliding speed of such materials, taking into account their viscoelastic properties. This model provides an estimation of when instabilities may occur during sliding interactions between a conducting material and a viscoelastic friction material, an area that has not been extensively researched.

In addition to the mathematical model, a finite element approach was established as an alternative means to estimate the critical speed of viscoelastic friction materials and to validate the mathematical model. The work investigated three physical material parameters of viscoelastic friction materials: relaxation time $\zeta/E_1$, elastic
parameter $E_1/E_2$, and thermal conductivity parameter $K_1/K_2$. The following deductions are drawn:

- The study found that the relaxation time parameter plays a significant role in reducing the critical sliding speed of the system. However, the effect of the relaxation time on the system depends on how it affects the phase angle, $\delta$ during sliding. For larger relaxation parameters, the phase angle may approach zero, which means that the linear elastic solution is recovered, potentially increasing the critical sliding speed. Therefore, a viscoelastic friction pad material may either encourage or discourage the onset of instability, depending on the resulting phase angle due to the relaxation time.

- Also, it was revealed that increasing the elastic parameter significantly reduces the critical sliding speed, making the system more susceptible to instability. This effect is due to the resulting peak contact pressures and stresses increasing, which also leads to thermomechanical instability.

- On the other hand, the thermal conductivity parameter significantly reduces the effect of the relaxation time and elastic parameters, leading to an increase in the critical speed and a reduction in thermomechanical instability. The study concludes that a reasonable thermal conductivity ratio is required for a stable system that discourages noise and vibration.

In summary, the study demonstrates that the appropriate combination and interaction of the physical parameters studied in this work are crucial in manufacturing viscoelastic friction materials for automotive disc brakes and clutches. The mathematical model and finite element approach developed in this study provide valuable tools for predicting the critical sliding speed of such materials and improving their design.
5.1 Introduction

As part of this research, an industrial training program was undertaken to gain insight into the braking tendencies of autonomous vehicles. Over a duration of seven months, a comprehensive study was conducted at CalmCar Vision System LLC, focusing on the Advanced Parking Assist (APA) system of autonomous vehicles and instabilities during braking. The study delved into various parking scenarios, including parallel parking, angle parking, and perpendicular parking, which are illustrated in Figure 5.1.

The hardware configuration of the parking system encompasses a total of 12 ultrasonic sensors (USS), distributed as 8 short-range USS for proximity object detection and 4 long-range USS for detecting objects at a distance from the vehicle. These ultrasonic sensors establish connectivity with the Electronic Control Unit (ECU) through a conversion box, which facilitates the conversion of USS signals into the CAN signal format. Additionally, the ECU interfaces with four fish-eye cameras responsible for visual perception. To power these devices, a dedicated power supply unit is employed, and connectivity to a WIFI system is established via a router.
The visual representation of the hardware architecture of the Advanced Parking Assist (APA) system is depicted in Figure 5.2.

Figure 5.1: Parking Scenarios a) perpendicular parking with vision perception b) angle Parking with vision perception c) perpendicular parking with ultrasonic sensors d) parallel parking with ultrasonic sensors

Figure 5.2: Hardware components of the APA system
5.2 Estimation of the rotor and brake pad temperature

A modern method for estimation of rotor and brake pad temperature was studied during the Testing Expo North America 2022 held in Michigan while at CalmCar. A rubbing thermocouple sensor and an embedded brake pad thermocouple were used to determine temperature in brake rotor and friction brake pads respectively. The rubbing thermocouple sensor as shown in Figure 5.3a is installed such that the sensor tip is lightly touching the rotor surface to sense the temperature at a specified path during rotation. For the embedded brake pad thermocouple Figure 5.3b, a hole is drilled in the pad from the backplate side. The thermocouple wire is then routed through the created hole from the braking pad surface side.

![Figure 5.3: set-up of the rubbing thermocouple sensor on a rotor and the embedded thermocouple sensor in a brake pad b) image of the rubbing thermocouple sensor](image)

5.3 Effect of variable speed on friction coefficient

Most research work on brake analysis assumes that the disc rotor rotates under constant speed and therefore neglects the effects of variable speed. In the present
work on thermomechanical instabilities, constant velocities were considered in all analyses, which is not the case in the real world. The industrial training at Calmcar provided the opportunity to consider variable speed during braking analysis. At CalmCar, parameters such as the vehicle speed, distance traveled, brake torque, etc. of vehicles could be determined using a vector canoe. However, it was a bit challenging to effectively measure parameters such as the temperature distribution and friction coefficient directly from the vehicle during braking.

For this reason, a simulation platform was provided by the company to study the effect of variable speed on friction coefficient. CarMaker software was utilized to simulate the braking performance of a Tesla S vehicle. Table 5.1 shows the basic parameters of the Tesla S vehicle.

<table>
<thead>
<tr>
<th>Vehicle parameter</th>
<th>Value, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unloaded weight</td>
<td>2108.0</td>
</tr>
<tr>
<td>Vehicle length</td>
<td>4976.0</td>
</tr>
<tr>
<td>Vehicle width</td>
<td>1963.0</td>
</tr>
<tr>
<td>Vehicle Height</td>
<td>1435.0</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>2959.0</td>
</tr>
<tr>
<td>Track width front</td>
<td>1661.0</td>
</tr>
<tr>
<td>Rear overhang</td>
<td>1009.6</td>
</tr>
</tbody>
</table>

The main objective was to understand how different road conditions affect the dynamic friction coefficient between the rotor disc and the brake pads during braking. In addition, to study how variable speed impacts the development of thermomechanical instability during braking, the vehicle moving on a straight road under variable speeds while applying multiple braking for 73.0s was considered. From
the simulation, it was possible to measure all the required parameters such as the braking torques and the vertical forces acting on each wheel.

Figure 5.4: a) Initial set-up velocity profile for the vehicle b) Applied brake pedal profile with time for the vehicle
However, the simulation does not provide a direct means to measure the squeezing force between the brake pads and the rotor of the vehicle during braking. But we can approximate the squeezing force $F_S$ of one wheel by equating it to the braking force $F_B$ acting on the wheel. Thus, the force required to stop the wheels. Moreover, knowing the vertical force or weight acting $F_N$ on the wheel and the squeezing force $F_S$, the sliding friction coefficient $\mu_s$ between the rotor and the pad can also be estimated. Considering the above-mentioned conditions, CarMaker Simulink vehicle model was utilized to compute the friction coefficient during braking.
Figure 5.5: Output result from vehicle for different road conditions a) traveling speed with time b) sliding friction coefficient with time c) vertical force acting on the front left wheel with time

To investigate the influence of different road conditions on the sliding friction coefficient, road friction $\mu_r$ values of 0.02, 0.2, 0.5, and 1 were considered. To begin the analysis, an initial variable speed profile with time was defined for the vehicle as shown in Figure 5.4a. The applied brake pedal distance during the traveled time is shown in Figure 5.4b.

Figure 5.5a shows the vehicle’s moving speed under different road conditions. It was observed that the vehicle traveling speed reduces significantly as the road friction value decreases. This is obvious as there will be slipping and skidding at very low road friction which will affect the horizontal motion of the vehicle. Figure 5.5b shows the resulting sliding friction coefficient $\mu_S$ between the brake rotor and the brake pad for braking under different road friction conditions. It was observed that when the brake pedal is applied at a constant period, the resulting sliding friction coefficient becomes unstable when the road friction is small. For example, when the brake pedal distance was more than half-pressed, (0.55) from 22.0 s to 27.0s, the resulting sliding friction
coefficient stability decreases as the road friction decreases. For a road friction value of 1.0, the vehicle required less sliding friction to slow the vehicle. Meanwhile, for a road friction value of 0.02, the friction value increased for the specified time and was unstable. In addition, the dynamic weight transfer during the braking period for the different road conditions as shown in Figure 5.5c increases with increased road friction. This is because there is enough contact between the road and the tire, which eliminates lateral and longitudinal slipping.

5.4 Conclusion

From this work, it can be deduced that the resulting sliding friction between the brake pad and the disk does not depend entirely on the brake friction materials. It is greatly influenced by the road condition and the traveling velocity profile. Variable vehicle speed and road friction conditions play an essential role in the braking performance of the vehicle as seen in this study. Most research work on brake analysis neglects these two factors. A variable speed can greatly influence the resulting sliding friction between the brake pad and the brake rotor. This is also influenced by the road conditions as revealed by this study.
6.1 Conclusion

Experimental and numerical investigations have been carried out on a metal-based friction material to examine various factors that may trigger the onset of instabilities such as thermal buckling, wear, and hotspots when used in high-speed sliding systems. The insights gained from the study are utilized to computationally explore effective strategies for mitigating instabilities that may rise in next-generation friction material when used in speed-sliding systems.

The approach involved preparing copper-based friction material by the powdered metallurgy technique by utilizing unique chemical compositions. The friction characteristics and wear rate of the copper-based material when sliding against 65Mn steel were evaluated using a Universal Mechanical Tester. Two operating variables, sliding speed, and temperature, were considered during the engagement process of the friction pairs. The dynamic friction coefficient was translated into friction torque capacity with time using the normal applied force and dimensions of the clutch disc. The results showed that instability can be excited at low operational conditions, where the friction coefficient is high. The dynamic friction torque oscillates with time at
temperatures of 25°C and 400°C. Generally, more stable friction torque was obtained when the sliding speed was varied, as opposed to varying the temperature.

The second consideration was the influence of operating temperatures and sliding speeds on thermal buckling and thermoelastic instability of the friction disc. The onset of thermoelastic instability occurred when the sliding speed exceeded 200rpm. Thermal buckling was highly dependent on the temperature difference between the inner and outer radius of the friction disc.

To understand how metal-free materials respond to TMI in order to enhance their performance and minimize the emission of hazardous particulate matter into the environment, a two-dimensional non-linear transient thermo-mechanical model was developed. The model incorporated the anisotropic properties of a metal-free friction material sliding against a steel disc using a finite element approach. The effect of the material anisotropy of the friction material was studied when used in disc clutch or brake. The model was validated by comparing the result to an existing eigenvalue approach. A parametric study on the material anisotropy revealed that elastic modulus $E_{11}$ in the sliding direction and thermal conductivity $K_{22}$ in the thickness direction discourage thermo-mechanical instability (TMI) formation for contact with separation effect. Conversely, thermal conductivity $K_{11}$ in the sliding direction and elastic modulus $E_{22}$ in the thickness direction make the friction material more susceptible to TMI formation for full-contact regime and contact with separation effect. Additionally, analysis was carried out on the thermal expansion coefficient to determine how it influenced TMI in metal-free friction materials.

In addition, a viscoelastic friction material was proposed to tackle instabilities such as noise, vibration, and squeal. A mathematical model was developed to predict the onset of instability in such materials during sliding interactions.
The model was derived from and expanded upon the fundamentals of Burton’s model for thermoelasticity in pure elastic materials. The study considered three physical material parameters: relaxation time, elasticity, and thermal conductivity. Prior to the parametric study, the effects of the material properties in relation to thermoelasticity were not fully understood. Therefore, a finite element analysis was developed and used to validate the mathematical model by comparing the variation of the critical speed as a function of thermal conductivity. The results revealed that an increase in the relaxation time significantly reduces the critical sliding speed. Changing the elastic parameter further increases the effect of relaxation time by also reducing the critical sliding speed. However, increasing the thermal conductivity parameter dampens the effect of the elastic parameter and relaxation time on the critical speed.

The study concludes that there is a critical value of the relaxation time and elastic parameter above which the system stability is improved, meanwhile thermal conductivity attempts to counter the stability gained from other material properties. The study is instrumental in understanding the influence of viscoelastic parameters in sliding systems and provides an intuitive means of predicting the onset of thermomechanical instability.

6.2 Future Work

6.2.1 Next-Generation friction materials in Electric Vehicles

The traditional way of operating vehicles with internal combustion engines is becoming less and less prevalent, sparking discussions about the role of friction materials in electric vehicles (EVs). Some people speculate that with the presence of regenerative braking in EVs and autonomous vehicles, friction materials will become entirely unnecessary.
However, the author of this argument asserts that until the longstanding issues with regenerative braking are resolved, it will be impossible to completely eliminate the need for friction pads.

Currently, manufacturers of EVs still include friction brakes in regenerative brake systems to ensure that the vehicle can stop accurately when necessary. More importantly, for EVS, the battery must be below approximately 80% state of charge in order to fully utilize the potential of the regenerative braking system for accurate braking. In situations where the battery is fully charged, the driver must rely solely on the friction brake pad.

Furthermore, it is worth noting that electric cars have a higher torque from the start compared to internal combustion engines, which require some time to reach a specific torque after gaining speed. This means that friction pads still play an essential role in ensuring smooth and safe driving experiences for EV users, especially during sudden stops or emergency situations at high speed.

6.2.1.1 Problems with corrosion of rotor and possible fast wear of friction materials

In electric vehicles (EVs), regenerative braking is utilized at all times except when it is insufficient to bring the vehicle to a complete stop. In such situations, traditional brakes are used to supplement regenerative braking. However, as a result of the extended periods when the rotor and pad are not utilized, the surface of the rotor may start to corrode. This corrosion can cause other possible instabilities and may impact the longevity and performance of the friction pad.

Additionally, EVs and autonomous vehicles rely on various sensors, such as cameras, LIDARS, and radar, to detect obstacles, monitor the surrounding environment, and gather data to make informed decisions about braking. While this process does not directly impact the performance of the friction pad, it can influence braking behavior.
and affect the interaction between the pad and rotor, potentially leading to TMI (thermal-mechanical instability). For instance, if these sensors detect an obstacle near the front of the vehicle at high speed, they may signal the brake system to engage quickly and forcefully, causing increased wear on the brake pads. Similarly, if a slippery road is detected, the braking system may engage in a way that maximizes traction and minimizes skidding, potentially impacting the performance of the friction pad and increasing wear rates.

Given these factors, it is essential to study how Advanced Driver Assistance Systems (ADAS) interact with the braking system to ensure optimal performance and longevity of next-generation friction pads. This research will help to improve the performance of friction pads and ensure the safety and reliability of EVs and autonomous vehicles.

6.2.1.2 Problem with variable speed

Up until now, all of the analyses of Thermo-Mechanical Instability (TMI) conducted in this study and in other works have been based on the assumption that the rotor disc or clutch disc rotates at a constant speed. However, in reality, this is not always the case, unless there is a malfunctioning clutch. The variable speed of the disc rotor or clutch disc may significantly impact the occurrence of TMI compared to when a constant speed is assumed. Thus, it is crucial to take into account the variable speed of the rotor disc in future studies on TMI in order to depict real-life situations. By doing so, a more accurate analysis can be conducted, leading to a better understanding of the behavior of the friction materials under different conditions, and ultimately leading to the development of more reliable and durable materials for use in various applications. Therefore, it is essential to consider variable speed in future work on TMI.
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